CRANFIELD UNIVERSITY

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TECHNO-ECONOMIC ANALYSIS OF GAS TURBINE COMPRESSOR WASHING TO COMBAT FOULING

SCHOOL OF ENGINEERING

MSc
Academic Year: 2011 - 2014

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ABSTRACT

Among the major deterioration problems a gas turbine encountered while in operation is compressor blade fouling. This is the accumulation and adhesion of dirt and sediment on the compressor blade which contributes between 70 to 85% of gas turbine performance loss. Fouling reduces turbine air mass flow capacity, compressor pressure ratio and overall gas turbine efficiency. In most cases, its effect does not manifest immediately in gas turbine power output and efficiency since they are not measured directly. However, it is apparent on the gradual increase in Turbine Entry temperature (TET) and Exhaust Gas Temperature (EGT). More fuel is burnt in the combustion chamber to maintain turbine power output which leads to high combustion flame temperature and thus reduces creep life of hot components.

This research seeks to analyse the technical and economic consequences of compressor fouling in overall gas turbine performance. The work begins with simulation of TS3000 engine and examination of its design and off design performance. Subsequently, medium size gas turbine engine was modelled, simulated and its performance at different condition was examined to validate the outcome of field data analysis.

Three months field operating data of Hitachi H-25 gas turbine generator used for power generation at bonny oil and gas terminal in Nigeria was collected and corrected to international standard ambient condition, using thermodynamic calculations. These data were analysed to determine the effect of fouling on the engine fuel consumption, power output in order to determine the plant profitability.

The above analysis gives an estimation of fuel cost saving benefit of $41,000 over the period of one year plant operation due to regular two weekly compressor online water wash which is a good indication of the engine efficiency.

Keywords: Compressor Fouling, Thermodynamic condition, Degradation, ISO, Turbine Entry Temperature

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LIST OF ABBREVIATIONS

HEPA	High Efficiency Particulate Arrestance
EPA	Efficient Particulate Arrestance
SGT	Siemens Gas Turbine
GE	General Electric
UCP	Unit Control Panel
OEM	Original Equipment Manufacturer
NDMF	Non Dimensional Mass Flow
PR	Pressure Ratio
DP	Design Point
MW	Mega Watt
K	Kelvin
TET	Turbine Entry Temperature
N	Non Dimensional Rotational Speed
BOGT	Bonny Oil and Gas terminal
CCGT	Combine cycle gas turbine
CW	Compressor work
EP	Engine parameter
COT	Combustion Outlet temperature
SFC	Specific Fuel Consumption
ETA	Isentropic efficiency
FOD	Foreign object damage
GE	General Electric
H	Number of particles colliding with surface
HPT	High pressure turbine
L	Number of particles that fall to the surface
LMP	Larson Miller Parameter
MW	Megawatt
NLNG	Nigeria liquefied Natural gas
OEM	Original equipment manufacturer
Qcc	Combustion chamber heat
QCC	Heat in Combustion chamber

Qr	Heat rejected
RE	Real Engine Data
SE	Simulated engine
SW	specific work
TET	Turbine entry temperature
Tw	Turbine work
UW	useful work
EOH	Equivalent operating Hours
FO	Force outage hour
R	Reliability

1 INTRODUCTION

As demand for energy in the world is continuously rising, gas turbine production has been witnessing steady rise, particularly in non-aviation sector over the past years and the spate is expected to continue to the near future. This is due to its small weight to power ratio, lower fuel cost and pollution emission. Thus, efforts have been channel to enhancing and sustaining gas turbine performance in other to retain its relevance.

However, Compressor fouling poses a major and prevalent problem to effective and economic advantage of operating industrial gas turbine [1] in fact it contributes up to three quarter of gas turbine performance loss [2].The steady rise in the price of fuel globally due deregulation of energy sector has made every gas turbine operator pay special attention to efficient utilisation of fuel in order to get good financial return from their investment [3].

Compressor washing has been identified as an effective method in controlling gas turbine performance deterioration. Since, it has been identified that 70-85% of gas turbine performance degradation is caused by fouling [4] then it is a worthwhile venture to research and figure out simple and efficient method to combat fouling.

Operators and manufacturers of gas turbine have recognised the economic loss incurred by gas turbine poor performance, its adverse effect on the engine components integrity which compromises turbine availability and reliability being major advantage gas turbine has over other prime movers. Industrial gas turbine is designed to be in service for about 100,000hours without major maintenance activities. However, fouling could prolong equipment mean time to repair (MTTR) if major gas turbine components like axial compressor, power turbine which could not be easily purchase off self are allowed to terribly deteriorate. Though the Mean Time between Failure (MTBF) of major gas turbine components are designed to be longer but the equipment life could be further elongated by regular online washing to take care of fouling.

This work investigates effectiveness of online compressor washing and its techno-economic benefit. Literature review on gas turbine performance and compressor washing was done while turbo-match simulation software was used to investigate the overall gas turbine performance after compressor water wash activity was done and its effects on various key components of gas turbine. Hitachi H 25 field data has been collected and analysed to investigate the engine performance at degraded condition and economic implication of running fouled engine. Literature review on proactive method of reducing fouling through effective filtration system is done and special feature of H25 engine was highlighted.

1.1 Aim and Objectives

The aim of this research is to analyse technical and economic consequences of compressor fouling in overall gas turbine performance. The specific objectives are:

- To investigate compressor fouling phenomenon and its effect on gas turbine performance
- To Develop model approach for simulation of compressor fouling with the use of Turbo-Match software
- To analyse real engine data with a view to study its overall performance
- To simulate compressor washing with Turbo-match and analyse its effectiveness
- To compare clean, fouled and washed engine performance
- To evaluate economic benefit of frequent compressor online washing, complemented with offline washing

1.2 Thesis Structure

The entire work is covered under six chapters while each of the chapters is well connected towards achieving the aims and objectives of the work:

Chapter 1: Introduction

In this chapter gas turbine performance problems are defined, its economic and environmental effect is stated, the aim and objectives of the research work are presented and the relevance of compressor washing as solution to the gas turbine problem.

Chapter 2: Literature Review

This chapter contains classification of gas turbine performance degradation, degradation mechanism, effect of degradation on turbine components and overall engine performance, gas turbine reliability, availability, engine proactive protection methods and compressor washing methods.

Chapter 3: Gas Turbine Performance Simulation

This chapter entails the use of Turbo-match software tool to demonstrate gas turbine performance understanding, design point and off design performance of engine model simulation, discussion of results of turbine performance due operating and environment changes

Chapter 4: Engine running parameters Data Analysis

This chapter contains plant overview and Hitachi H25 engine description, data investigation at different periods (1584-7848) which involves engine performance without water wash compared with the engine performance after frequent water wash.

Chapter 5: Economic implication of running deteriorated engine and effectiveness of compressor washing

The engine economic performance at design stage is examined in terms of power produced and fuel consumption. This is compared with real time performance considering that the engine is operated mostly at part load. The cost implication of frequent compressor washing, taking into account the washing fluid cost.

Chapter 6: Conclusions and Recommendations

This chapter contains brief summary of research work, findings and suggestions on how to obtain optimum performance from gas turbine

2 LITERATURE REVIEW

2.1 Gas Turbine Performance Degradation

Gas turbine like any other prime mover undergo wear and tear over life time which reduces its thermal efficiency, increases overall maintenance and operating cost as result of increased fuel consumption and increase in pollution emission due rise in TET. Overall gas turbine function is as a result of smooth interrelationship of gas path components which includes compressor, compressor turbine and power turbine. Gas turbine has to operate in an environment which deteriorates the behaviour of the gas path components due to the presence of solids and liquid particles in form of aerosol and this affect the matching of components [11].

2.1.1 Mechanism of Degradation

The understanding of the phenomenon of degradation of gas path components is important to the operators and manufacturers of gas turbines. The most prevalent form of degradation experience by gas turbine engine is fouling, which may not be immediately noticed in the engine performance but could cause compressor surge or prevent engine from starting if allow to build up and occupy gas flow path. Fouling and other form of degradation mechanism are explained in detailed under compressor fouling, component erosion and hot corrosion sub headings.

2.1.1.1 Compressor Fouling

Fouling is adherence of airborne particles in the atmosphere to the compressor aerofoils and annulus. Particles in form of smoke, dust, carbon, oil mist, insect, sea salt are common example. Figure 1 shows accumulated dirt on turbine inlet blade (IGV) taken from field during maintenance activities. Ingestion of flare, exhaust from other gas turbine and oil mist from other producing process are considered and avoided in the installation and plant layout. Frequent changes in wind speed, wind direction and high level of humidity in the environment makes particles stick and build up on the aerofoil resulting in reduction of air flow capacity and plugging of turbine blade cooling holes which would result into high turbine entry temperature TET and promotes overheating. Particles that cause fouling are typically smaller than between 2µm to 10µm [6]

Figure 1: Picture of dirty Compressor (SGT400) Inlet Guide Vane [48]

2.1.1.2 Compressor Fouling Mechanism

Atmospheric air contains suspended particles which move randomly and collide continuously with air molecules and component around the compressor blade and annulus. Particles travel to the surface of compressor blade by diffusion, impaction, settling and interception process in accordance with Brownian motion which are represented in the Figure 2 [17].

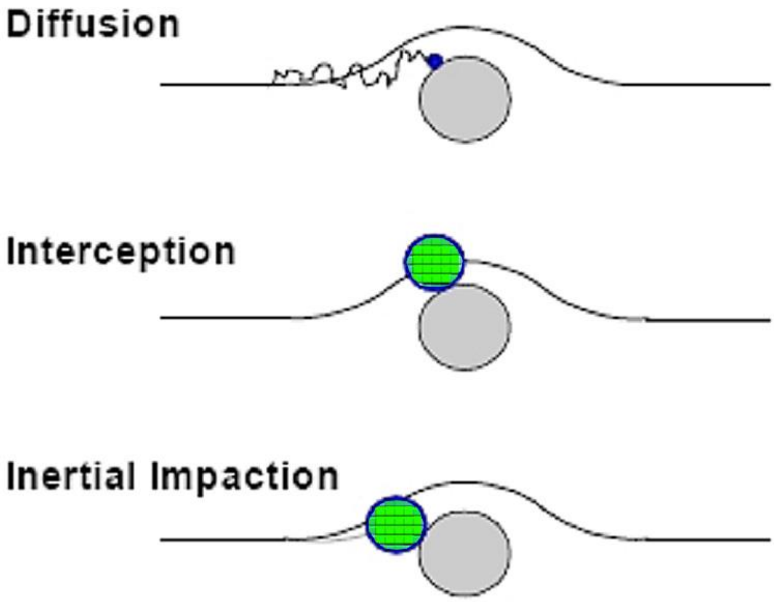

Figure 2: Axial Compressor Particle Deposition Mechanisms [18]

2.1.1.3 Fouling Influencing Factor

Compressor fouling and the rate of fouling are influenced by the factors discussed below:

- **<u>Gas turbine design parameters</u>:** Design parameters such as inlet velocity, mass flow rate, compressor pressure ratio, inlet guide vane, aerodynamics and geometric characteristic, blade geometry and angle of attack determine the sensitivity compressors to fouling. The higher the mass flow the greater the tendency of particle injection into the compressor[19]

- **Gas turbine operating area:** Different climatic conditions influence gas turbine fouling rate. Turbine performance varies due to location (dusty, windy, temperate, tropical region) in which it is operated. Gas turbine located in a heavily industrial area tends to be affected by emissions from surrounding equipment.
- **Plant Design and layout:** consideration is often given to the height of air filter installation, wind direction and exhaust emission from adjacent machinery and heavily motorised area in gas turbine installation. This is done to take care of identified source of fouling and eliminate it at plant layout stage.
- **Plant maintenance Culture:** Maintenance practises deployed in plant and quality of spares used determines gas turbine performance efficiency and availability. A plant where preventive maintenance is held in high esteem will likely have minimal compressor fouling problem, and plant availability tends to be higher.

2.1.1.4 Component Erosion

Erosion is removal of metal components along the gas flow path by particles bigger than 10µm in size. Erosion problem has been taken care off by the state of the art filtration system installed in industrial gas turbine but occur occasionally after maintenance activities when solid object are left in the engine e.g. bolts or tools used or it is as a result of droplet of water which is incompressible in the compressor part of engine. It as well occurs when sudden removal of turbine component happened within the engine which would wreak havoc on the entire engine.

2.1.1.5 Hot Corrosion

Hot corrosion is chemical reaction of certain contaminants like reactive gases, salt, mineral acid with flow path components which cause flow path components deterioration. Scale is formed around the flow path component as a result of the chemical reaction. Chemical reaction also occurs between atoms of metal components around the hottest area-combustion chamber and oxygen known as Temperature oxidation resulting into mechanical failure like cracks and spalls. Figure 3 depicts the effect corrosion hot corrosion on Turbine blade [23; 24]

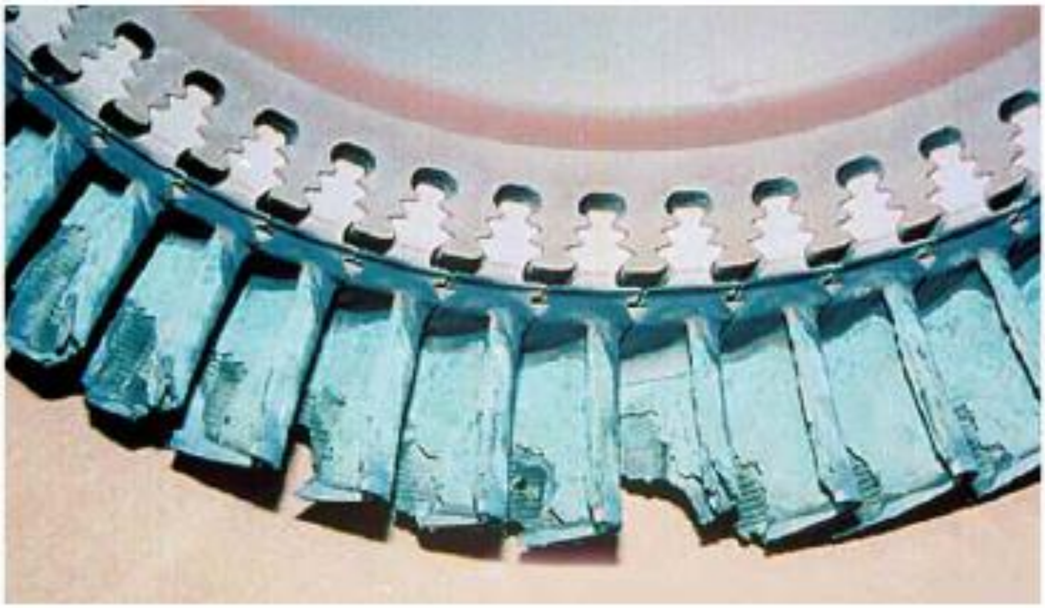

Figure 3: Hot Corrosion Effect on Turbine Blades [11]

2.1.1.6 Corrosion

Corrosion causes engine degradation due to chemical reaction between the contaminants in the inlet air, fuel gas and combustion derived contaminants. Sodium and potassium are major cause of corrosion even though lead and vanadium are found as culprit as well. Sodium chloride is an impurities present in air and posed major problem for engine operating in offshore because sea water contains large amount of salt. Once the sodium impurity exceeds the tolerance considered by the manufacturer, then it becomes a problem which continues unabated even after the source is eliminated.

Small particles of between 2µm to 10µm get into engines despite the filtration process fuses to the gas path components and block cooling air passages, alter surface shape, severely interfere with heat transfer and leads to thermal fatigue.Table1 displayed list of common trace metal elements found fuel, tolerable limit and their effect on gas turbine components.

Table 1: Trace Metal Specification and Effects

Trace Metals	Limits in Raw Fuel	Effects in Turbine	Types of Treatment	Typical Limits in Fuel to Turbine
Sodium Plus Potassium	150ppm	High Temperature	Fuel Wash	1ppm
Calcium	10ppm	Fouling Deposit	Fuel Washing to a Limited Extent	10ppm
Lead	1ppm	High Temperature corrosion	Inhibited by Magnesium	1ppm
Vanadium		High Temperature Corrosion	Inhibited by Magnesium	0.5ppm

2.1.2 Recoverable and Non-Recoverable Degradation

Mechanism of degradation discussed above can be broadly categorised into recoverable and non-recoverable degradation. Degradation that can be salvaged or reversed by either online wash or off-line wash is regarded as recoverable. Degradation caused by fouling is usually reversed by compressor cleaning or water wash while degradation that require repair, replacement of engine component or complete engine overhaul is regarded as Non-recoverable.[3] Erosion, corrosion, abrasion ,hot corrosion and foreign object damage result in non-recoverable engine deterioration. Figure 4 shows non recoverable degradation

Figure 4: Cracked Stator blade on SGT400 [49]

2.1.2.1 Non Recoverable Performance Curve

Due to wear and tear experience by the engine components as a result of frequent start up, running and engine shut down, the engine performance output drops as the number of engine fire hours increases.

The graph in figure 5 is useful in estimation of compressor performance loss expected after the first engine overhaul which occurs when an engine has been in service for about 5 to 6 years equivalent of 43000 to 50000 running hours [22]

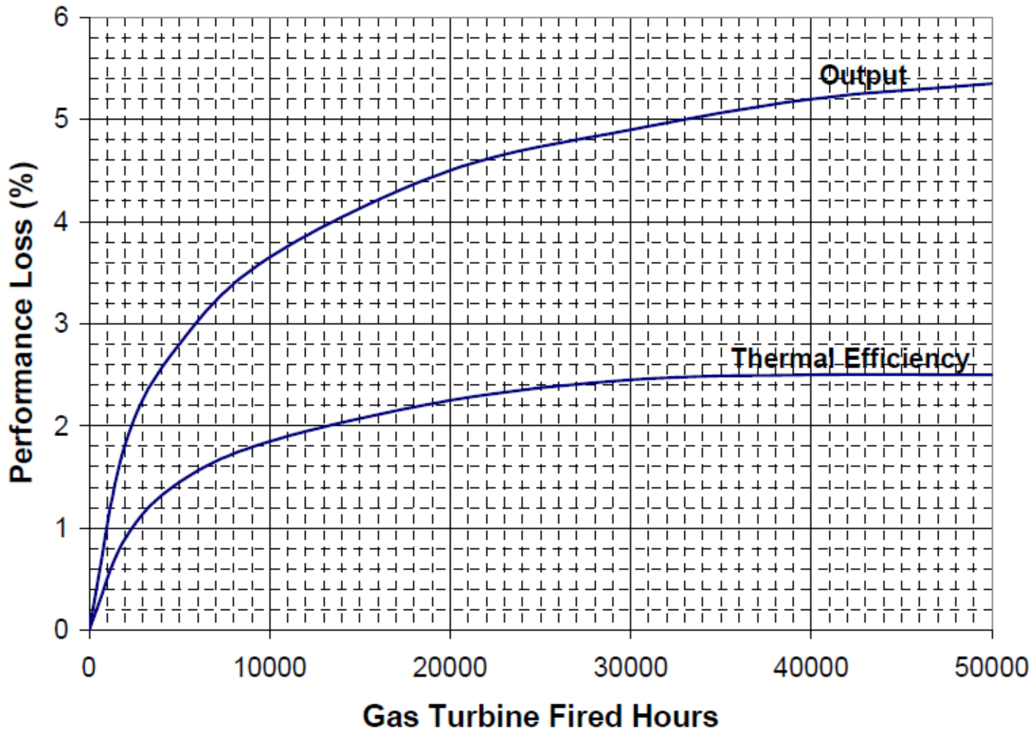

Figure 5: Estimated non recoverable performance loss [22]

2.2 Effect of Compressor Fouling on Gas Turbine Performance

The gas turbine compressor utilises nearly 60% of the work produced by gas turbine thus dirt accumulation on compressor results into significant drop in engine performance. Compressor fouling reduces compressor flow capacity, pressure ratio and efficiency which in turn lead to poor overall turbine power output. Higher compressor pressure ratio engine are more susceptible to fouling which could lead to surge and blade failure. The effect of fouling on compressor is more pronounced on high compressor pressure ratio machine [11; 25]. The higher the pressure ratio of an axial compressor the greater its reduction in overall thermal efficiency due to fouling effect shown in Figure 6 [20; 25-28]:

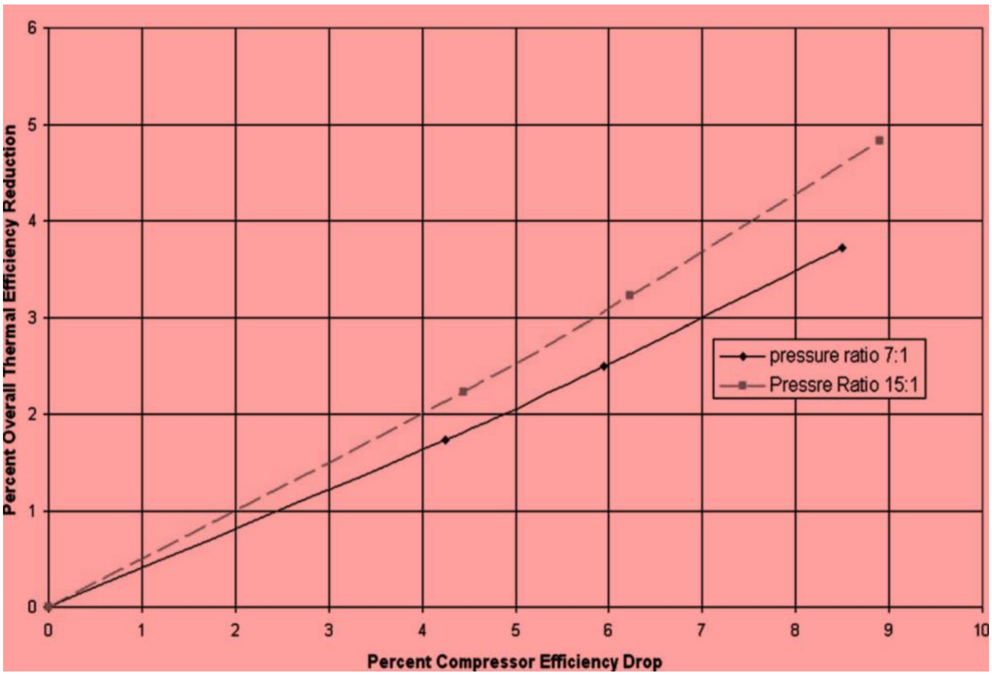

Figure 6: Effect of Compressor Efficiency on Gas Turbine Performance [11]

2.3 Gas Turbine Performance Recovery

2.3.1 Introduction

Gas turbine performance degradation is inevitable like other machinery that operates continuously in an environment that is filled with all sought of impurities. However, with appropriate air inlet filtration system and schedule compressor water wash in place gas turbine performance optimisation could be achieved. Optimum performance of gas turbine would results into improved power output, reduced heat rate, improved engine life cycle and reduced maintenance cost [2; 2].

2.3.2 Air Inlet Filtration System

Gas turbine inlet filter is a very important component of air inlet filtration system that prevents admission of dirt be it in form of solid, dust, particles, insect, exhaust fume and aerosol from entry gas turbine inlet. An effective air inlet filtration system will increase turbine availability and reliability thereby reduces maintenance cost and maintains turbine performance optimally.

However, overtime dirt accumulation on filters results into filter fouling which often manifest from increase in air inlet filter differential pressure which reduces mass flow of air into turbine as a result of pressure loss. The reduction in mass flow of air results into reduced power output and engine efficiency.[3] Figure 7 below shows the impact of air inlet pressure loss as a result of fouled filter on engine heat rate and power output.

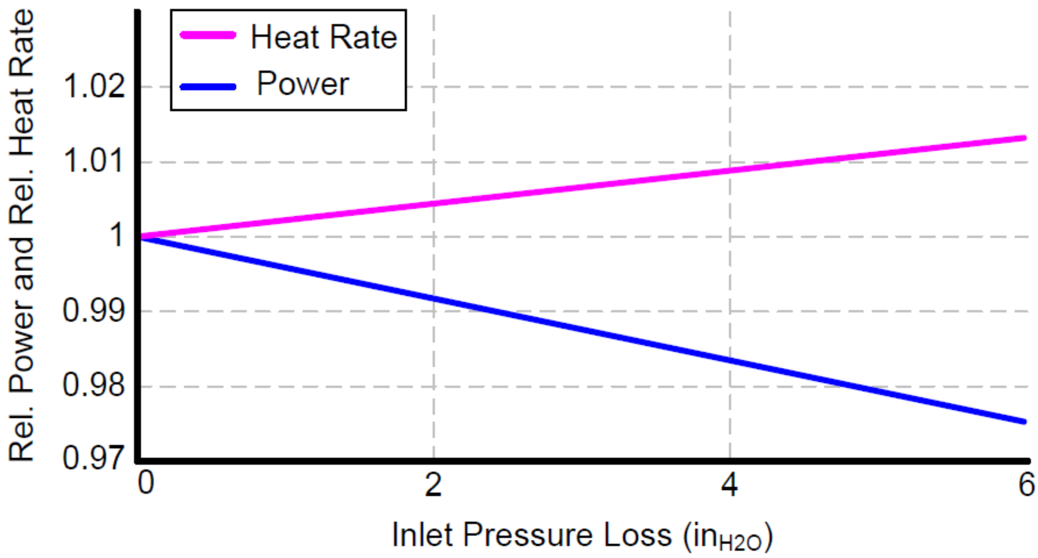

Figure 7: Effect of Fouled Filter on Engine Performance [4]

2.3.3 Gas Turbine Filtration System

Due to the importance of filtration system in gas turbine, different stages of filtration are usually available on gas turbine inlet system. Hitachi 25 and Siemens SGT 400 are provided with 3 different stages of filtration as highlighted below:

1. **Weather Protection and Trash screen:** weather louvers, hood and trash screen reduce amount of moisture and particles particularly large particles that may accompany air flow and it's usually installed in high

efficiency filtration system. Figure 8 below indicate Siemens SGT 400 weather hood.

Figure 8: Gas Turbine Inlet Filter Weather Hood [4]

2. **Pre-filter**: This type of filter was not initially installed but due to frequent maintenance of filter cartridges as a result of pressure loss due to dirt accumulation, pre-filter was latter installed which extend filter service period and thus reduces maintenance cost. Effect of high pressure drop across filter and rise in other environmental parameters like ambient pressure, temperature and relative humidity cause reduction in turbine power out and increase in turbine heat rate as shown in the table 2.

Table 2: Effect of parameter change on turbine power output and Heat Rate [5]

Parameters	Parameter change	Power output	Heat Rate
Ambient Temp	$10^{o}C$	-6.5%	2%
Ambient Pressure	10mbar	0.9%	0.9%
Ambient relative Humidity	10%	-0.0002%	0.0005%
Pressure drop in Filter	25mmWC	-0.5%	0.3%

Pre-filter is made up of disposable fibres which removes large particle of between 2-10 microns from flow stream.

3. **High Efficiency Filter**: Cartridge type high efficiency filter as shown in figure 9 below are provided for last stage of filtration to prevent particles that could cause compressor fouling and plugging of blade cooling passage from entry into the turbine. High efficiency filter particularly cartridge type has high dust holding capacity with filtration efficiency of between 80-99% for EPA and HEPA filter type respectively.

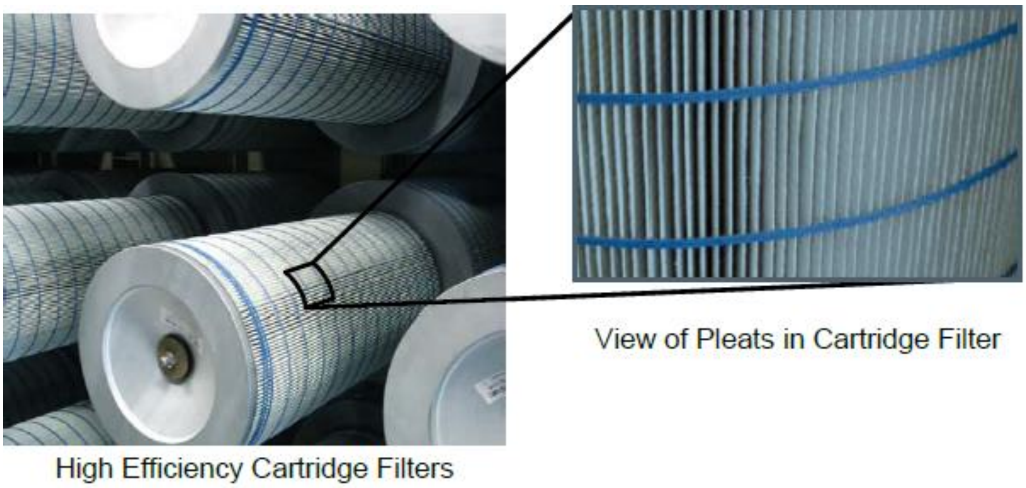

Figure 9: Cartridge Type High Efficiency Filter [4]

30The cartridge filter installed on H25 and Siemens gas turbine is surface type with self-cleaning system.

2.3.3.1 Self-Cleaning Filtration System

Due to high dust rate, to prolong filter service life and reduce maintenance cost of filter replacement, a system whereby the filter cleans its self when certain pressure loss as being attained. A reversed pulse compressed air of between 80 t0100 psig cleans the filter by blown off the dirt accumulation, this occurs between 100ms to 200ms and only about 10% of dirt is blown off to avoid flow disruption and reduce the frequency of the use of compressed air. The principle of operation of self-cleaning filter system is shown in Figure 11. Of course, the filter has it service life about 2 year which would necessitate filter outright replacement and in effectiveness of filter would manifest in efficiency and increase in frequency of pulse system operation as figure10.

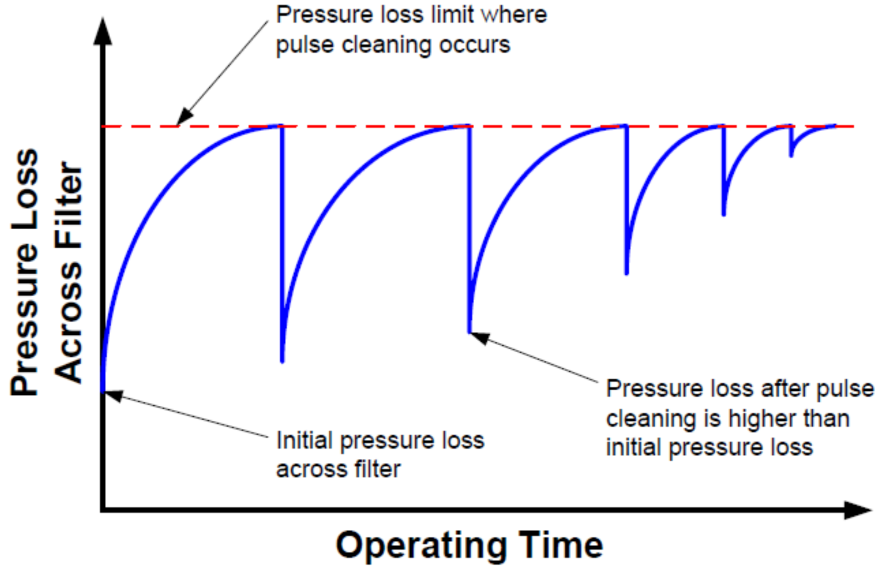

Figure 10: Pressure Loss Curve on Self-Cleaning Filter [4]

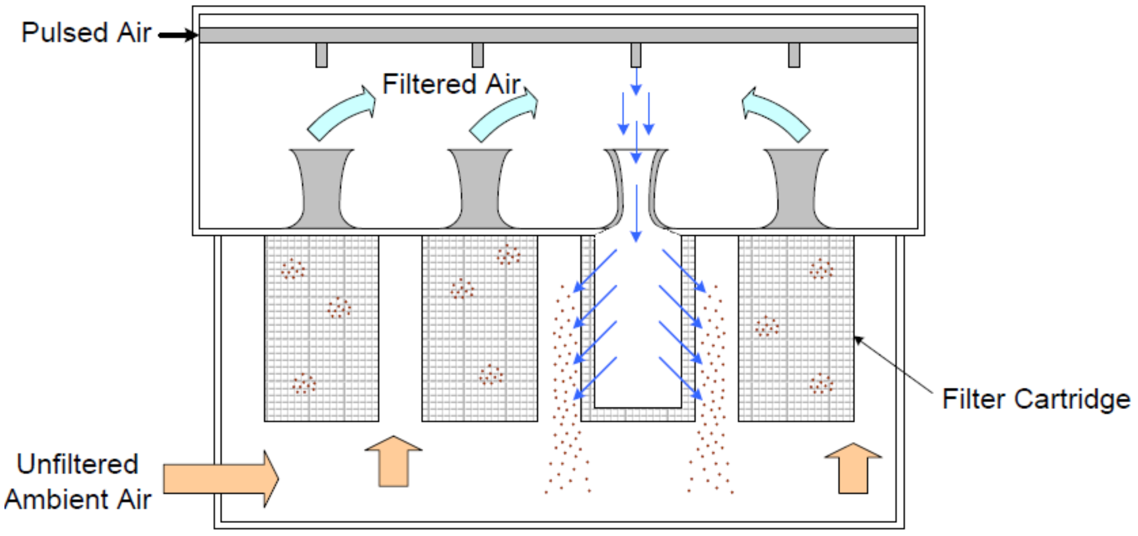

Figure 11: Operation of Self-Cleaning Filtration System [4]

2.4 Gas Turbine Compressor Cleaning

Compressor cleaning is an effective way of recovery gas turbine performance loss as a result of fouling. Several methods of achieving compressor cleaning are enumerated below with their pros and cons:

Manual Compressor washing: This involves shutdown of engine, strip the engine to have access to the compressor blade and clean fouled engine with brushed and other cleaning materials. This method has been proven to have significant performance recovery but very task with length downtime.

Grit Blasting Method: this method evolved to eliminate the intense labour involved in manual hand cleaning. It involves the use of abrasive material like rice, charcoal, nutshell or synthetic resin to remove deposits on the blade. The abrasive material is introduced into the engine during operations because high air speed is needed to achieve desire impact on the blade. Satisfactory results was achieved even with its limitation which includes inability to remove oily deposit, tendency for sealing system contamination, poor cleaning of last stages of compressor, blockage of blade cooling openings and damage of blade coating on state of the art engines[6].

Online Compressor washing: This is another method of recovery gas turbine performance degradation. It involves injection of washing fluid which is demineralized water and detergents while engine still in full operation. This method is recommended before any significant fouling is noticed on the engine and it is used as complementary to offline washing because it unable to recover full power after washes. The effectiveness of online washing as compared with offline washing is shown in the figure 12.

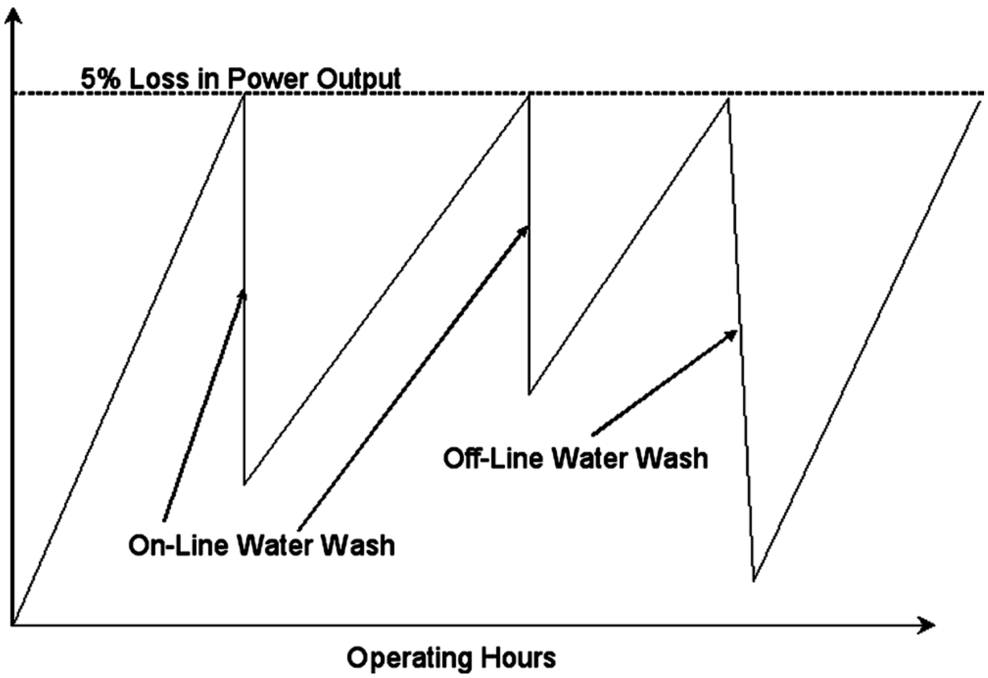

Figure 12: Effectiveness of Compressor Washing on Power Output [7]

Offline compressor washing: This method is very effective in performance recovery and it effect is far more than the result of using abrasive cleaning [6]. It is as well referred to as Soak or Crank washing; it involves shutting down the engine and allows it to cool before injection of demineralized water and detergent. The engine is only rotated for about 20% to 30% of full operating speed by starter motor [6].Offline water wash is done after all the air extraction point has been isolated to prevent effluent from air sealing piping system. Advantages and disadvantages of different methods of washing are enumerated in the table 3.

Table 3: Summary of Advantages and Disadvantages of various washing methods [6]

Methods	Advantages	Disadvantages
Manual cleaning(brushes and washing agents)	Very effective	Shutdown of engine, Very Laborious
Grit Blasting(Rice, Nutshell, Charcoal and synthetic resin particles)	Simple and fast, No engine downtime, effective in cold environment	Less effective at rear stage and for oily deposit, Clogging of internal cooling passage, Erosion, Increase surface roughness, Damage of blade coating
Fired or online washing(Demineralized water and detergent)	No interference with load profile, Extends interval of crank wash	Less effective, cannot replace offline washing
Unfired, cranked, soaked or offline washing	Very effective	Shutdown of engine, incurred more down time

2.5 Gas turbine Compressor Washing Procedure

GT washing package are either built together with the new gas turbine package or as retrofit to the existing turbines. Compressor washing recovers turbine performance loss which manifests in form of reduction in power output or increase in heat rate. Washing module is equipped with heating element, piping connections on the base, insulated tank for water, motor operated valves for water injection, and spray manifold for online and offline washing. Compressor water wash schematic of a GE gas turbine wash module is shown Figure 13. Drains are provided for waste from turbine inlet plenum, combustion area, exhaust frame and exhaust plenum.

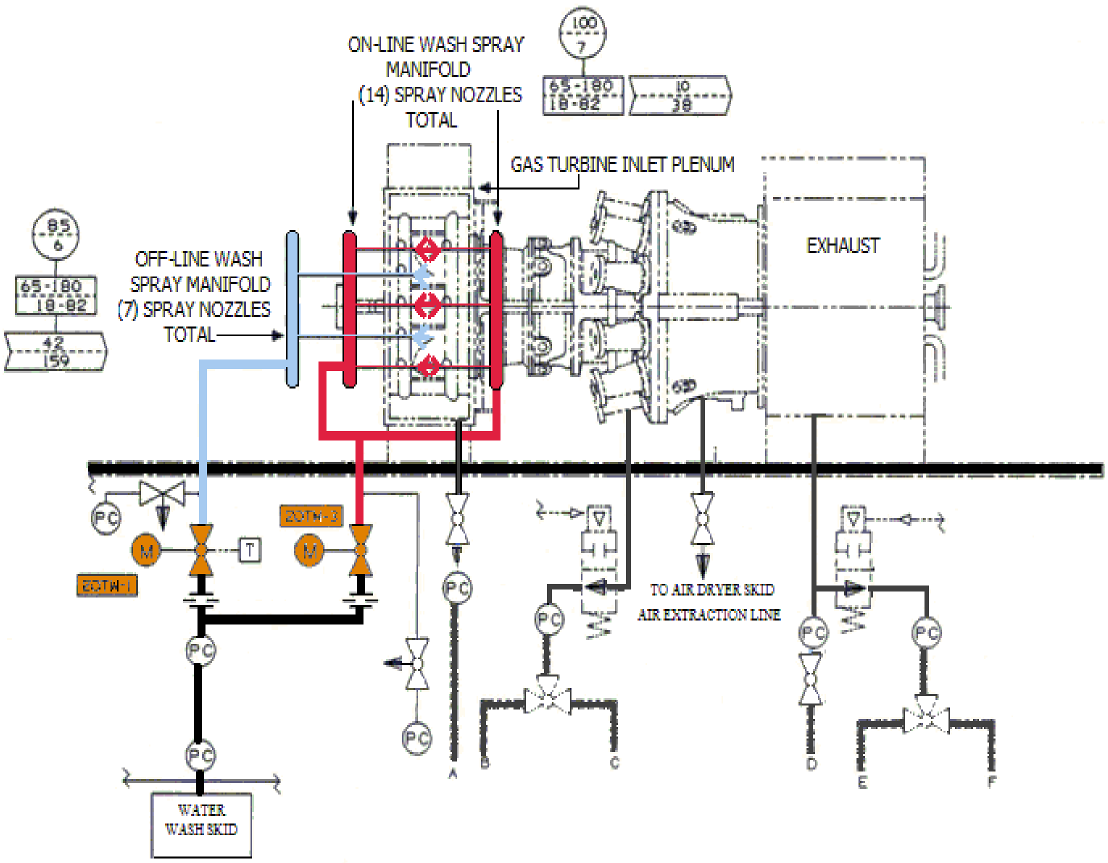

Figure 13: Compressor Water Wash Schematic [8]

2.5.1 Offline Water Wash Procedure

Figure 14 below shows GE compressor water wash path during offline washing process. The procedure water wash involves:

Ensure wash solution which in demineralized water and detergent meet specifications as instructed by the OEM

Shutdown gas turbine and allow to cooling down. However, temperature difference between wash water and inter-stage compressor must not be more than 67^{o}c to avoid thermal shock.

Select water wash on the turbine unit control panel (UCP); ensure the air extraction is isolated on the engine.

Initiate offline wash on the UCP and the control system will follow through the washing sequence ranging from injection of water wash solution, soak, spinning and draining of the wash fluid after completion of the entire activity.

Open the isolated air extraction line and initiate start to dry the compressor of liquid within the system

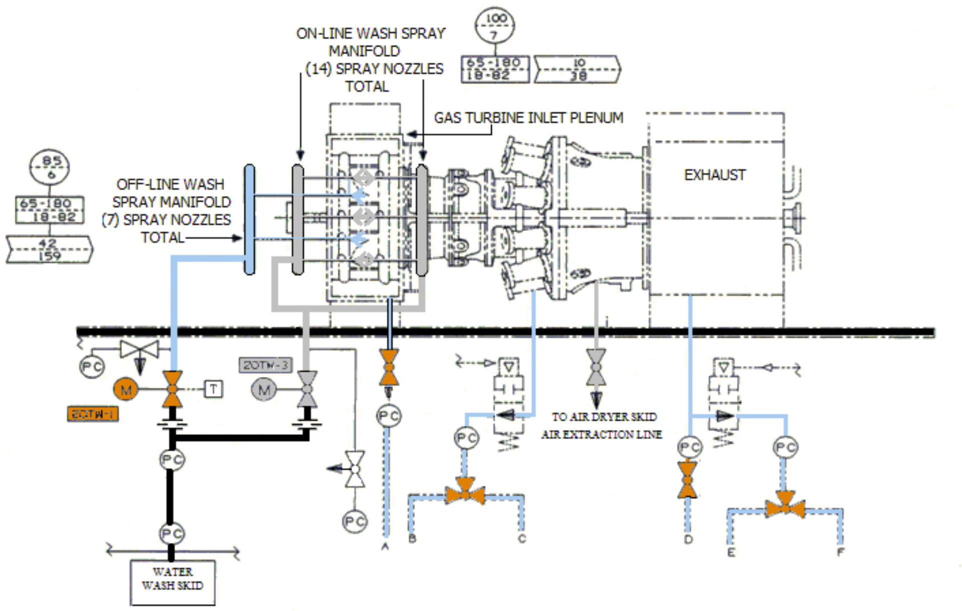

Figure 14: Compressor Offline Wash Path [8]

2.5.2 Gas Turbine Compressor Washing Fluid

Axial compressors are washed with three main type of cleaning chemicals:

- Water and Kerosene are oldest established washing fluid with history of effective washing with or without emulsifier. It is cheap and readily available though, it is losing acceptance mainly because the emulsion could split into water and kerosene during hot wash which could dangerous to engine particularly aircraft engine.
- Solvent based cleaner contains about 50-70% white spirit (solvent) while water and detergent makes other proportion. The detergent is present to emulsify mixture of solvent and water. However, the trend is shifting from the use of solvent based fluid due to it health safety (offensive smell and hazardous) even though it being accepted as a good cleaning chemical. It being found to very corrosive, it attacks rubber, seal, metal and even paints.
- Aqueous based cleaners (ZOK27, ZOK mx) this is the latest cleaning fluid in compressor washing, it is solution of detergent in water, it contains corrosion inhibitor, and safe to transport and handle. It is usually mixed in ratio of 1-4 with demineralised water [9].

2.5.3 Improvement in Compressor Online Washing Technology

There has been record of axial compressor vibration and disruption of power output while carrying out on-line washing. These are among reasons why operators hesitate to allow on-line compressor washing but with improvement in gas turbine compressor cleaning technology, the challenges are being addressed:

- Washing fluid positioning, the injection nozzle is positioned without protrusion into the airflow stream and that the spray injection is done radially into the air flow.
- Stratification of wash fluid and airflow, this is saturation of layer of air by the duct wall where nozzles are positioned with wash fluid, the wash fluid is introduced into the entire body of air flow as it moves through inlet nozzle with little or no interference with air stream. There is smooth change in direction of flow from duct through the plenum and into the

compressor (Fig 15).The benefit is that the fluid moves further away downstream of plenum and has better coverage of the blade.

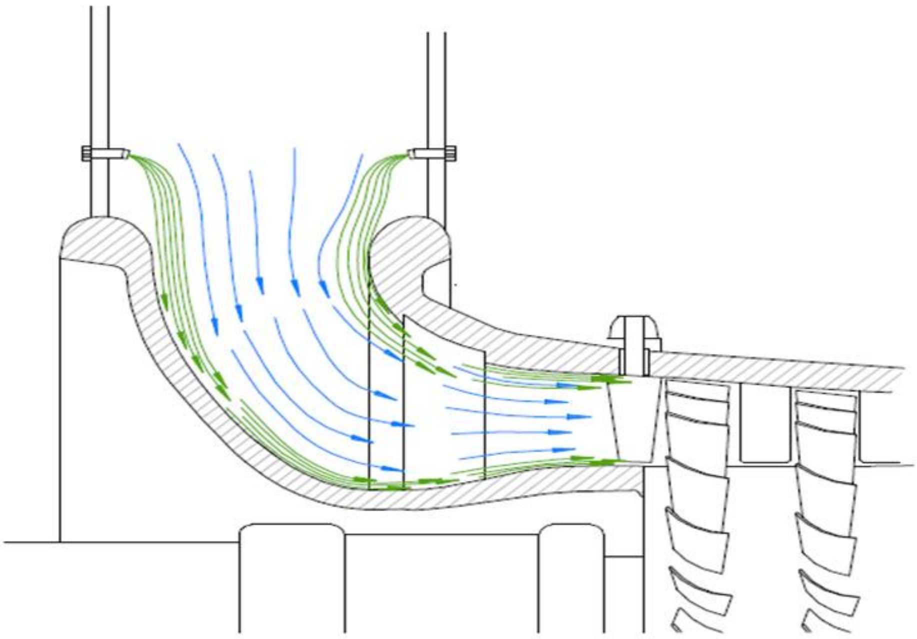

Figure 15: Stratification of Airflow Fluid with Wash Fluid [10]

Burying of wash fluid nozzle in the inlet plenum, this would provide less intrusion of airflow by nozzle (Fig16) but might prevent mixing of wash fluid with air flow. Pressurised wash fluid (45-90bar) is injected to achieve better penetration of wash fluid ahead of air flow with no disturbance until they reach compressor.

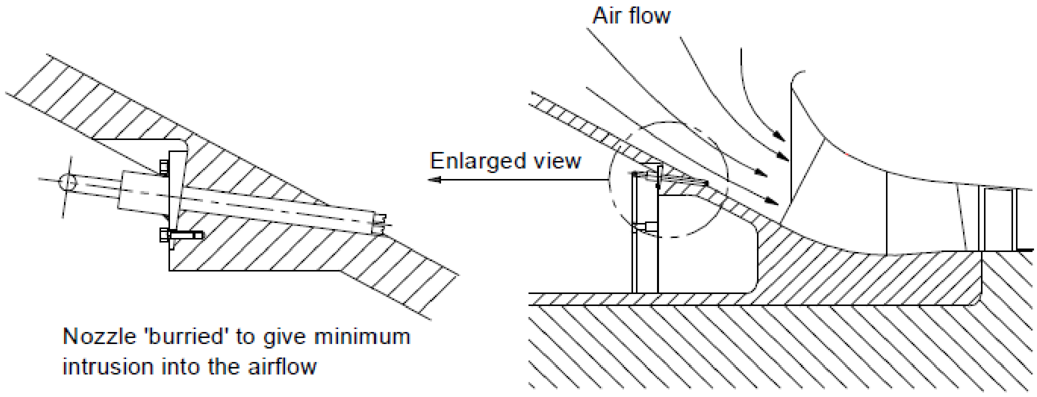

Figure 16: Position of Wash Fluid Nozzle on Inlet Plenum

2.5.4 Advancement in Compressor Blade Technology

The compressor cleaning technology is advancing with the improvement in compressor blade technology. The blades are finely developed with high aspect ratio (chord to tip ratio), thinner leading edge and smaller clearances to achieve high pressure ratio.

As the quality of blade aerofoil and surface improved, deposition on the blade is more detrimental to the compressor performance. Vortices are formed due to deposit on the leading edge and even on the middle end of the blade thereby causing streamline to break away from blade surface (Figure 17).

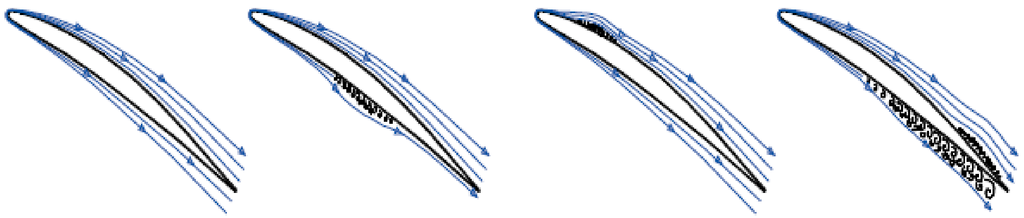

Figure 17: Streamline Disruption due to Deposit

Injection of usual 4-9 bar wash fluid have little or no effect on the deposit, it would follows the streamline without any impact on the deposit.

However, development of small opening nozzle with high injection pressure of up 90bar, heating of washing fluid and addition of surfactant reduces the fouling surface tension and effective cleaning achieved [10].

2.5.5 Washing Fluid Droplet Behaviour

The dilemma of achieving large enough droplet size of wash fluid that would remove deposit from blade surface without eroding the blade surface coating is being a challenge. Large wash fluid droplet would be centrifuge out of the air stream without impact while small droplet that would cause no erosion but would follow through the stream without effect on blade boundary layer deposit (Fig18).

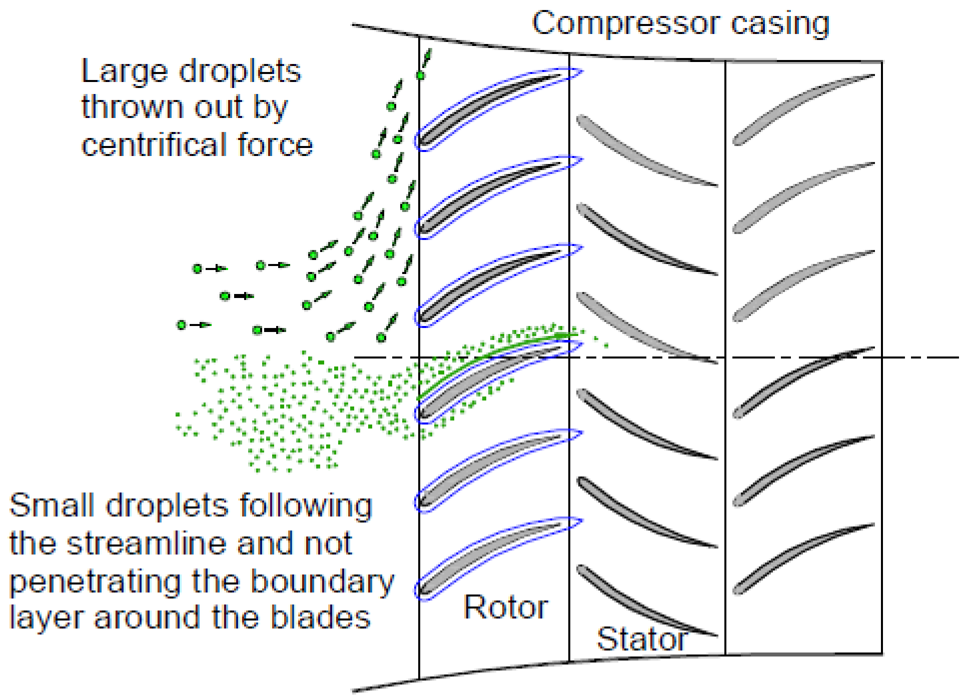

Figure 18: Wash Fluid droplet behaviour across Blade

However, with the increase in wash fluid injection pressure, heating and addition of surfactant the small droplet are reduced and the large droplet are eliminated by reducing the total spread of droplet size.

2.5.6 Benefit of High Temperature Washing Fluid

Compressor wash fluid get vaporised by air temperature at fifth to sixth stage, and thus, the dirt removed are deposited on the latter stages. However, with the evolution of high temperature carrier in washing fluid, the fluid boiling temperature is raised to the compressor exit temperature and pressure. This ensures complete cleaning of the compressor blade with reduced injection pressure to vary the droplet size, the adhesion of boundary layer is achieved

2.6 Plant Reliability Indices

Most of the force outage recorded is due to turbine auxiliary control system which includes fuel gas system, Speed control system, combustion temperature sensing probes, Bentley Nevada vibration monitoring system and air inlet system.

Figure 19 indicates engine system ranking of forced outage rate and force outage total down time, engine control system often contributes higher number of force outage thereby reduce turbine reliability but with holding right spare, right technical personnel and right maintenance approach, the down time might be reduced. The main hot path gas turbine component seldom fail which suggest high reliability but once they fail it may take longer time to repair because spare availability, technical know-how and high repair cost. Apart from sophisticate monitoring instrumentation installed, the operator has also taken a proactive measure to purchase capital spare for storage to reduce downtime once it occurred.

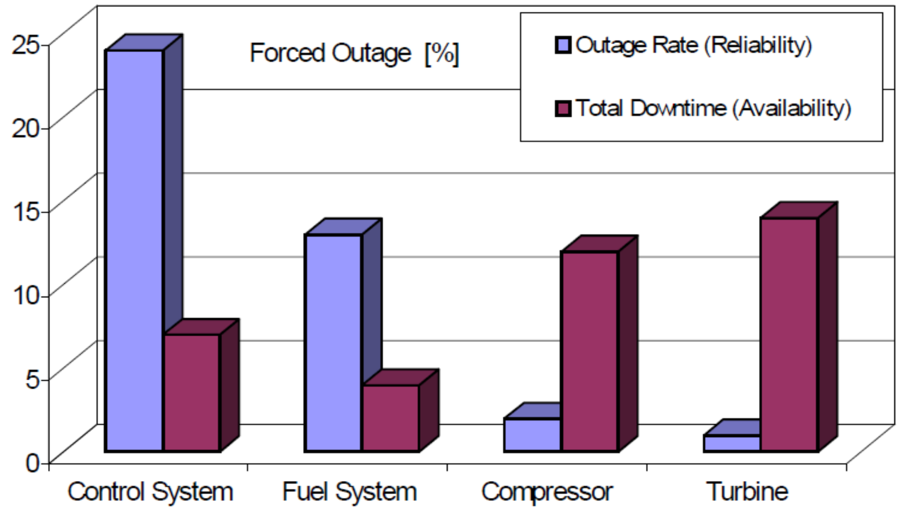

Figure 19: Ranking of Force Outage and Force Outage Total down time [11]

Among other factors that could offer solution to the plant unavailability is improvement in the technical competency of the maintenance team, proactive ordering of spares and regular communication with OEM on equipment performance.

Plant availability is the percentage of time the plant is available to generate power in any given period at its acceptance load.

Reliability is the probability that the plant will perform it desire function within specify time without failure. Availability is a function of reliability

2.7 Summary

Gas turbine performance deteriorates gradually in operation due to accumulation of foreign particles contained in air. The gas turbine operating a tropical environment has shown greater collection of dirt particles in form of dust which are evident on clogged filter systems.

Experience has shown that high performance filtration system only reduces dirt admittance into the turbine air inlet and enclosure. However, Boroscopic inspection of turbine axial compressor has reveal that even high efficient performance filter does not stop dirt particle in the air from being admitted into gas turbine gas generator.

Advancement in technology has improved the gas turbine performance recovery through regular cleaning of turbine axial compressor with the introduction of cleaning fluid (detergent) into the gas turbine for washing off dirt accumulated and thereby sustaining turbine efficient performance. This washing is either done when turbine is at rest on even while running and on load.

3 Gas Turbine Performance Simulation

Introduction

The study begins with investigation of gas turbine performance with the use of TURBOMACH SOFTWARE to demonstrate thorough understanding of inter-relationship between major components of turbine. TS3000 engine, a single shaft gas generator engine that is aerodynamically coupled with power turbine is designed using Turbo-match code. The performance of its key components are studied both at deign and off design point.

3.1 TS 3000 Inspired on LM2500+ Gas Turbine

Aero-derivative gas turbines are aircraft engines converted to industrial engine due to rise in demand for a highly reliable, small size (compactness), high power output, efficient, easy maintainability and environmental friendly prime mover in the Industries. The increasing demand for oil and gas which are the main energy sources has led engineers to develop and adapt aircraft engines for industrial use.

Aero-derivative engines benefited from the long time and huge budget allotted for aircraft research and development. This earns aero-derivative engine some operational and economic advantages, and of course is attractive to end users.

Aero-derivative LM 2500 has over 31million operation running hours which shows its excellent reliability and performance. LM 2500 series belong to the F39/CF6-6 family.[8]

LM2500+ gas turbine conception was based on improvement of successful design history and performance of LM2500. The current LM2500+ is rated at thermal efficiency and power output of 41% and 31.3MW which was advancement from its original 37.5% and 27.6MW thermal efficiency and power output respectively. Figure 3-1 shows isometric view LM2500+.

3.1.1 Engine Technical Specification

LM2500+ gas generator operates at 22:1 compressor pressure ratio with 2 stage High speed power turbine design at 6100rpm, operating range of 3050 and 6400 rpm for mechanical drive application which can operate over a cubic load curve. It also has another design with 6 stage low speed power turbine with design speed of 3600 rpm for power generation application as shown in Figure20.

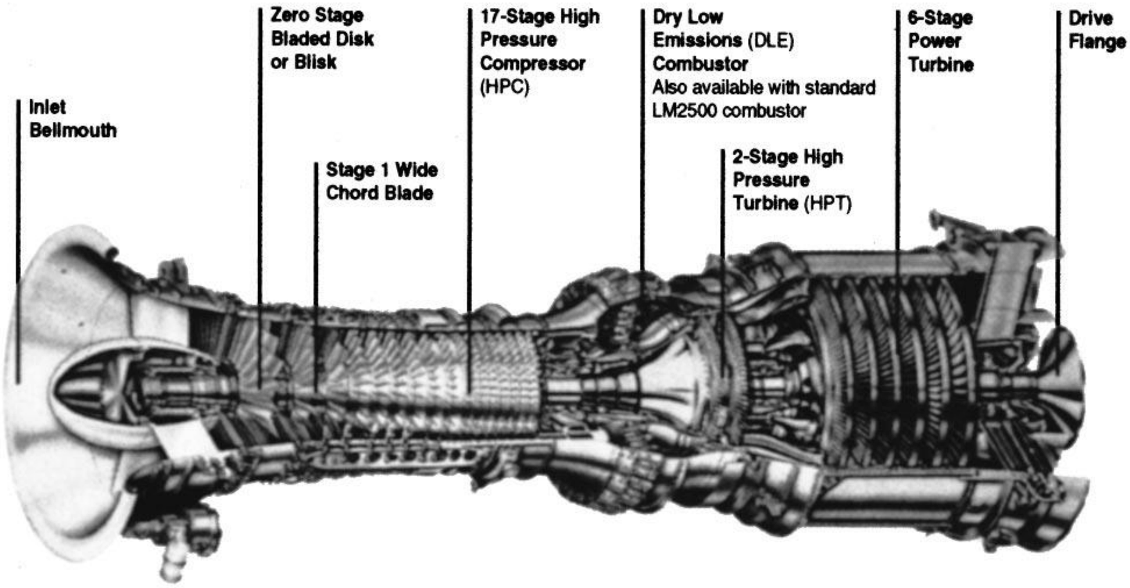

Figure 20: LM 2500+ Major Components [8]

LM2500 gas turbine is a single rotor gas turbine with an aerodynamically coupled power turbine. It has seventeen stage axial flow design compressor with an added zero stage ,single annular combustor(SAC) of 30 individually replaceable fuel nozzles and two stage high pressure turbine [8].The inlet guide vane (IGV) and the first sixth stage of stator blades are variable with air cooled blades ,pressure turbine and nozzles. Table 4 shows the performance characteristic of LM2500+ mechanical drive application.

Table 4: LM2500+ Data Specification

Power output (kW)	Heat rate (J/kWh)	Exhaust Flow (kg/s)	Exhaust temperature (^{o}C)
31320	8756	84.3	499

3.2 Design point performance simulation of TS3000

Design point is defined as a specific point within the operating range of an engine when the gas turbine is running at specific speed, pressure ratio and mass flow for which the components were designed for. The design point of this engine is calculated based on data gotten from open domain with standard atmospheric parameters which are listed below.

1. Atmospheric Temperature : 288.15 k
2. Atmospheric pressure 101.325 kPa
3. Altitude ; 0
4. The result gotten from TS3000 design point is displayed in Table 5

Table 5: Design point result

ENGINE PARAMETER	VALUE
Power output(MW)	31.3
Fuel flow (Kg/s)	1.7918
Specific fuel consumption (mg/kW.s)	57.19
Exhaust Temperature(K)	788.84
Thermal efficiency	40.6%

Table 6 shows comparison between the engine public domain value and the simulated design point value to validate the accuracy of turbomatch software.

Table 6: Public domain engine parameter compared with design point simulated result

ENGINE PARAMETER(EP)	ENGINE PUBLIC DOMAIN VALUE(RE)	SIMULATED ENGINE VALUE(SE)

Power output (MW)	31.3	31.3
Exhaust mass flow (Kg/s)	84.3	84.3
Exhaust Temp (K)	787.15	788.84
Thermal Efficiency (%)	41	40.6
Pressure ratio	22:1	21.9:1

Table 6 summarises the accuracy of design point modelled engine simulation carried out and OEM engine specification. The outcome shows 100% accuracy in both power output and exhausts mass flow as specified by OEM while there is slight decrease in engine thermal efficiency and pressure ratio. The exhaust temperature also shows some increment which is an indication that more energy is being loss to the atmosphere. The simulated design point value compared favourably with engine parameter from public domain as displayed in the table 6 which confirm the viability of the software

ENGINE PARAMETER	VALUE
Power output(MW)	31.3
Fuel flow (Kg/s)	1.7918
Specific fuel consumption (mg/kW.s)	57.19
Exhaust Temperature(K)	788.84
Thermal efficiency	40.6%

3.3 Design Point Compressor Performance

Figures 21 and 22 represent the TS3000 axial compressor performance characteristic with Pressure ratio, isentropic efficiency plotted against non-dimensional mass flow. At each constant speed line, the compressor pressure ratio increases with mass flow which is the expected compressor behaviour. However when the compressor maximum efficiency is attained further increase rotational speed will not achieve corresponding rise in pressure ratio because the air molecule will separate from the compressor blade surface. This phenomenon is referred to as choke condition.

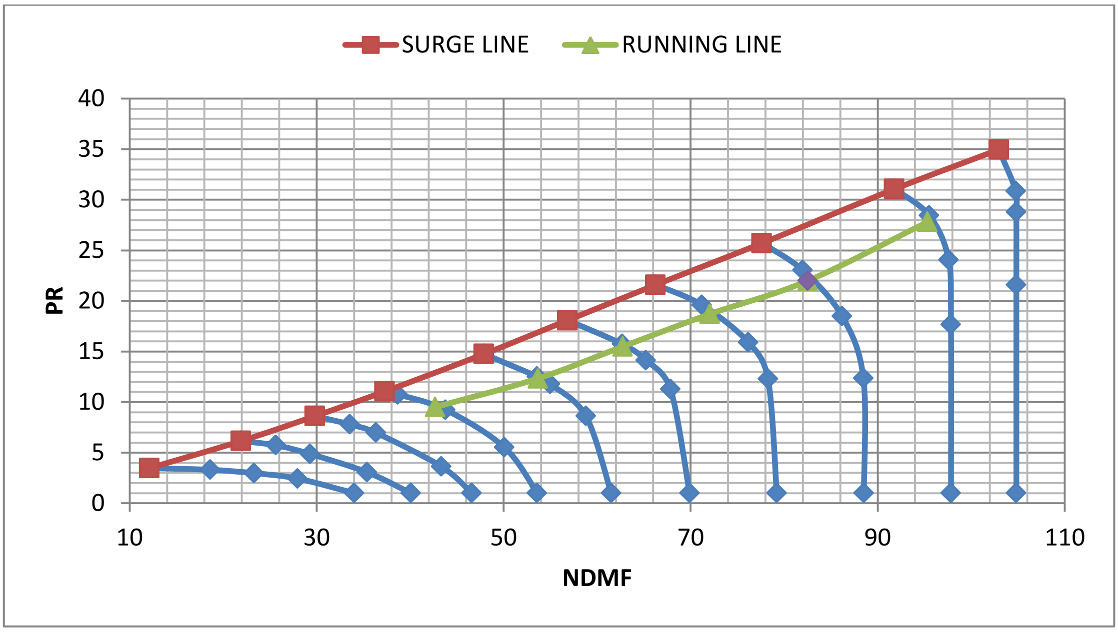

Figure 21:TS3000 DP Compressor Performance Curve

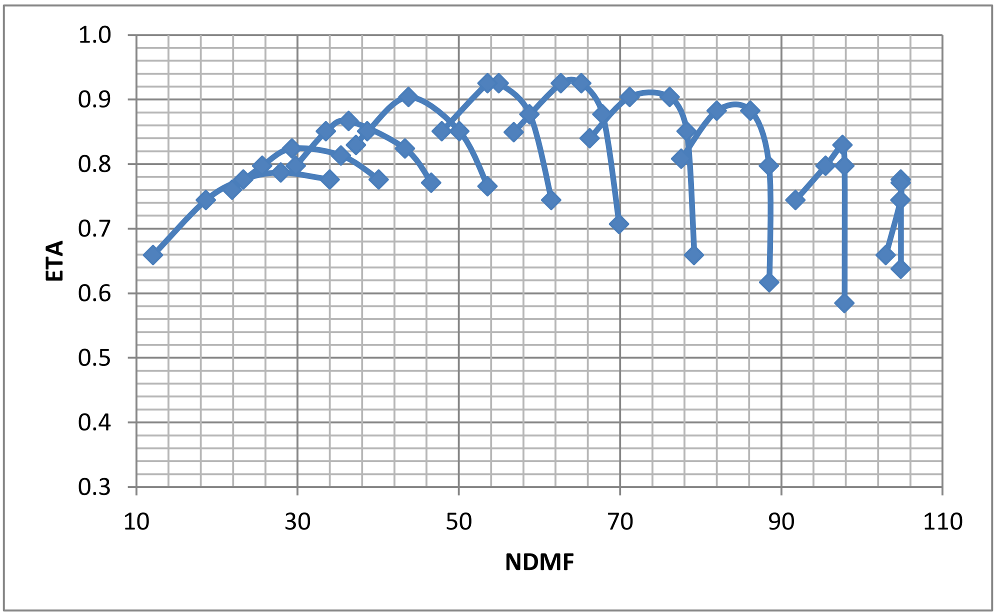

Figure 22: TS3000 DP Compressor Map

3.4 Off Design Performance Simulation

Off design performance simulation is important to establish engine overall performance characteristic over a wide range of operating condition for safety and operational economic benefit. Design engine is simulated at varying degree of ambient conditions, altitude and part load.

Figures 23 to 25 represent the performance of the engine when simulated under various ambient condition (-5 to 40) ^{o}C as attached in the appendix A 1. The performance shows optimal power output and increase pressure ratio when operating in cold atmosphere because more dense air molecule is able to consumed and compressed by the axial compressor and thus give rise to maximum power output. The power output begin to decline when ambient temperature increases which is the expected turbine performance. Furthermore, fuel flow increases at cold ambient conditions which enhance maximum power delivery and associated with rise in specific fuel consumption. Detailed simulation data is attached in Appendix A2

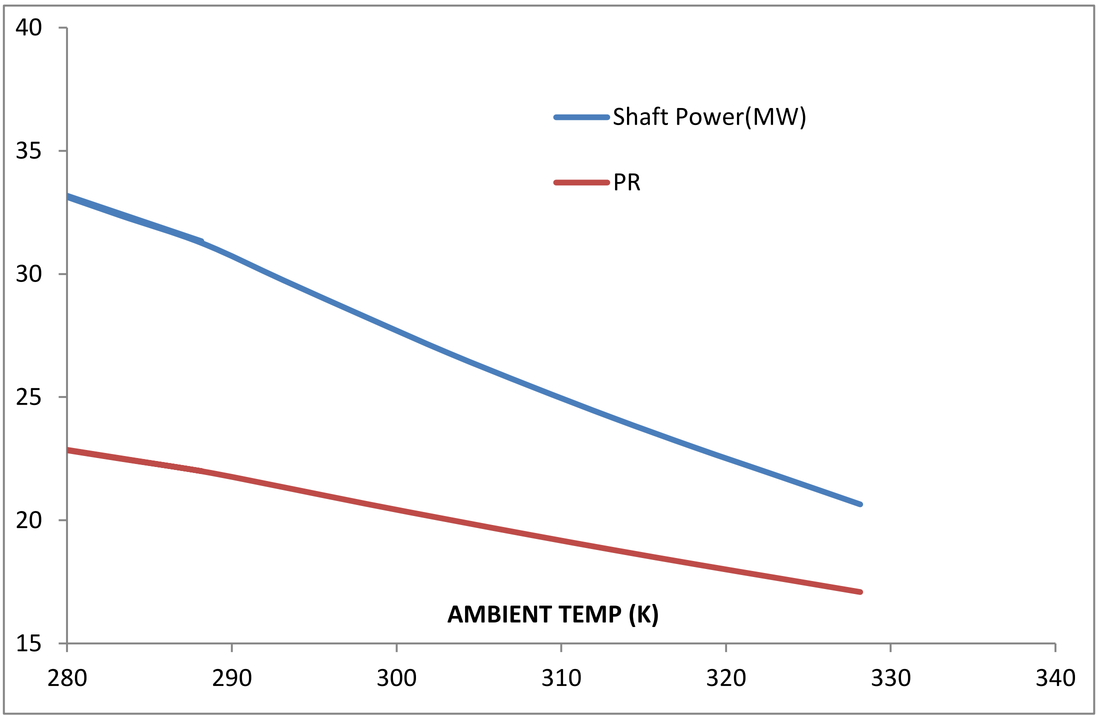

Figure 23: Variation of Shaft Power output, Pressure Ratio with Ambient Temperature

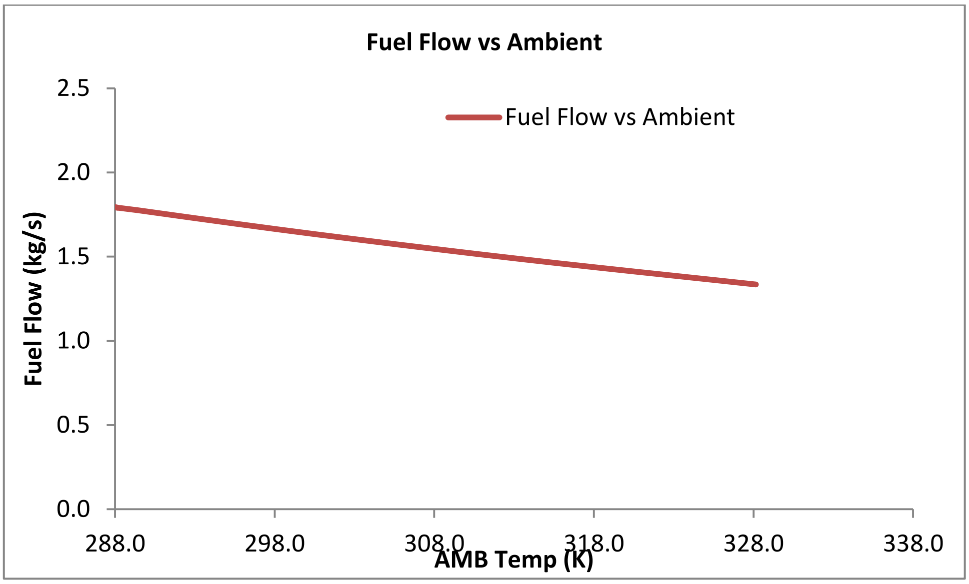

Figure 24 : Fuel Flow (kg/s) against Ambient Temp (k)

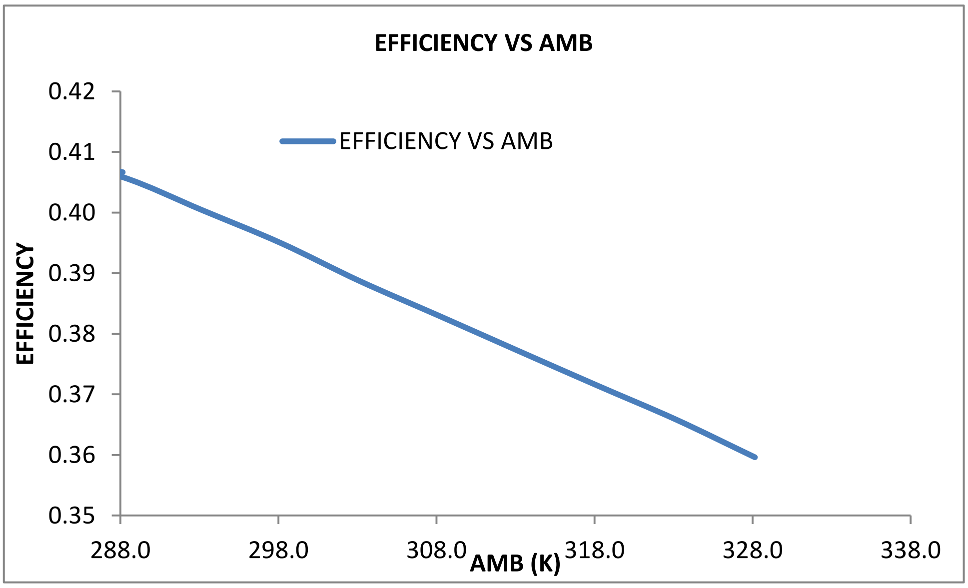

Figure 25: Variation of Thermal Efficiency with Fuel Flow

(k)

3.5 Engine Performance at varying TET

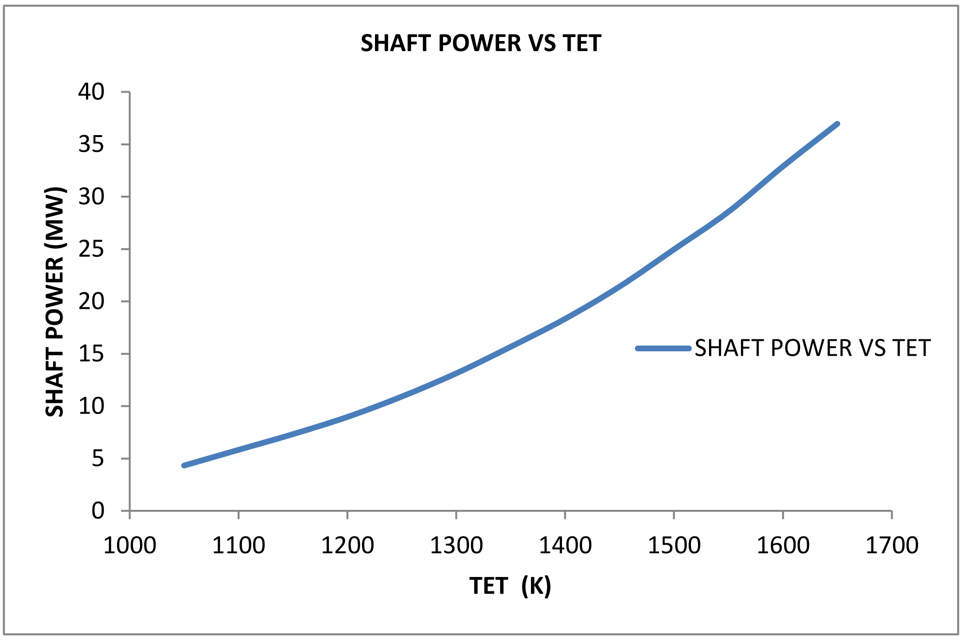

Figure 26: Shaft Power (MW) Turbine Entry Temperature TET (K)

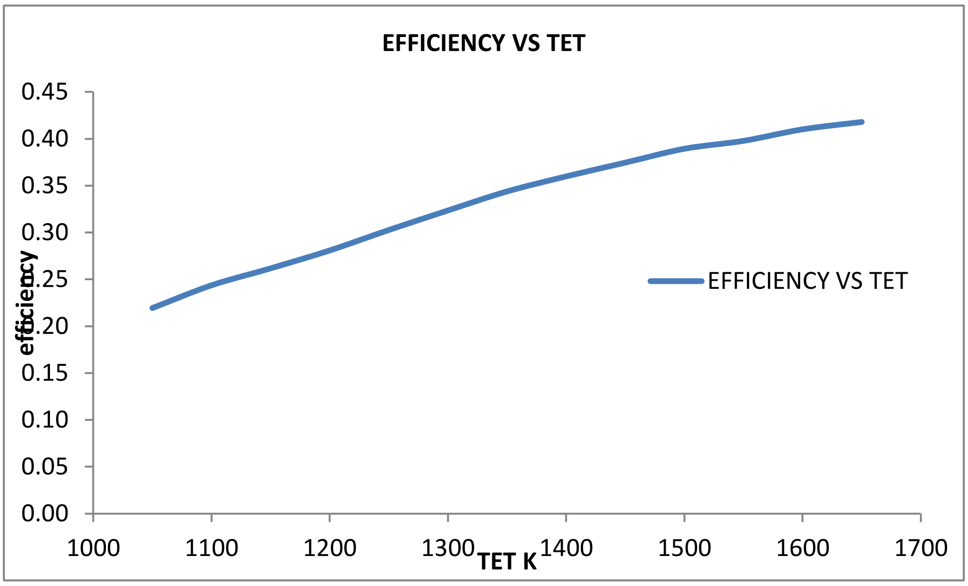

Figure 27: Efficiency versus Turbine Entry Temperature TET (K)

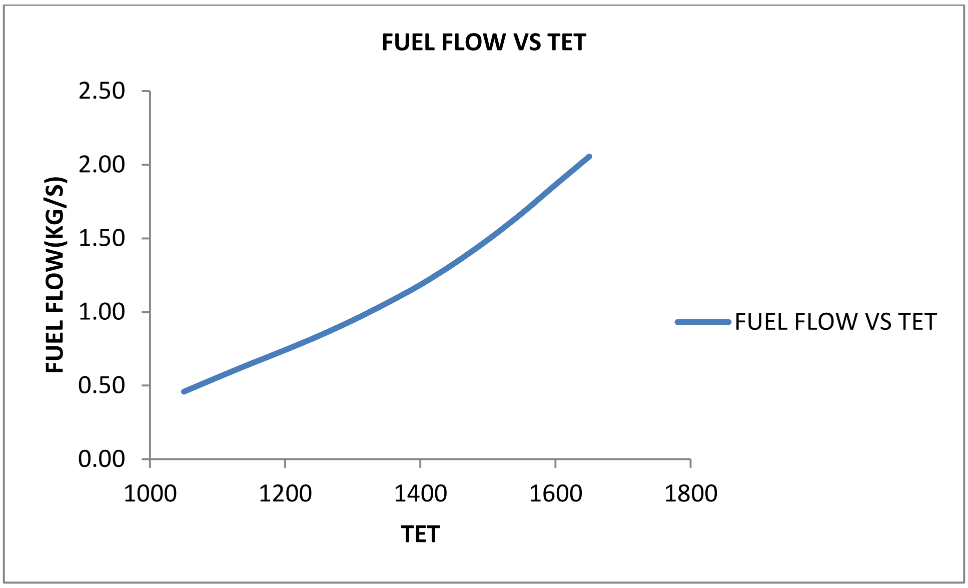

Figure 28: Fuel Flow (kg/s) against Turbine Entry Temperature TET (K)

Figure 26 to 28 depict the performance of gas turbine engine with influence of changes in turbine entry temperature (TET).There is huge drop in shaft power at low TET accompanied with large fuel consumption which an indication of engine poor performance at TET (1000k) but as TET increases there is indication of sudden drop in specific fuel consumption and appreciable increase in shaft power output with increased fuel flow. Appendix A3

3.6 Axial Compressor Fouling simulation

In order to demonstrate good understanding of gas turbine performance and to aid actual field data analysis, turbo-match software was used to model the engine to be investigated (H25). The performance at different off design condition was simulated and various interaction between various components of the engine was studied. This provided better insight into engine characteristics at different condition.

The effectiveness of online compressor wash on overall gas turbine performance was simulated with used of TURBOMATCH software. An industrial single shaft engine (Fig29) called B54 with the under listed specifications(Table 6) was retrieved from public domain and investigated at clean, fouled and washed condition

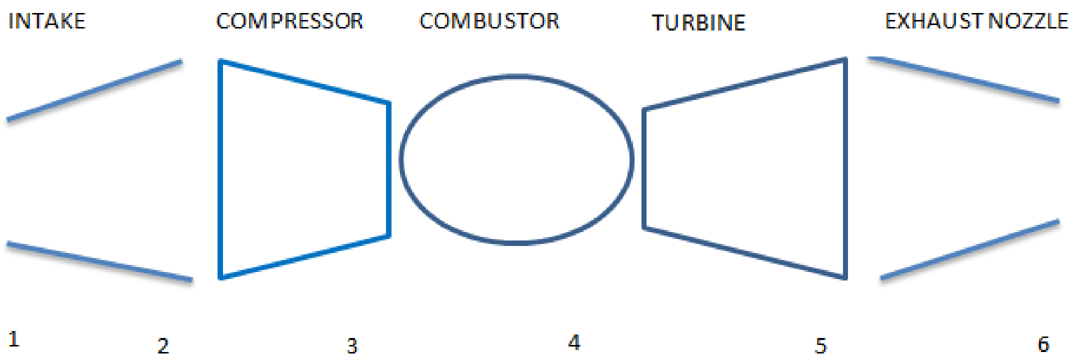

Figure 29: Shows Single Shaft simple Gas Turbine

Table 7: B54 Engine Model Design Point Objective

Design Parameters	Value	Simulation DP results
Mass Flow	88.0kg/s	88.0Kg/s
Pressure ratio	14.7	14.7
Turbine Inlet Temperature	1548K	1548
Exhaust temp	843K	840K
Thermal Efficiency	33.8%	33.7%
Compressor isentropic efficiency		0.8%

The engine is modelled at design point and run with turbo-match software; the result is very valid when compared with the specification from open domain (Table7) with minor disparity from exhaust temperature. This creates a good platform to simulate and investigates the engine at off design performance condition.

Fouled degradation of about 5.0% reduction in compressor efficiency and 2.0% reduction in compressor inlet mass flow was adopted based on the outcome of an experiment which suggests 5.3% reduction in compressor efficiency equivalent to about 12months of operations without wash.[12] An online compressor wash was conducted and high optimistic performance recovery of about 50% was implanted in the model for the study. The equation (3-1, 3-2) below summaries the degraded model condition as explained above. It is important to note here that the engine was controlled at different handles (TET and PCN) while compressor surge margin, speed, power turbine output and mass flow were allowed to vary. Detail input file attached in appendix A and B

$$\eta_{(cnew)} = (1 - \Delta\eta_c\%) \times ETASF \times \eta_{cmap} \quad \textbf{(3-1)}$$

The engine is run at ISA condition to study its characteristics, the compressor pressure ratio increases as the compressor speed increases (N 0.56 –N1.0)

$$\eta_{(cnew)} = (1 - \Delta\eta_c\%) X\ ETASF\ X\ SF\ X\ w_{map} \quad \textbf{(3-2)}$$

from about 40% up till 100% rotational speed which is maximum design point speed, more mass flow of air is being pulled through the compressor, the compressor operating line at steady state with different rotational speed and surge margin is indicated (Fig 30). The compressor characteristics map indicates that at low speed, the compressor tends to stall due to low mass flow of air and rising differential pressure at the back of compressor. This phenomenon is usually controlled by the IGV, VGV and bleed valve installed on the compressor. These control devices usual come to play during engine start up to prevent engine from stalling. However, at higher rotational speed the compressor tends to choke due higher mass flow of air and reduce pressure ratio, which compressor cannot cope with. The compressor would go into surge at higher speed beyond design speed and the compressor is protected by installation of various anti-surge control systems. The compressor performance does not further increase when the designed point efficiency has been attained (Fig 31) and in fact further increase in rotational speed cause drastic downward shift in compressor efficiency.

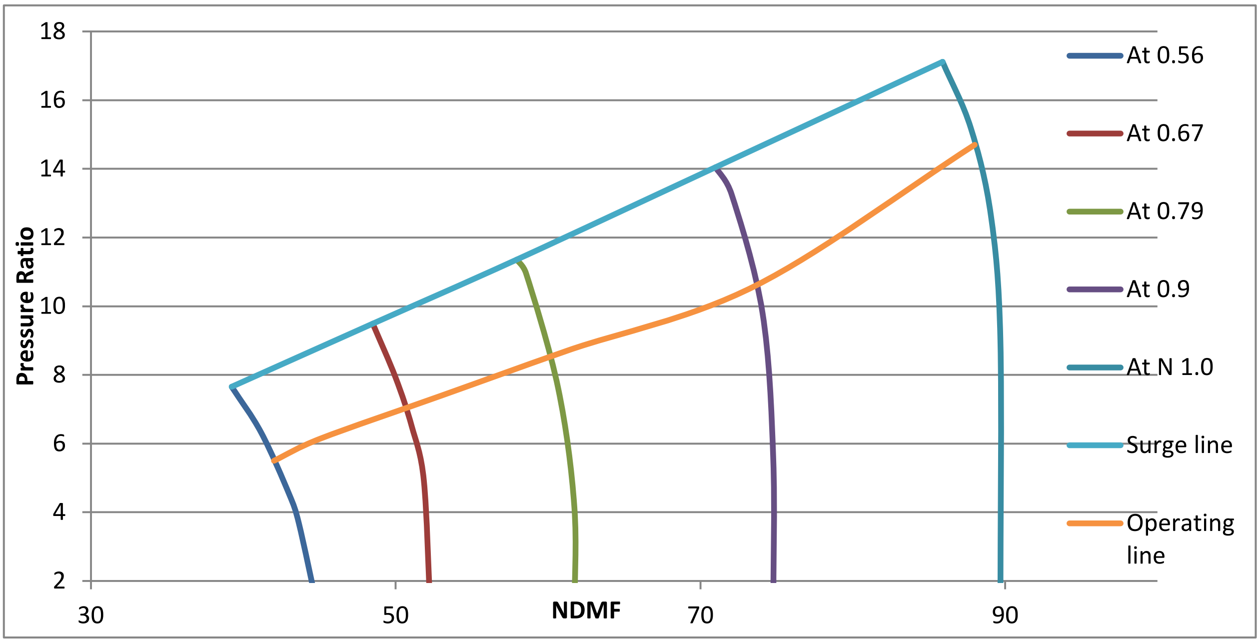

Figure 30: Axial Compressor characteristics at different Rotational Speed

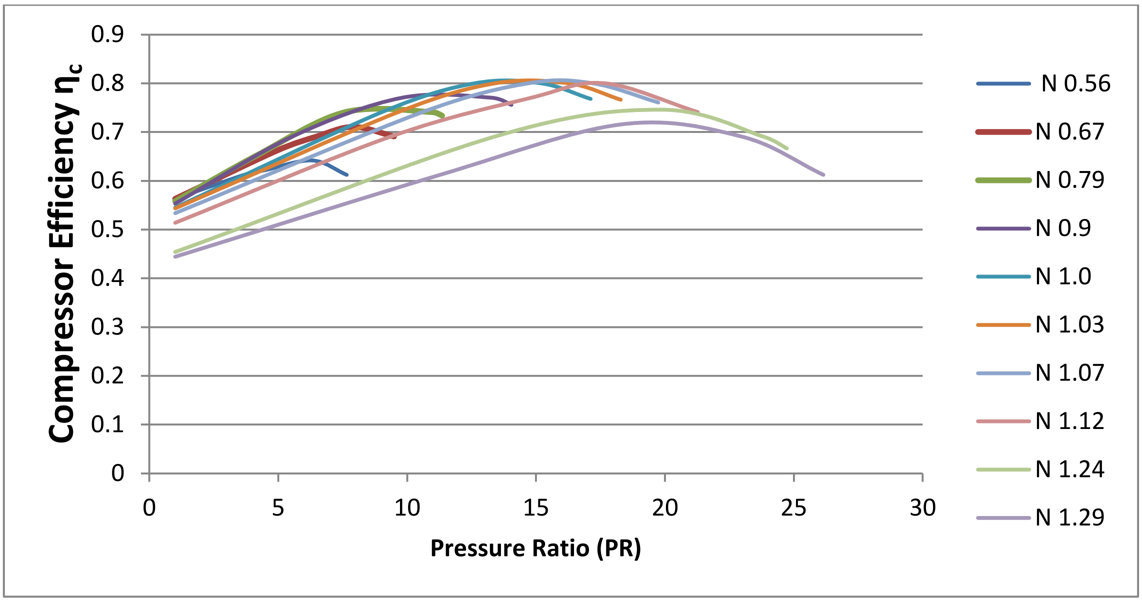

Figure 31: Axial Compressor Efficiency against Pressure Ratio

3.7 Gas Turbine Performance at Varying Turbine Entry Temperature

The performance simulation shows that with increase turbine entry temperature (1000-1500K), the engine thermal efficiency experiences rises which means that more fuel flow is admitted into the combustion chamber to meet the demand of load increase.

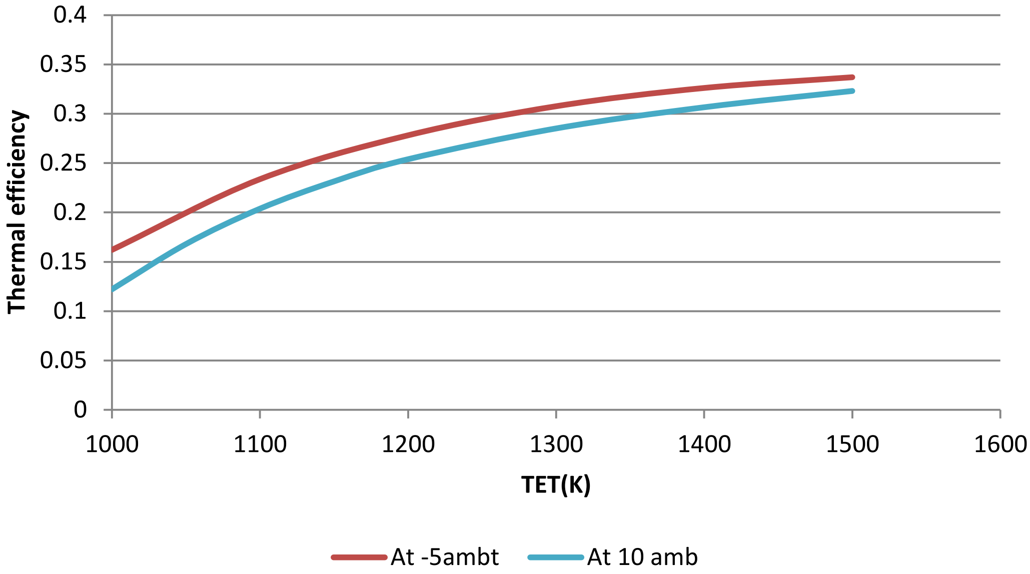

Figure 32: Shows Gas Turbine Thermal Efficiency versus Turbine Entry Temperature TET (K)

However, ambient temperature play significant role in performance of gas turbine and can be seen in the Figure (32-34). These indicate that at same power setting of TET but with colder ambient temperature of -5^{o}C, an additional gain of 4% thermal efficiency is achieved. This is due to the fact that more dense air is able to move through axial compressor with addition of proportionate fuel flow to achieve the desired power output.

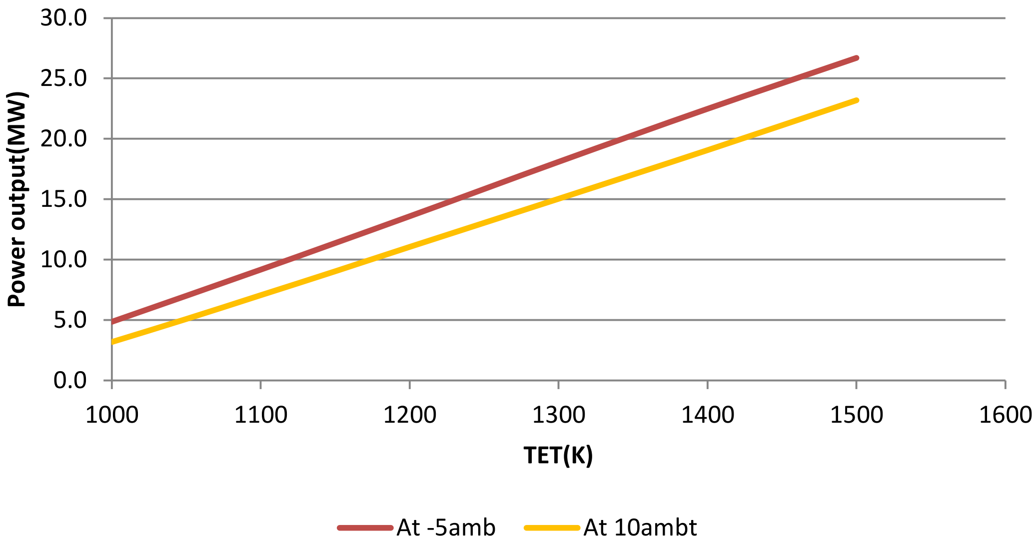

Figure 33: Shows variation of Turbine Power Output with Turbine Entry Temperature TET (K)

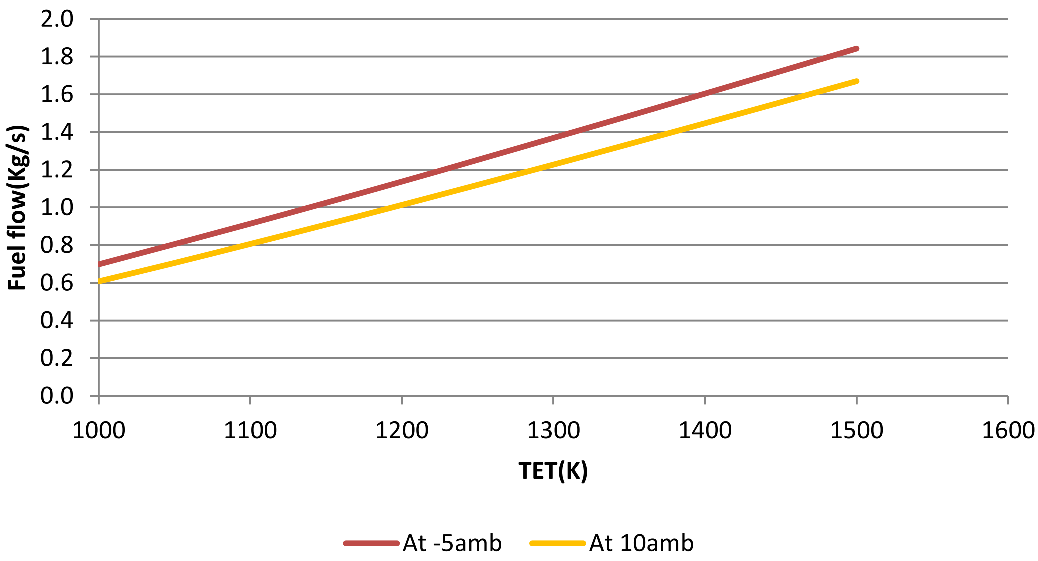

Figure 34: Shows variation of Fuel Flow (kg/s) with Turbine Entry Temperature TET (K)

3.8 Compressor Map Shifted Due to Implanted Fouling

Base on the past work that was done in investigation of impact of compressor fouling on gas turbine performance [13] compressor fouling of 5% reduction in compressor efficiency and 2% reduction in mass flow was implanted in the turbo-match engine input file (Appendix c) and an optimistic recovery of 50% in efficiency and mass flow was simulated for washing effectiveness. The compressor characteristics after simulation is presented (fig 35)

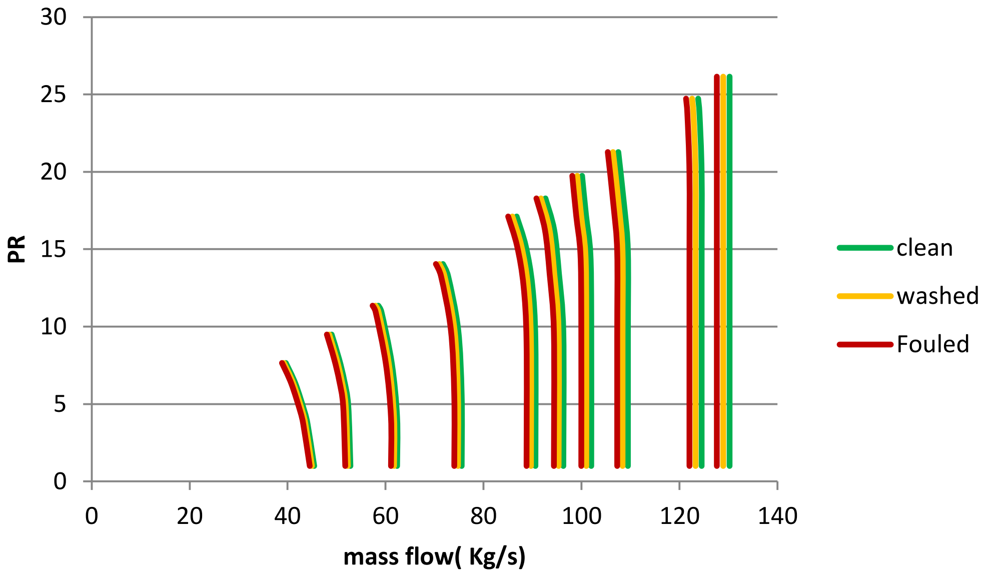

Figure 35: Effect of implanted fouling condition on compressor characteristics

Compressor map at clean, degraded and washed condition was plotted as shown in figure 35.Fouled case shown some per cent of reduction in mass flow which shifts compressor map entirely from its design characteristics map, it assumed another characteristics map and the compressor will not be able deliver its design maximum pressure ratio. Table 8 is the result of reduction in compressor pressure ratio and non-dimensional mass flow due implanted fouling and percentage recovery after simulated washed condition.

Table 8: Effect of Implanted Fouling on Compressor Performance

Compressor condition	Pressure Ratio	NDMF	Result
Clean	14.7	0.01489	0
Fouled	14.3	0.01461	↓ 2%
Washed	14.57	0.01475	↑ 0.9%

The shift in compressor map is more obvious (Fig 36) when the compressor isentropic efficiency is plotted against pressure ratio for both design point condition and fouled condition.

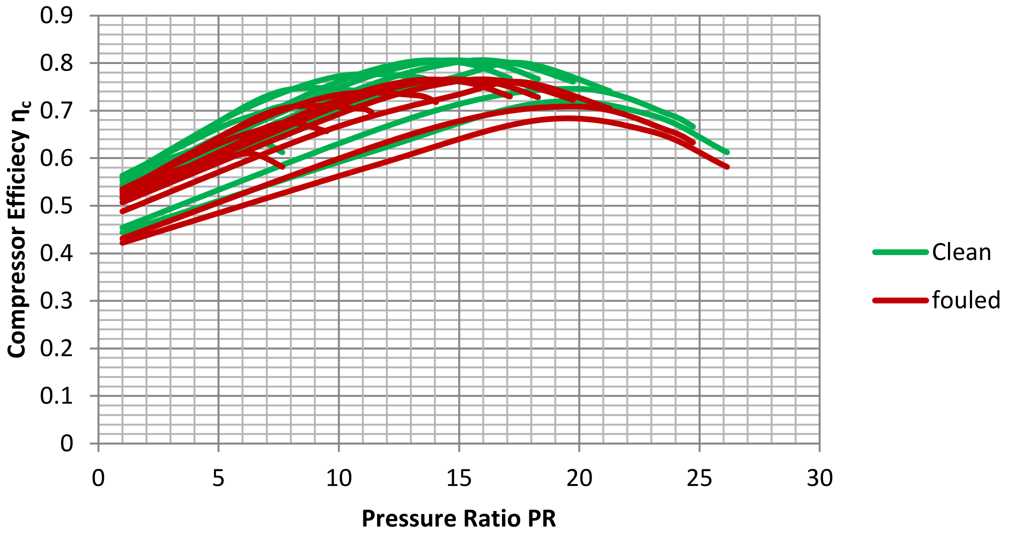

Figure 36: Shows Compressor Isentropic Efficiency versus Pressure Ratio

The impact of simulated and optimistic recovery of 50% in compressor performance is illustrated in (Fig 37)

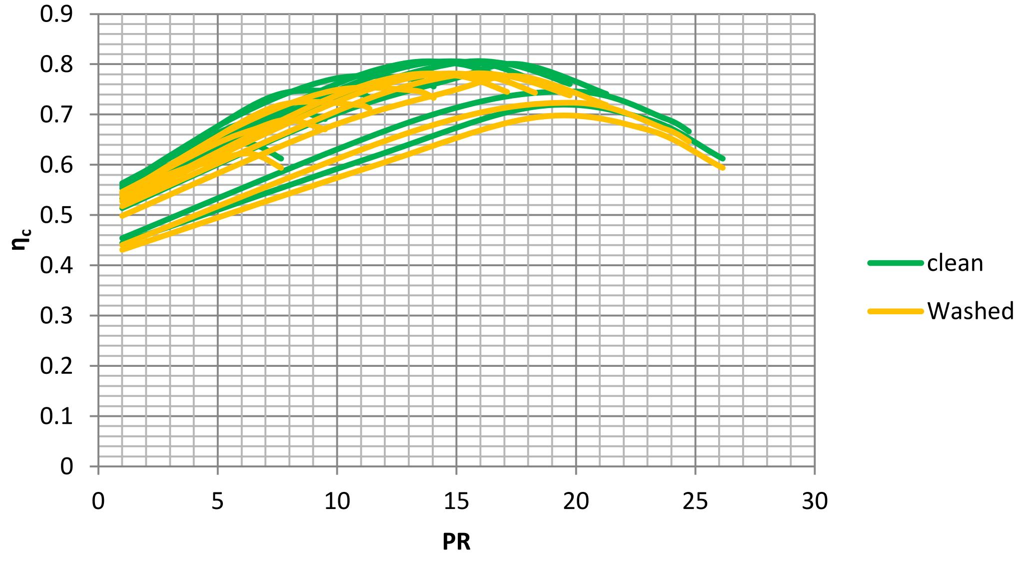

Figure 37: Shows Effect of simulated washed condition

The recovery is due to improvement in the non-dimensional mass flow of air that was able to flow through compressor as indicated in table 8.

Table 9: Displayed percentage recovery in power output

Engine cond.	Compressor Isentropic eff.	Thermal efficiency	Power output	Deviation in power output
Clean	0.80	33.8%	27.5MW	0
Fouled	0.76	32.1%	25.2	↓8.3%
Washed	0.78	32.9%	26.2	↑4.7%

The shows an improvement of 4.7% recovery in power output after washing and improvement in consumption which manifested in improvement in thermal efficiency (table9).

3.9 Gas turbine performance decreases due to rise in ambient temperature

The air mass flow across the axial compressor reduces as the ambient temperature increases, the air molecules gain more kinetic energy as the temperature increase and this makes the air less dense and thus, leads to reduction in the quantity air that flows through the compressor. The turbine performance is simulated at different ambient temperature from -5^{0}C to 40^{0}C.The reduction in mass flow of air leads to decrease in compressor pressure ratio, fuel flow and useful power output (Fig 38).

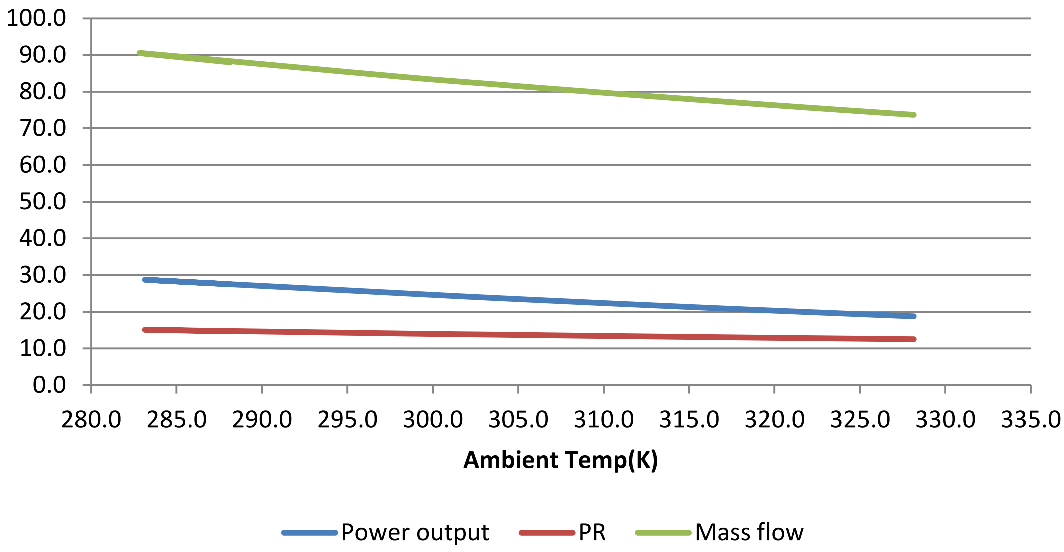

Figure 38: Shows Effect of Ambient Temperature rise on Gas Turbine Performance

The gas turbine overall thermal efficiency drops as the ambient temperature increases (Fig 39) which affect fuel flow as well because as ambient temperature increase, the fuel gas become less dense and the quantity of fuel flow decreases with increase ambient temperature.

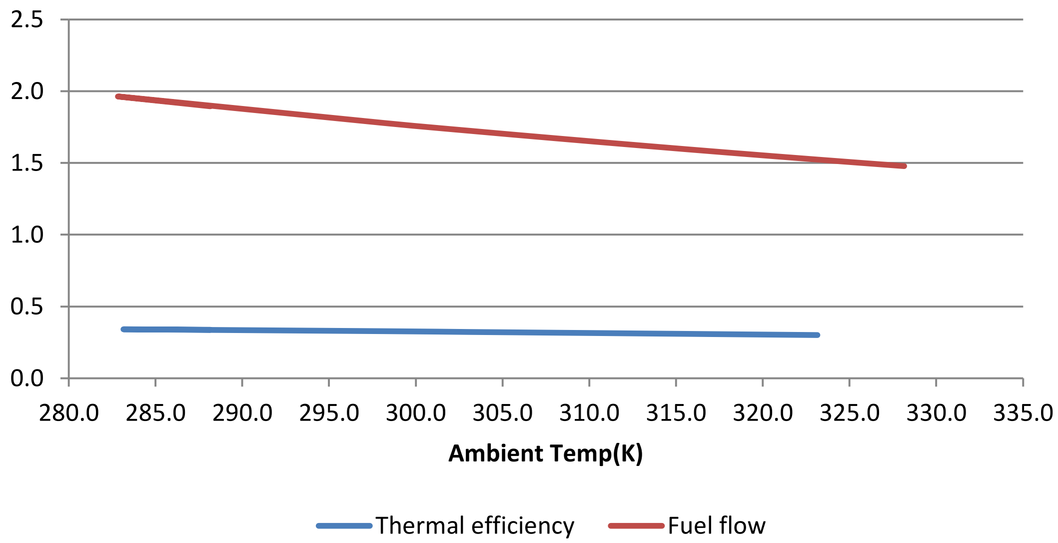

Figure 39: Shows Decline in Gas Turbine Thermal efficiency as Ambient Temperature rises

3.10 Summary

Turbo-match software has proven to be important tool in understanding gas turbine performance at different operating condition. TS3000 and H 25 engine were modelled, the off design performance has shown that the engine behaviour is dependent on the prevailing operating atmospheric condition. Thermal efficiency and power produced decreases with rise in ambient temperature while same engine exhibit better efficiency with less fuel consumption when operating in a cold atmospheric environment.

The Turbo-match revealed the effect of gas generator compressor fouling on the overall gas turbine performance. The reduction in flow capacity, mass flow and entire shift in compressor characteristics due to fouling is depicted in compressor map which is detrimental to gas turbine operation because the compressor equilibrium operating line is closer to the surge point.

The effectiveness of compressor washing in performance recovery is shown with simulation exercise which results into recovery of gas turbine flow capacity and thermal efficiency recovery.

4 Field Data Analysis

Many industries used gas turbine as their prime mover because of its known advantages over diesel engine for example and thus expectations are high in maximising its performance. Among other factors that contribute to gas turbine deterioration is its operating environment, mode of its operating condition i.e. load condition (base load or part load), type of driven equipment (alternator, pump, propeller). This chapter will investigate the performance of gas turbine, Hitachi H-25 model that is operating in coastal area, bonny terminal, Nigeria. The aim is to study the plant engine performance over period time and identify deterioration symptoms.

4.1 Plant (BOGT) Overview

The Bonny Oil and Gas Terminal (BOGT) located in onshore is mainly for crude oil sales, it receives inventories from every Shell oil fields located in eastern region, Nigeria. Crude oil is dehydrated, stabilised in BOGT before storage. It has capacity to handle 1.2million barrel of oil per day. The process crude oil is transported through offshore pipeline to the tankers and to Nigeria National Petroleum Company refinery. The terminal is the biggest in the entire country and thus its availability is paramount to the nation's economy. The plant supplies power to the neighbouring community as a social responsibility.

The Process Flow Scheme is illustrated figure 48 . Five incoming crude streams enter the Terminal and are blended into one common header. These are Land Light, Land Medium, Swamp Light, Cawthorne Light and Utapate Light

The blended crude is dehydrated partially and stabilized in Crude Dehydration and Stabilization unit consisting of five (5) Free Water Knock out (FWKO) trains. These are Trains A, B, C, D and E.

Following crude dehydration and stabilization, the treated crude is transferred to Crude Storage Tanks and the separated water is pumped to Produced Water Treatment plant consisting of one surge/buffer tank and five (5) trains.

The off gas (called Associated Gas) from the stabilization process is compressed and piped to the Nigerian Liquefied Natural Gas (NLNG) plant through the Bonny Non Associated Gas (BNAG) plant

Emulsion extracted from the Free Water Knock out (FWKO) during the crude dehydration process is sent to Emulsion Treatment plant consisting of three (3) trains and two (2) tanks. The emulsion is separated to oil and water, the oil returns back to the crude oil dehydration and stabilization unit while the water is piped to the water treatment unit.

The crude in Crude Storage Tanks is dehydrated to 0.5 % BS&W the required quality ready for export and transferred to the old and new Offshore Export lines or to the lines to the Refinery by booster pumps and export pumps.

Water from the FWKO trains, Emulsion treatment unit and Tank farm blend to a single header and sent to the Produced Water Treatment [50]

Electrical power for the Terminal is supplied by three (3) existing SOLAR turbine generator sets and three (3) new HITACHI turbine generator sets.

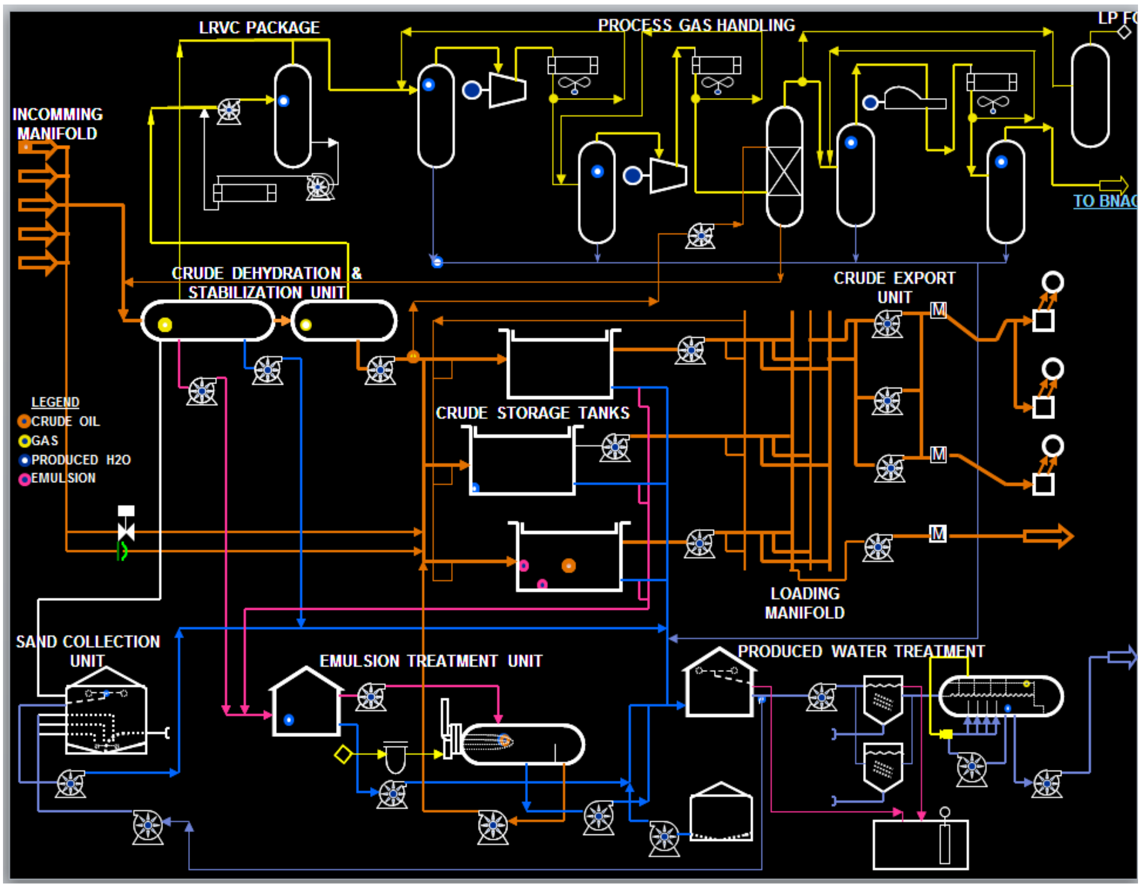

Figure 40: Shows BOGT Plant overview

The plant has three Hitachi, two solar, gas turbine for power generation. The power distribution system is three phase, 33kV with 50Hz supplied. This work will only be investigating the performance of one of the H25 model gas turbines at three different periods in its operation:

Period A 1548-3104 running hours (Fig 41)

Period B 3104-5650 running hours (Fig 42)

Period C 5650-7841 running hours (Fig 43)

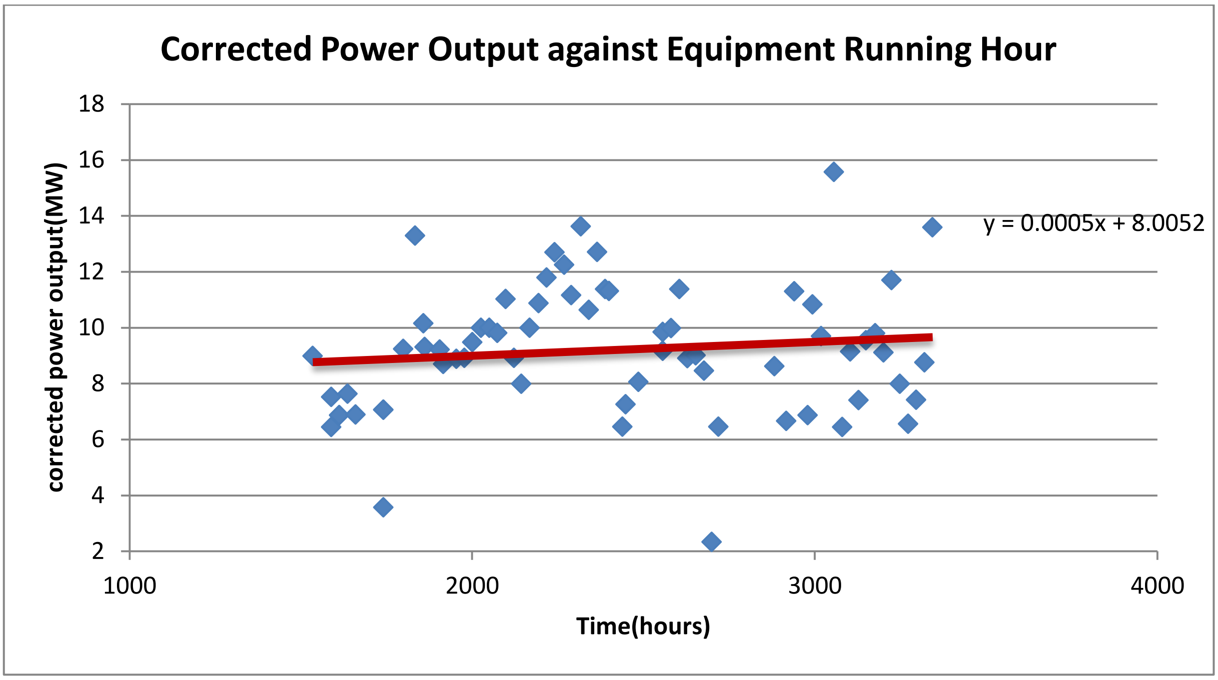

Figure 41: Shows Period A

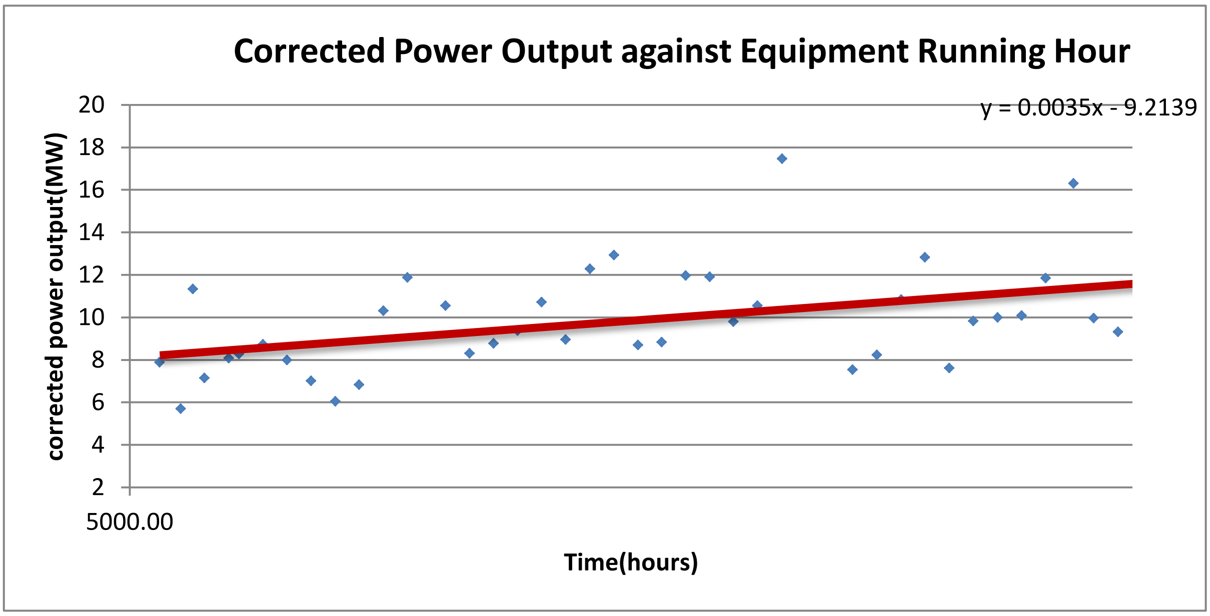

Figure 42: Show Period B

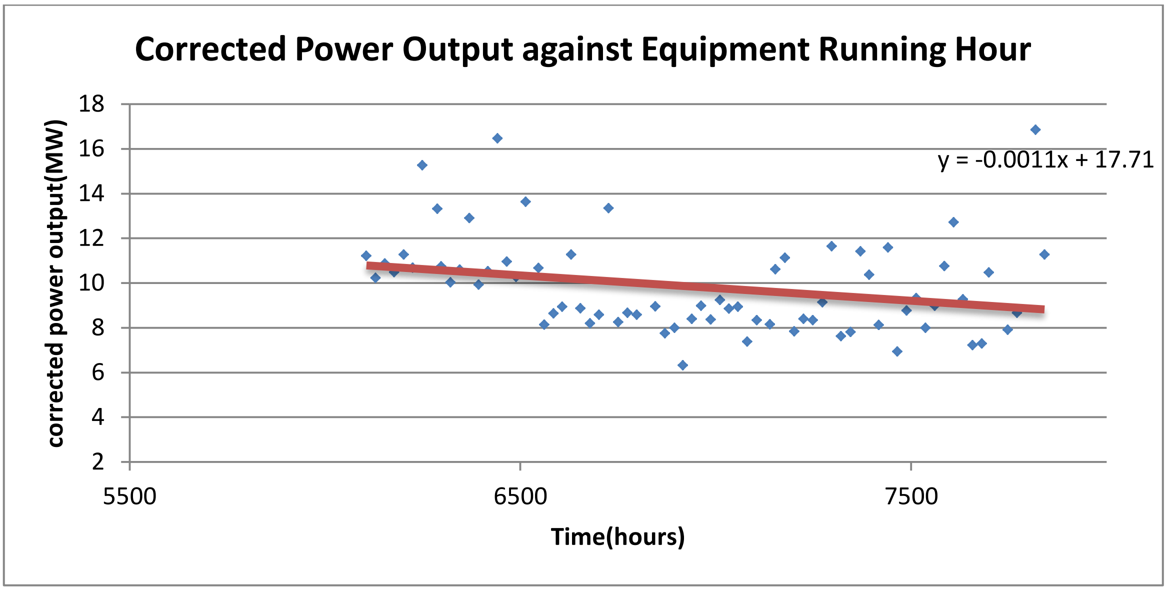

Figure 43: Shows Period C

Power output trend for different period within the investigation period show that there was increase in power output in period B which was as a result of two weekly online washing activities that was done during this period .However, there was reduction in power output over time at period C which was a result of no wash activities. Period A trend shows the engine performance shortly the engine was commissioned.

There were no available equipment running data recorded from day one the engine was commissioned till 1548 running period in the field logbook and this explains why the analysis did not start from the inception of the equipment.

However, the data appeared scattered due to dynamic condition of field operations. The average power utilised over the period of investigate is **9.5MW** with occasional higher power consumption of **17.1MW** when more crude oil loading pumps are required at the peak of vessel loading while power generation at off peak when no loading operation take place is around **2.2MW** from the plant that has capacity to generate up to **27.5MW** from a single Hitachi turbine.

4.2 Overview of Hitachi H25 Engine

H-25 model gas turbine is an open simple cycle, single shaft gas turbine with 17 stage axial compressor blade and three-turbine stages. The compressor pressure ratio is 14.7 with combustor exit temperature of 1260^{0}c and 27.5MW power capacity. [14]. It is equipped with pre-mix combustor of dry low NO_x emission of about 25ppm or less. Table 10 shows further H-25 gas turbine performance specifications available in the public. It also has multi-can combustor chamber (10 cans). Figure 44 shows the main components of H-25 model [15; 16].

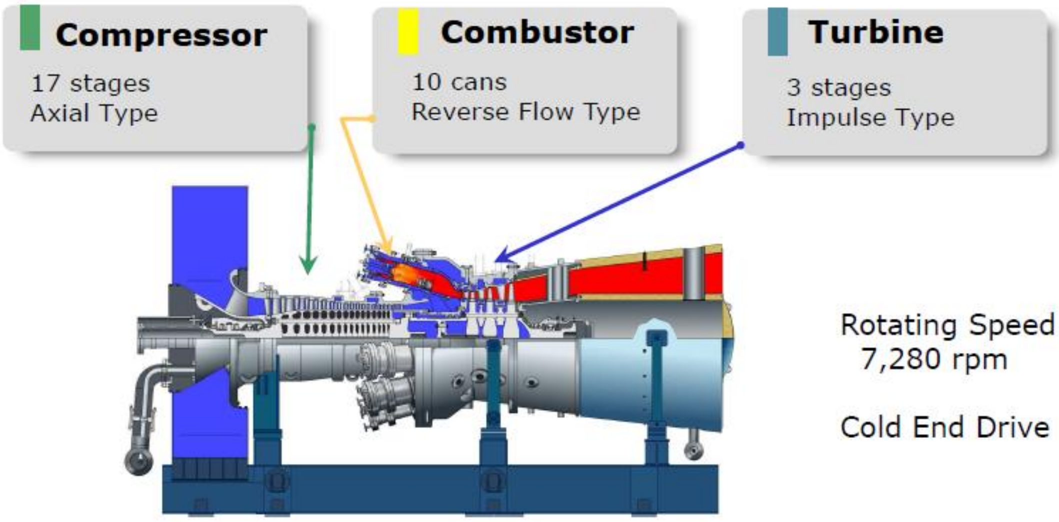

Figure 44: Main Components Arrangement of H25 Engine

Table 10: Shows H5 Gas Turbine Specification

Item	Unit	
Output	MW	27.5
Gross power generation efficiency	%	33.8
Airflow at compressor inlet	Kg/s	88
Exhaust	^{0}C	555
Pressure Ratio		14.7
Rotational Speed	RPM	7280

.

4.3 Engine Data Representation after 7841 Running Hours

Real engine data of Hitachi H25 (from August 2008 to September 2009) shortly after the engine was commissioned and run for 7841 running hours will be used for this part of study. The aim is to evaluate the performance over this period and to demonstrate thorough understanding of gas turbine performance. The engine data collected includes fuel flow (Kg/hr.), power output (MW), exhaust temperature, and running hours. The data were corrected to ISA condition and were converted to SI units.

Ambient temperature around the operating area usually hover between 22-28^{0}C on a typical day (Fig 45) but could go up to 35^{o}C during dry season while the average daily ambient temperature is around 26^{o}C

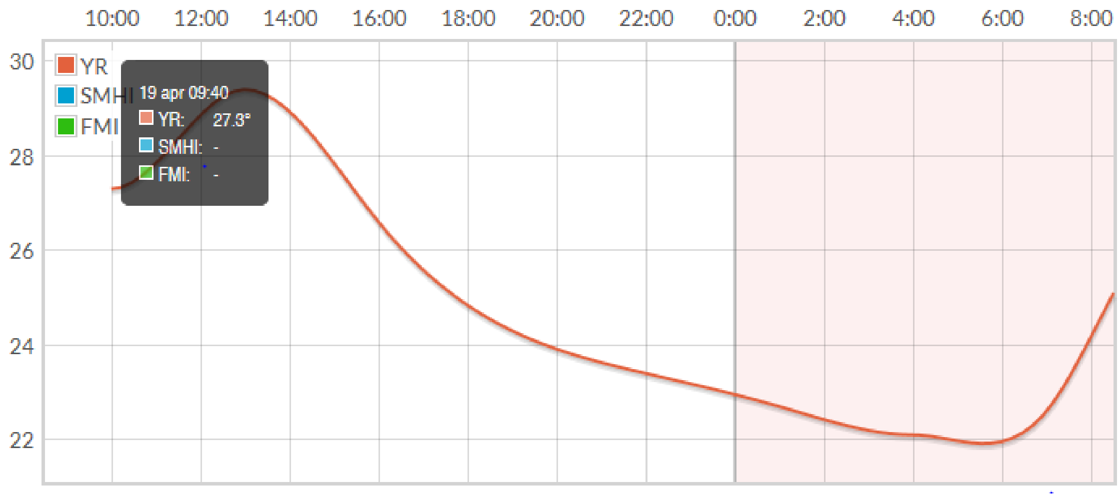

Figure 45: Shows Typical Daily Temperature Variation

Ambient condition pressure, humidity and temperature affect engine performance. The atmospheric pressure of the environment where the engine is operating experiences little or no change over times while relative humidity changes but there is no instrumentation installed to measure humidity on the engine, thus, their effect might be neglected

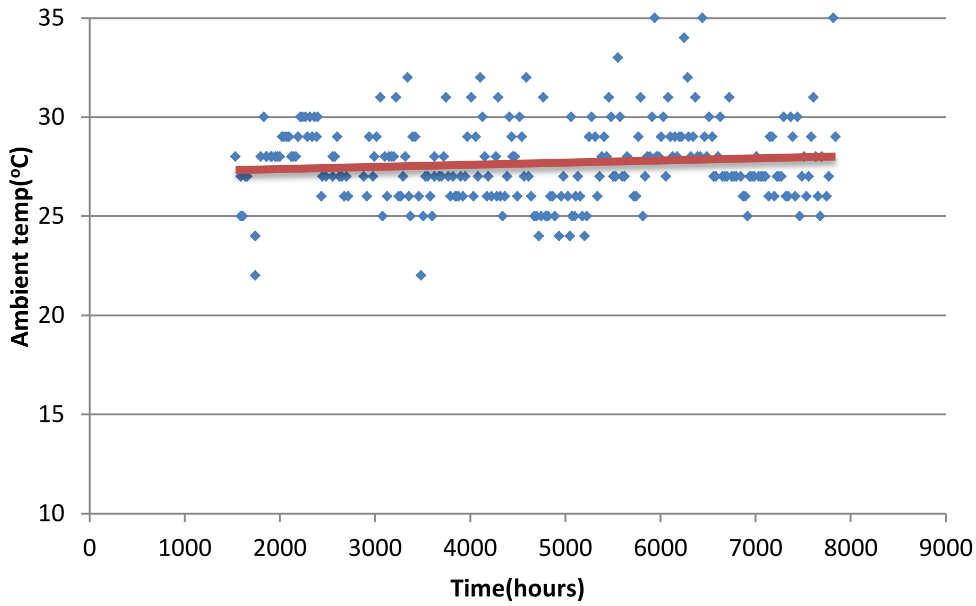

Figure 46: Shows Ambient Temperature Variation

The ambient temperature rise (Fig46) is usually in the afternoon and mostly around 5^{th} – 9^{th} month of the year. The trend of power consumption (Fig 47) in plant shows progressive demand which could be associated to installation of new equipment (additional illumination and pumps) and growth in the population of the communities. The investigation revealed that 6.1MW appears to be the regular power demand because it has higher number of frequency from the data gathered.

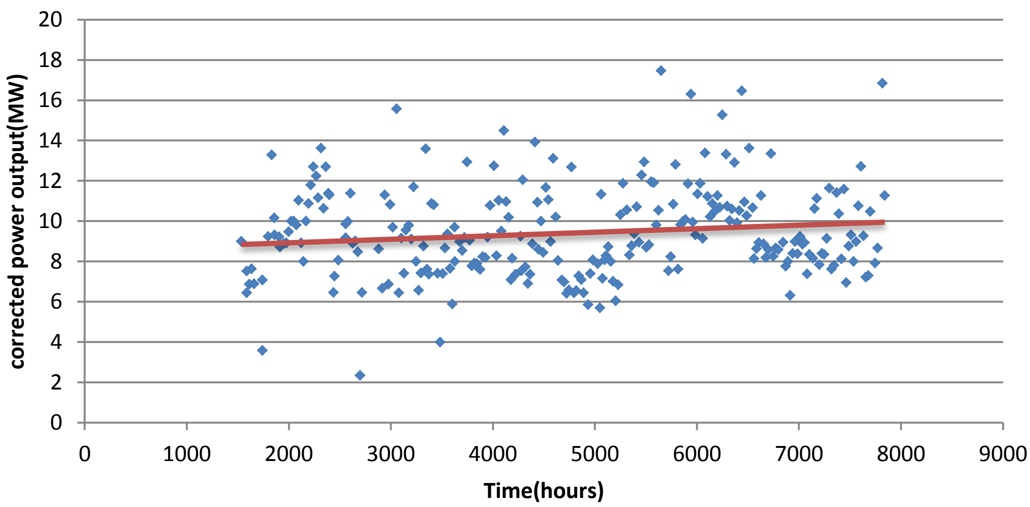

Figure 47: Corrected Power Output versus Running Hours

The average power consumption over the period of study is 8.86MW; the lowest being 2.2MW while the maximum power demand was 17.5MW. This shows that the engine is running on part load which necessitated low engine performance as indicated in fuel consumption analysis. The analysis shows that 28% of the engine capacity is being utilised during the period of this investigation.

The minimum load (2.2MW) recorded was at off peak period when vessels are not available to evacuate crude oil, thus plant operational activities are minimal. The maximum load was at the peak of operation when vessels were on queue to be filled with crude oil and the inventory from other locations was optimal.

The trend of fuel flow (Fig 48) over the period of study indicates progressive rise as power demand increases.

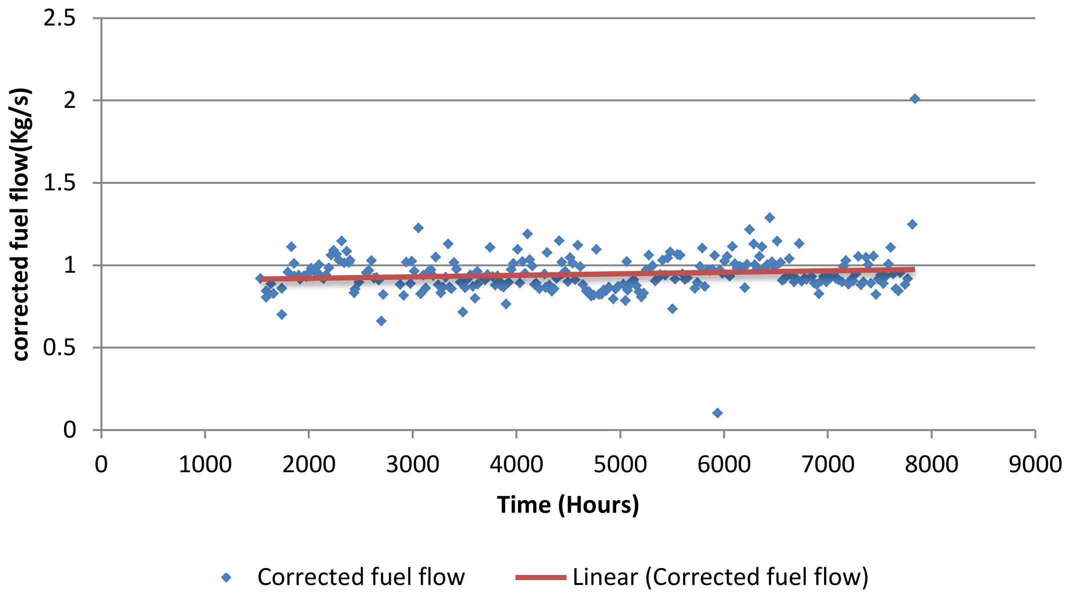

Figure 48: Corrected fuel Flow (kg/s) versus Running Hours

The engine performance (Fig 49-50) shows normal characteristics, increase in fuel supply as the load requirement changes and there is corresponding change in power output with rise in fuel flow.

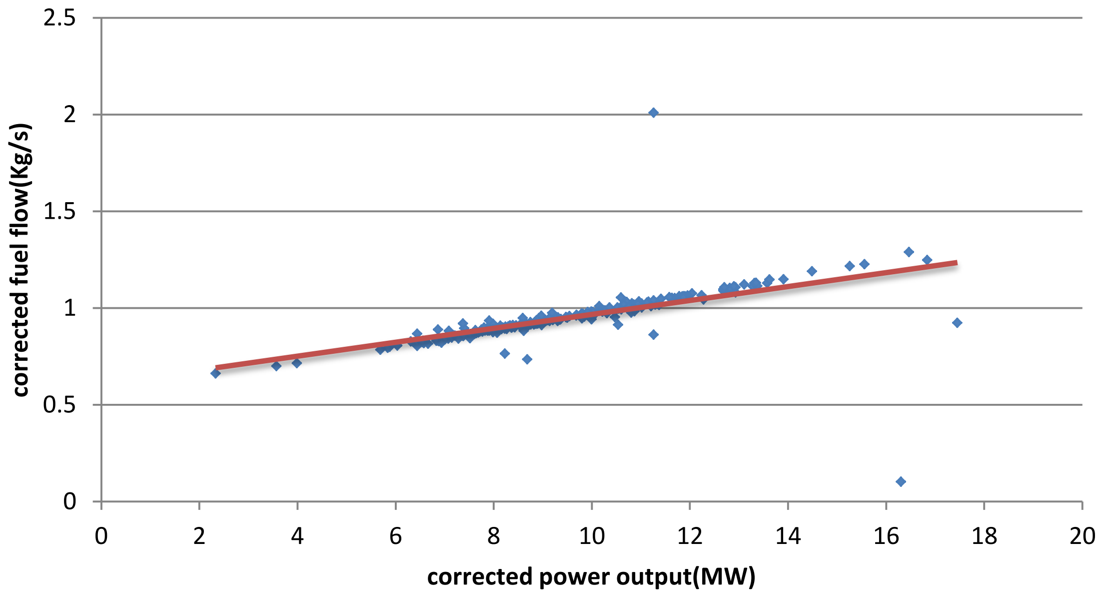

Figure 49: Shows Corrected Fuel Flow (kg/s) versus Corrected Power Output (MW)

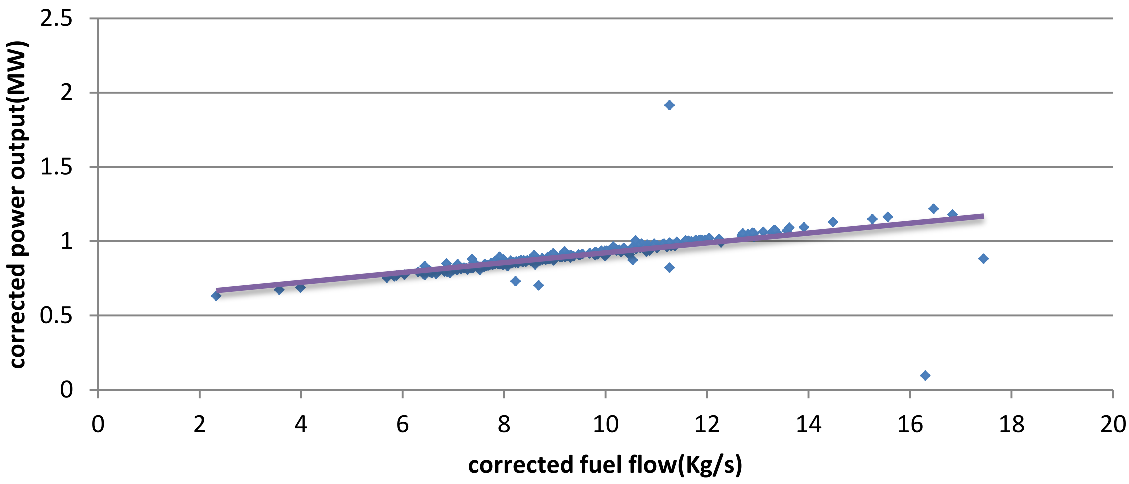

Figure 50: Shows Corrected Power output versus Corrected Fuel flow

4.4 Measurement Data Correction

Gas turbine operates under varying conditions of ambient temperature, pressure, rotating speed and load in the field. Though, the field data collected were representation of measurement data at engine steady state. The data analysed in this work is average data taken per day to eliminate errors and improve the quality of the data. In order to remove the effect of ambient conditions in field data, and estimate gas turbine performance at international standard atmospheric (ISO) condition (ambient temp.303.15K, atmospheric pressure 1.013 bars and relative humidity of 60%) the following correction formula were employed in this analysis.

$$W_{acorr} = \frac{w_a\sqrt{\frac{T_1}{288K}}}{\frac{p_1}{1.013\ bar}} \tag{4-1}$$

$$W_{fcorr} = \frac{w_f\sqrt{\frac{T_1}{288k}}}{\frac{P_1}{1.013bar}} \tag{4-2}$$

$$P_{corr} = \frac{P\sqrt{\frac{T_1}{288}}}{\frac{P_1}{1.013bar}} \tag{4-3}$$

$$T_5 = \frac{T_5}{\frac{T_1}{288K}} \quad (4\text{-}4)$$

The measured parameter is compared with corrected parameter (Fig 51-52) to eliminate the effect of ambient condition and trend engine performance at ISASLA condition. The corrected parameter is found to be slightly above the measured parameter.

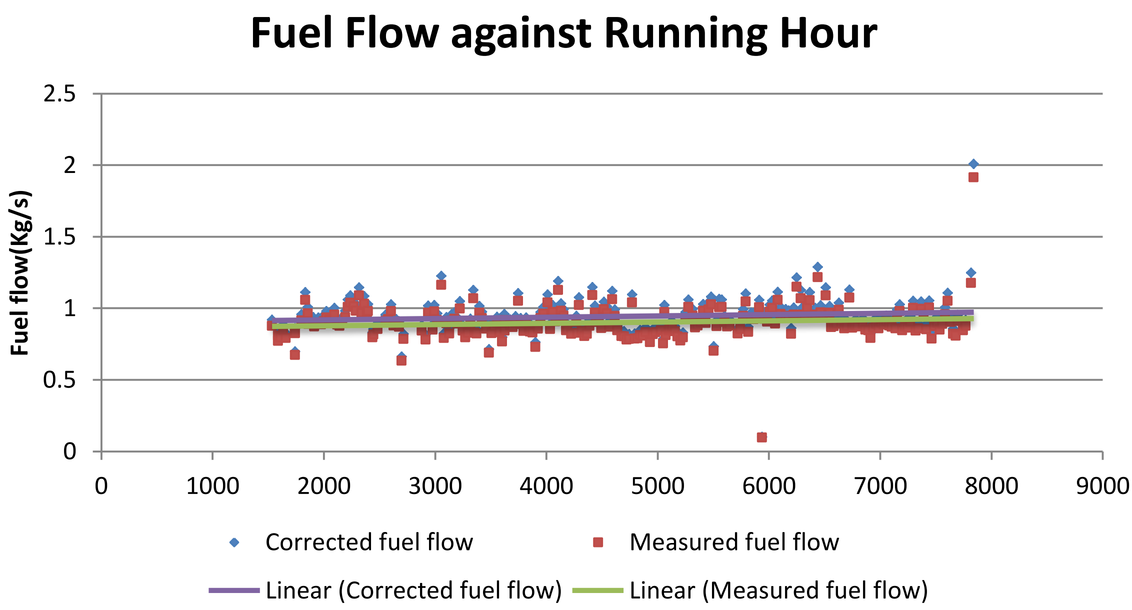

Figure 51: Comparison of corrected fuel Flow with measured Fuel Flow (kg/s)

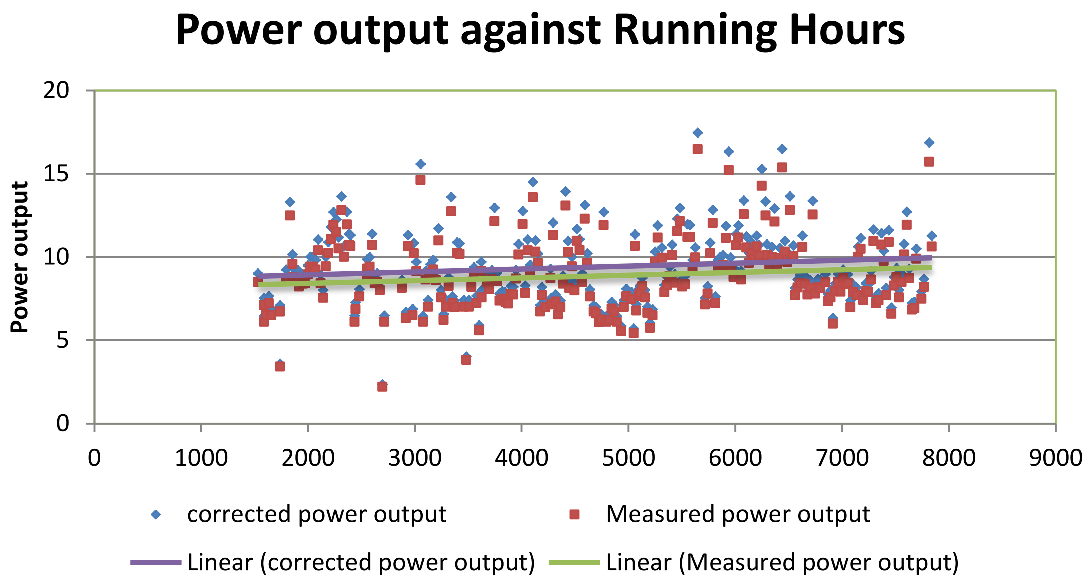

Figure 52: Comparison of Corrected power output with measured power output

4.5 Engine Control Mode

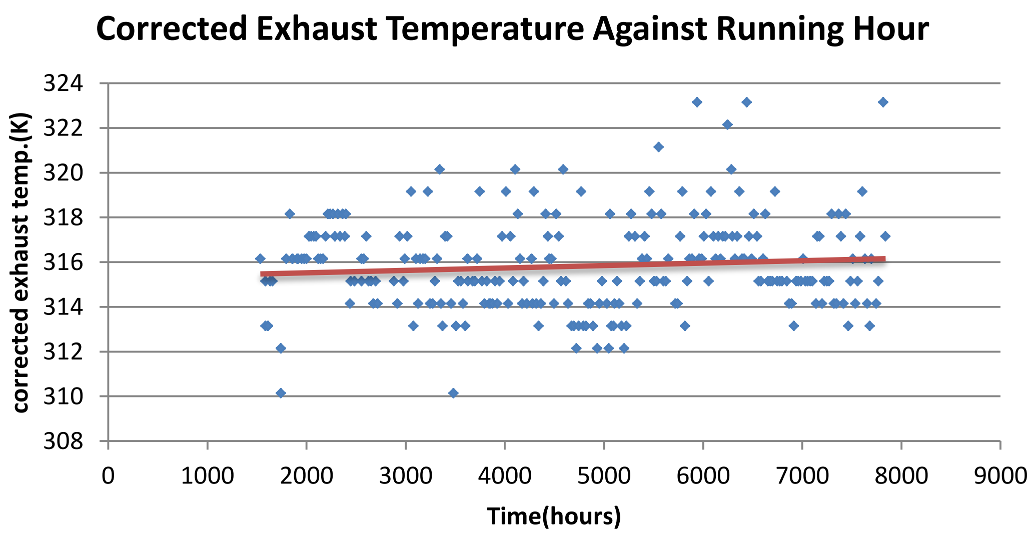

Figure 53: Corrected Exhaust Temperature versus Running Hours

Engines are controlled usually by either rotating speed or turbine entry temperature. It appears from the trend (Fig 53) that TET was this engine handles of control because the EGT trend maintains parallel straight line over the period of investigation. As the engine experiences increase in load requirement which is compensated by increase in fuel flow and proportional rise in exhaust temperature(Fig 54), TET being the handle of control responded by maintaining the EGT within the set temperature range (600K).

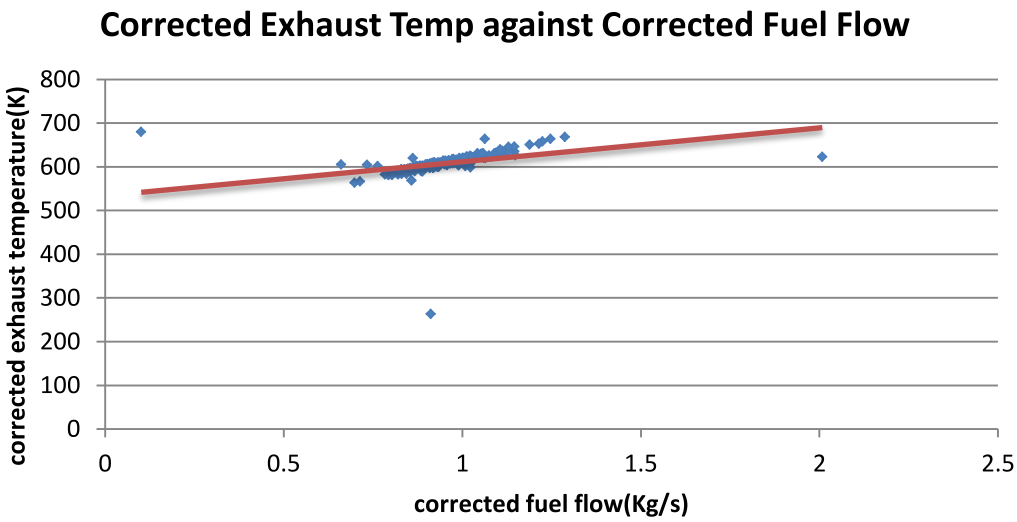

Figure 54: Corrected Exhaust Temperature (K) versus Corrected Fuel Flow (kg/s)

4.6 Engine Fuel Consumption

A good measure of engine efficiency is fuel consumption; the fuel flow of the engine over the period of time is investigated to determine the engine performance and its fuel consumption rate (Fig 55).

$$\text{Thermal Efficiency}=\frac{Power\ output}{Fuel\ supplied} \tag{4-5}$$

$$SFC = \frac{fuel\ flow\ rate}{useful\ power} \tag{4-6}$$

The fuel consumption was further investigated at different engine period of operations: Period A: 1535-3080, Period B: 3104-565, Period C: 5650-7841.It is observed that the rate of fuel consumption at low power setting 2.2MW is more than the fuel consumption rate at higher load requirement 6.1MW,which has frequency of appearance within the data collected.

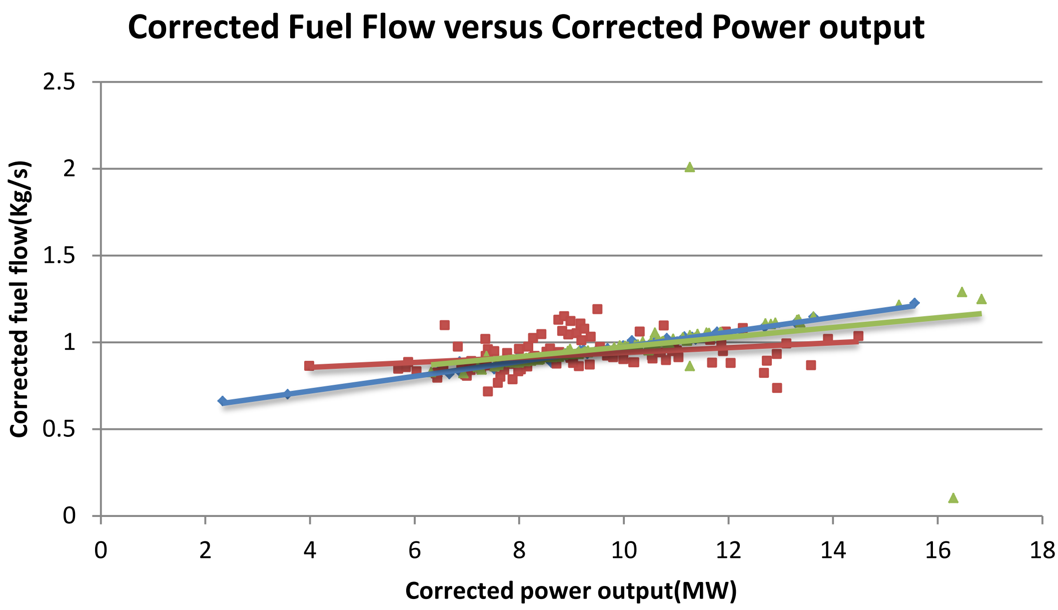

Figure 55: Corrected Fuel Flow versus Corrected Power Output

The fuel flow line during period A is intercepted by the fuel flow line of period B, the trend actually shows fuel flow reduction when compared with the trend indicated in period A. The improvement in fuel consumption was as a result of online water wash activities that were carried out on the engine. The trend during period C, shows rise in fuel consumption, which an indication of more fuel consumption. (Fig55)

The engine clean condition is taken as base line data (period B) when water wash was carried out, since, the engine performance at 0 operating hour is not available from the data collected. Table 9 depicts the fuel consumption for a given power output.

Table 11: Fuel Flow for a given Power Output

	Fuel flow			Deviation	
Power output	3014hrs	5650hrs	7841hrs	recovery(3014-5650)	degradation(5650-7841)
2.2	0.64	0.03	0.75	-0.95	23.30
6.1	0.81	0.09	0.86	-0.89	9.03
8.6	0.91	0.12	0.93	-0.86	6.69
8.86	0.92	0.12	0.94	-0.86	6.52
17.46	1.29	0.25	1.18	-0.80	3.79
Average				-0.88	9.87

$$\Delta \acute{z} = \frac{〚\Delta \acute{z}〛\_actual - 〚\Delta \acute{z}〛\_ideal}{\Delta \acute{z}_{ideal}} \qquad \textbf{(4-7)}$$

The fault signature ($\Delta \acute{z}$) denotes the changes in gas turbine quantity and this case; it represents engine fuel consumption from baseline condition which is a good indication of measurement of engine degradation over time.

Considering the most frequent load requirement of 6.1MW, the trend shows 9% increase in fuel consumption, the mean load of 8.86MW shows about 7% raise while maximum recorded 17.5MW load indicates 3.8% increase in fuel flow. The trend shows an average of 9.87% increase in fuel consumption over the period of investigation for an average load of 8.6MW while compressor washing show recovery of about 0.87% engine degradation.

The engine power out for a given fuel flow is trended (Fig 56)

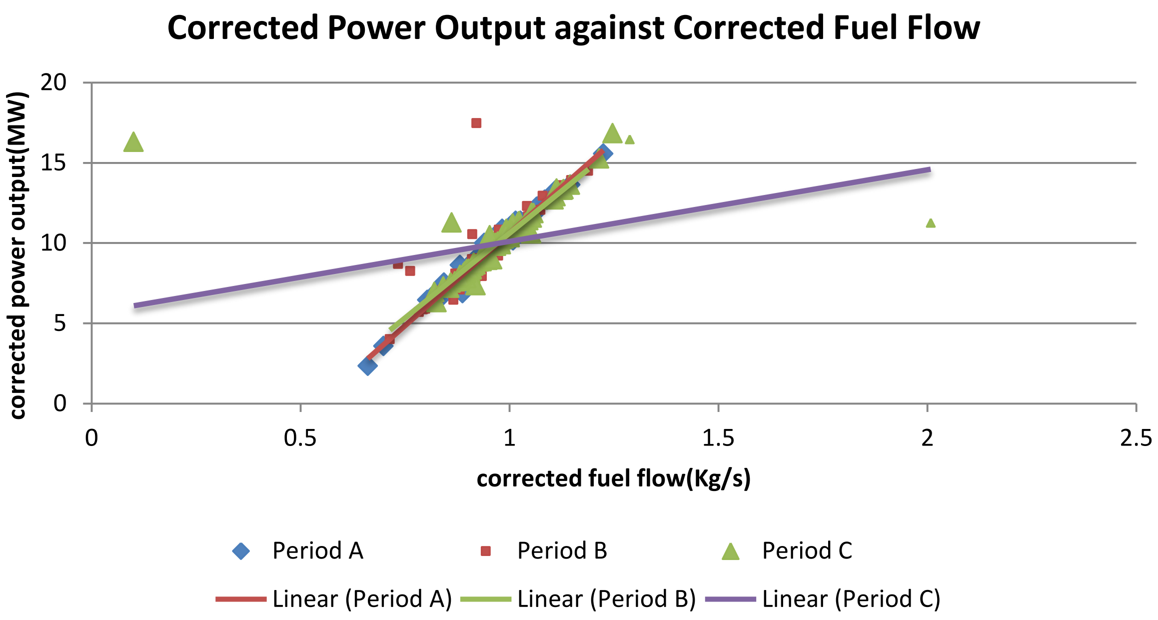

Figure 56: Corrected Power Output (MW) versus corrected Fuel Flow (kg/s)

Due compressor water washing maintenance activities during period B, the fuel flow rate gives almost equal power output as period A and this is indicated in the way the two trend matched into each other. Period C shows some increase in fuel flow as load increase but when compared with the fuel flow at higher power output, there is big reduction in Period C trend. Table 12 shows the power output for a given fuel flow

Table 12: Power output for a given fuel flow

	Power output			Deviation	
Fuel flow	3014hrs	5650hrs	7841hrs	Recovery(3104-5650)	Degradation (5650-7841)
0.8	5.99	6.38	9.21	-0.060	0.443
1	10.6	10.60	10.09	-0.0002	-0.047
1.2	15.20	14.82	10.99	0.0258	-0.258
Average				-0.0118	0.04567178

The trend shows an average of 4.5% reduction over the period for a given fuel flow. There is increase in power out from Period B when compared with period

C but the trend (power output) keep declining as the fuel supply increases. The compressor water wash activity shows power recovery of 1% when compared with period A

4.7 Summary

The general overview of the case study plant is given in the piping and instrumentation drawing included in this chapter and the overview of the gas turbine used in the plant. The data analysed shows typical plant energy consumption and the plant utilisation.

The engine data provided opportunity to study the turbine behaviour at different period within the operations. The high consumption in fuel is due to compressor fouling while effectiveness of compressor washing is evident in reduced fuel consumption for a given power output. The engine control mode was also identified from the data analysed which is shown Exhaust gas temperature consistency.

5 Economic Analysis of Hitachi Gas Turbine

The field data analysis executed in the previous chapter has provided a basis to x-ray the economic performance of Hitachi-H25 gas turbine used for power generation in bonny terminal, Nigeria. The engine designed for power requirement of 27.5MW at full capacity is being underutilised due to low power requirement in the plant as reflected in the available data. Though, this could extend the engine creep life because the engine most time operates below designed turbine inlet temperature of 1548K. However, the engine performance exhibit poor thermal efficiency as indicated in higher fuel consumption at part load. The operating cost of the engine at off design performance which involves engine performance at part load and prevailing ambient temperature were evaluated and benched mark with engine operating cost while operating at design point. The performance recovery in terms of fuel cost after compressor water wash is estimated to quantify the economic gains associated with the compressor washing. However, the cost of compressor washing module, washing fluid and the equipment downtime cost while carrying out cold wash are included in the cost evaluation. It is important to showcase the cost implication of the current power generation operating strategy to the company for better business decision in the future.

5.1 Economic Evaluation of Hitachi H25 at design point

The Turbo-match engine model result indicates Hitachi H25 attain its optimum performance of 27.5MW power output with 2.43Kg/s fuel consumption at engine rotational speed of 7280rpm and 28^{0}K ambient temperature.

Engine performance is better appreciated when presented in monetary gains, thus the need to estimate the plant profitability if the engine is running at its design point.

The fuel price for natural gas is calculated using the formula below:

Natural gas price for single spool engine fuel consumption is $4.39 per 1000 cubic foot

The unit of thousand cubic foot is converted to kilogram [45]

$$\left(16\frac{g}{mol}\right)\times\left(1000\frac{L}{m^3}\right)\times\left(22.4\frac{L}{mol}\right)=0.714\frac{Kg}{m^3} \qquad \textbf{(5-1)}$$

1000litres is contains in one cubic meter. Meanwhile 22.4 litres of any gas is contains in one mole. Molar mass of methane CH4 is 16 g per mol which means there are 0.714 kilograms of gas in a cubic meter.

$4.39 per thousand cubic foot = 1000 cubic foot = 28.3168 m^3 = 20.2062Kg

1kg = 1.4014m^3,therefore,2.43Kg/s = 3.40m^3/s = 120.07 cubic foot = $0.53

The cost of fuel consumption per year when at design point = fuel price per second * year = 0.53 * 3153600 = $1,671,408

5.2 Fuel Consumption Cost of Hitachi at Prevailing Operating Temperature

The engine operates in an atmosphere which is completely different from the design operating temperature. This results into increase in fuel flow and rise in the engine operating fuel cost, which takes its toll in plant profitability. The off design simulation exercise gives 2.55kg/s as fuel flow under normal ambient temperature of the engine operating environment.

The fuel price for natural gas is calculated using the formula below:

Natural gas price for single spool engine fuel consumption is $4.39 per 1000 cubic foot

The unit of thousand cubic foot is converted to kilogram,

1000litres is contains in one cubic meter. Meanwhile 22.4 litres of any gas is contains in one mole. Molar mass of methane CH_4 is 16 g per mol which means there are 0.714 kilograms of gas in a cubic meter.

$4.39 per thousand cubic foot = 1000 cubic foot = 28.3168 m^3 = 20.2062Kg

1kg = 1.4014m^3,therefore,2.55Kg/s = 3.58m^3/s = 126.43 cubic foot = $0.56

Fuel cost per year for single spool engine running under high ambient temperature = $0.56 * 3153600 = $1,766,016

5.3 Economic Analysis of Compressor Washing

The results obtained from the data analysis show that the fuel consumption for a given power output decreases due to two weekly compressor online water wash.

Estimation of the fuel cost while running the engine without water wash:

For a given power output of 6.1MW the fuel consumption is 0.86kg/s.

Note: \$4.39 per thousand cubic foot = 1000 cubic foot = 28.3168 m^3 = 20.2062Kg, 1kg = 1.4014m^3,

Therefore,0.86Kg/s = 1.204m^3/s = 42.52 cubic foot = \$0.19

The engine fuel consumption cost per year without compressor wash = \$0.19* 315600 = \$59,964

Estimation of the fuel cost while online water wash is done:

For given power output of 6.1MW the fuel consumption is 0.09kg/s

The fuel consumption cost of 0.09kg/s = 0.126m^3/s = 4.45 cubic foot = \$0.02

The engine fuel consumption cost per year with online water wash = 0.02* 315600 = \$6,312

Cost Analysis of Two weekly Compressor online water wash

It is important to estimate the cost implication of turbine online water wash activities. Turbine requires minimum or no disruption during this operation and thus no significant power loss is accounted for. However, the cost of detergent, demineralised water and the initial capital of water wash module are significant in the operation and needs to be included in the plant economic analysis. The cost of labour is negligible because the operation require no special technicality

Cost of 25litres of water wash detergent = \$2,300

Cost of 100litres of Demineralised water = \$135.57

Capital cost of water wash module = \$10,000

The total cost of water wash (Fixed cost and recurrent cost) = $12,435.57

The total cost of fuel consumption per year + the cost of Two weekly water wash = $6,312 + $12,435.57 = $18,747.57

Net gain in fuel consumption cost = fuel consumption cost without water wash – total fuel consumption with online water wash = $59,964 – $18,747.57 = $41,216.43

5.4 Summary

The results of field analysis have shown that two weekly online compressor washing is profitable for the plant operation despite additional recurrent cost of water wash. It reduces the plant operating cost tremendously through reduction in plant fuel consumption which takes greater per cent in plant running cost. The cost saving derived from two weekly compressor online water wash annually is estimated around $41,216.Besides, the plant has shown higher fuel consumption cost of about $1000 while operating at higher ambient temperature different from designed operating ambient temperature of 28K.

6 Conclusions and Recommendations

6.1 Conclusions

The main objective of this research has been to evaluate the effectiveness of gas turbine axial compressor online water wash, in relation to gas turbine performance recovery.

This work has shown other forms of degradations that have higher tendency to shorten the engine life cycle through catastrophe failure of the engine components. However, the failure frequency from this type of irrecoverable degradation such as foreign object damage, erosion, corrosion and hot corrosion are low, preventable and could be eliminated throughout the life of the machine by conscious effort from the plant operator and maintenance team.

Regular washing of axial compressor has become inevitable in order for the company to maximise her economic gains, optimise equipment utilisation and reliability. It is shown that installation of high quality filtration system only reduces the quantity of dirt ingestion into turbine but significant amount penetrates through the filter overtime and adhere to the axial blade in accordance with fouling mechanism, as discussed in this work. Though, technological improvement incorporated in filtration system like filter self-cleaning system only elongate filter life usage but huge dirt accumulates on the filters which shows the volume of particulate that are present in the mass flow of air consumes by the engine and within the turbine inlet volute.

The use of compressor washing detergent (ZOK27) has proven to be friendly to both machine and human unlike other washing chemical which have negative impact on environment, attack engine component and without safety guarantee for the users. It is effective in dissolution of dirt on the blade, aqueous and non-explosive. These important characteristic features of ZOK 27 enhanced the effectiveness of the compressor online washing.

The online compressor effectiveness clearly manifested in the recovery of power output loss from 25.2MW to 26.2MW which is equivalent to about 4.7%

power recovery from Turbomatch simulation while the field data analysis also revealed an estimation of $41,000 fuel cost saving due to two weekly online compressor regular water wash which is a huge benefit

6.2 Recommendations

For future research work it is recommended that the engine specific fuel consumption at design point is provided by the manufacturer to serve as basis for comparison on the equipment performance overtime. Although, the result from turbo-match was used for the economic analysis, the specific equipment design considerations will provide more accurate analysis.

Further work on techno-economic analysis should give considerations to the effect of relative humidity (RH) in engine performance, the operator logbook clearly omitted RH in daily equipment parameter logging. The frequency of field equipment running parameters use for analysis should be on per seconds interval as against per hour owing to the fact that field running conditions change regularly. This will provide more accuracy in the data analysis and eliminate scattered graphical result representation.

For regular implementation, online compressor washing should be included in company Computerised Maintenance Management system such as System Application Product software (SAP) which will prompt operator and maintenance team to execute this activity as part of equipment maintenance plan.

Robust maintenance plan should be developed for turbine ancillaries to reduce unscheduled equipment breakdown and improve equipment reliability.

Technical capability of maintenance staff is recommended to be regularly updated through both class room and on the job training. This will enhance the quality of technical intervention and reduce equipment MTTR.

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APPENDICES

Whilst Heading 1 to Heading 6 can be used to number headings in the main body of the thesis, Heading styles 7–9 have been modified specifically for lettered appendix headings with Heading 7 having the ‘Appendix’ prefix as shown below.

Appendix A

```
! TURBOMATCH SCHEME - Windows NT version (October 1999)

 ! LIMITS:100 Codewords, 800 Brick Data Items, 50 Station Vector

  !15 BD Items printable by any call of:-

  ! OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV

 ! GENERAL ELECTRIC LM2500 PLUS DESIGN POINT

! MODELLED BY ABASS KABIR, SOE, JANUARY 2013

! Turbo-match programme: Design Point simulation of GE LM 2500 plus Industrial Gas Turbine

 ////

OD SI KE VA XP

-1

-1

INTAKE S1, 2     D1-4            R100

COMPRE S2, 3     D5-11           R101     V5       V6

PREMAS S3, 12,4  D12-15

PREMAS S4,13,5  D16-19

BURNER S5,6     D20-22          R102

MIXEES S6,13,7

TURBIN S7,8     D23-30,101,31            V24

MIXEES S8,12,9

TURBIN S9,10    D32-41                   V32 V33
```

```
NOZCON S10, 11,1 D41              R107

PERFOR  S1,0,0   D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

CODEND

DATA ITEMS////

! INTAKE

1 0.0          ! INTAKE ALTITUDE

2 0.0          ! ISA DEVIATION

3 0.0          ! MACH NO

4 0.9951       ! PRESSURE RECOVERY

! COMPRESSOR

5 -1.0         ! Z PARAMETER

6 -1.0         ! ROTATIONAL SPEED N

7 22.0         ! PRESSURE RATIO

8 0.8825       ! ISENTROPIC EFFICIENCY

9 0.0          ! ERROR SELECTION

10 3.0         ! MAP NUMBER

11 0.0         ! STATOR ANGLE RELATIVE TO DP

! PREMAS

12 0.049       ! BLEED AIR

13 0.0         ! FLOW LOSS

14 1.0         ! PRESSURE RECOVERY

15 0.0         ! PRESSURE DROP

! PREMAS

16 0.095       ! BLEED AIR

17 0.0         ! FLOW LOSS

18 1.0         ! PRESSURE RECOVERY
```

19 0.0 ! PRESSURE DROP

! BURNER

20 0.075 ! FRACTIONAL PRESSURE LOSS DP/P

21 0.9995 ! COMBUSTION EFFICIENCY

22 -1.0 ! FUEL FLOW

! HP TURBINE

23 0.0 ! AUXILIARY WORK

24 -1.0 ! NDMF

25 0.6 ! NDSPEED CN

26 0.9 ! ISENTROPIC EFFICIENCY

27 -1.0 ! PCN

28 1.0 ! COMPRESSOR NUMBER

29 5.0 ! TURBINE MAP NUMBER

30 -1.0 ! POWER LOW INDEX

31 0.0 ! NGV ANGLE RELATIVE TO DP

! POWER TURBINE

32313320.00 ! AUXILIARY WORK

33 -1.0 ! NDMF

34 0.6 ! NDSPEED CN

35 0.9 ! ISENTROPIC EFFICIENCY

36 1.0 ! PCN

37 0.0 ! COMPRESSOR NUMBER

38 5.0 ! MAP NUMBER

39 -1.0 ! POWER LAW INDEX

40 -1.0 ! COMWORK

41 0.0 ! NGV ANGLE RELATIVE TO DP

```
! NOZCON
42 -1.0       ! THROAT AREA
! PERFOR
43 1.0        ! PROPELLER EFFICIENCY
44 0.0        ! SCALING INDEX
45 0.0        ! REQUIRED THRUST
-1
1 2 82.55     ! INLET MASS FLOW
6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE
-1
-3

! TURBOMATCH SCHEME - Windows NT version (October 1999)
 ! LIMITS:100 Codewords, 800 Brick Data Items, 50 Station Vector
  !15 BD Items printable by any call of:-
  ! OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV
 ! GENERAL ELECTRIC LM2500 PLUS OFF DESIGN
  ! MODELLED BY ABASS KABIR, SOE, JANUARY 2013
! Turbo-match programme: Off Design performance simulation for different ambient condition
  ////
OD SI KE VA XP
-1
-1
INTAKE S1,2    D1-4          R100
COMPRE S2,3    D5-11         R101    V5      V6
PREMAS S3,12,4  D12-15
```

```
PREMAS S4,13,5  D16-19

BURNER S5,6     D20-22          R102

MIXEES S6,13,7

TURBIN S7,8     D23-30,101,31           V24

MIXEES S8,12,9

TURBIN S9,10    D32-41                  V32 V33

NOZCON S10,11,1 D41             R107

PERFOR  S1,0,0   D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

CODEND

DATA ITEMS////

! INTAKE

1 0.0         ! INTAKE ALTITUDE

2 0.0         ! ISA DEVIATION

3 0.0         ! MACH NO

4 0.9951      ! PRESSURE RECOVERY

! COMPRESSOR

5 -1.0        ! Z PARAMETER

6 -1.0        ! ROTATIONAL SPEED N

7 22.0        ! PRESSURE RATIO

8 0.8825      ! ISENTROPIC EFFICIENCY

9 0.0         ! ERROR SELECTION

10 3.0        ! MAP NUMBER

11 0.0        ! STATOR ANGLE RELATIVE TO DP

! PREMAS

12 0.049      ! BLEED AIR

13 0.0        ! FLOW LOSS
```

14 1.0 ! PRESSURE RECOVERY

15 0.0 ! PRESSURE DROP

! PREMAS

16 0.095 ! BLEED AIR

17 0.0 ! FLOW LOSS

18 1.0 ! PRESSURE RECOVERY

19 0.0 ! PRESSURE DROP

! BURNER

20 0.075 ! FRACTIONAL PRESSURE LOSS DP/P

21 0.9995 ! COMBUSTION EFFICIENCY

22 -1.0 ! FUEL FLOW

! HP TURBINE

23 0.0 ! AUXILIARY WORK

24 -1.0 ! NDMF

25 0.6 ! NDSPEED CN

26 0.9 ! ISENTROPIC EFFICIENCY

27 -1.0 ! PCN

28 1.0 ! COMPRESSOR NUMBER

29 5.0 ! TURBINE MAP NUMBER

30 -1.0 ! POWER LOW INDEX

31 0.0 ! NGV ANGLE RELATIVE TO DP

! POWER TURBINE

32313320.00! AUXILIARY WORK

33 -1.0 ! NDMF

34 0.6 ! NDSPEED CN

35 0.9 ! ISENTROPIC EFFICIENCY

```
36 1.0        ! PCN
37 0.0        ! COMPRESSOR NUMBER
38 5.0        ! MAP NUMBER
39 -1.0       ! POWER LAW INDEX
40 -1.0       ! COMWORK
41 0.0        ! NGV ANGLE RELATIVE TO DP
! NOZCON
42 -1.0       ! THROAT AREA
! PERFOR
43 1.0        ! PROPELLER EFFICIENCY
44 0.0        ! SCALING INDEX
45 0.0        ! REQUIRED THRUST

-1
1 2 82.55     ! INLET MASS FLOW
6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE
-1
2 -15.0       ! OD CALCULATION; DT = 0, ISA 15 DEG
-1
-1
2 -10.0       ! OD CALCULATION; DT = 0, ISA 15 DEG
-1
-1
2 -5.0        ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
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```
2 -0.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 5.0           ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 10.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 15.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 20.0          ! OD CALCULATION; DT = 0; ISA 15DEG
-1
-1
2 25.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 30.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 35.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 40.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
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-3

A.1

```
! TURBOMATCH SCHEME - Windows NT version (October 1999)
 ! LIMITS:100 Codewords, 800 Brick Data Items, 50 Station Vector
  !15 BD Items printable by any call of:-
  ! OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV
 ! GENERAL ELECTRIC LM2500 PLUS DESIGN POINT
! MODELLED BY ABASS KABIR, SOE, JANUARY 2013
! Turbo-match programme: Design Point simulation of GE LM 2500 plus Industrial Gas Turbine
 ////
OD SI KE VA XP
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-1
INTAKE S1, 2     D1-4          R100
COMPRE S2, 3     D5-11          R101     V5      V6
PREMAS S3, 12,4  D12-15
PREMAS S4,13,5  D16-19
BURNER S5,6     D20-22         R102
MIXEES S6,13,7
TURBIN S7,8     D23-30,101,31          V24
MIXEES S8,12,9
TURBIN S9,10    D32-41                 V32 V33
NOZCON S10, 11,1 D41             R107
```

```
PERFOR  S1,0,0   D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

CODEND

DATA ITEMS////

! INTAKE

1 0.0         ! INTAKE ALTITUDE

2 0.0         ! ISA DEVIATION

3 0.0         ! MACH NO

4 0.9951      ! PRESSURE RECOVERY

! COMPRESSOR

5 -1.0        ! Z PARAMETER

6 -1.0        ! ROTATIONAL SPEED N

7 22.0        ! PRESSURE RATIO

8 0.8825      ! ISENTROPIC EFFICIENCY

9 0.0         ! ERROR SELECTION

10 3.0        ! MAP NUMBER

11 0.0        ! STATOR ANGLE RELATIVE TO DP

! PREMAS

12 0.049      ! BLEED AIR

13 0.0        ! FLOW LOSS

14 1.0        ! PRESSURE RECOVERY

15 0.0        ! PRESSURE DROP

! PREMAS

16 0.095      ! BLEED AIR

17 0.0        ! FLOW LOSS

18 1.0        ! PRESSURE RECOVERY

19 0.0        ! PRESSURE DROP
```

```
! BURNER
20 0.075      ! FRACTIONAL PRESSURE LOSS DP/P
21 0.9995        ! COMBUSTION EFFICIENCY
22 -1.0       ! FUEL FLOW
! HP TURBINE
23 0.0        ! AUXILIARY WORK
24 -1.0       ! NDMF
25 0.6        ! NDSPEED CN
26 0.9        ! ISENTROPIC EFFICIENCY
27 -1.0       ! PCN
28 1.0        ! COMPRESSOR NUMBER
29 5.0        ! TURBINE MAP NUMBER
30 -1.0       ! POWER LOW INDEX
31 0.0        ! NGV ANGLE RELATIVE TO DP
! POWER TURBINE
32313320.00  ! AUXILIARY WORK
33 -1.0       ! NDMF
34 0.6        ! NDSPEED CN
35 0.9        ! ISENTROPIC EFFICIENCY
36 1.0        ! PCN
37 0.0        ! COMPRESSOR NUMBER
38 5.0        ! MAP NUMBER
39 -1.0       ! POWER LAW INDEX
40 -1.0       ! COMWORK
41 0.0        ! NGV ANGLE RELATIVE TO DP
! NOZCON
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42 -1.0       ! THROAT AREA

! PERFOR

43 1.0        ! PROPELLER EFFICIENCY

44 0.0        ! SCALING INDEX

45 0.0        ! REQUIRED THRUST

-1

1 2 82.55     ! INLET MASS FLOW

6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE

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A.2

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! TURBOMATCH SCHEME - Windows NT version (October 1999)

 ! LIMITS:100 Codewords, 800 Brick Data Items, 50 Station Vector

  !15 BD Items printable by any call of:-

  ! OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV

 ! GENERAL ELECTRIC LM2500 PLUS OFF DESIGN

  ! MODELLED BY ABASS KABIR, SOE, JANUARY 2013

! Turbo-match programme: Off Design performance simulation for different ambient condition

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OD SI KE VA XP

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INTAKE S1,2    D1-4          R100

COMPRE S2,3    D5-11         R101    V5       V6
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```
PREMAS S3,12,4  D12-15

PREMAS S4,13,5  D16-19

BURNER S5,6     D20-22          R102

MIXEES S6,13,7

TURBIN S7,8     D23-30,101,31             V24

MIXEES S8,12,9

TURBIN S9,10    D32-41                    V32 V33

NOZCON S10,11,1 D41              R107

PERFOR  S1,0,0   D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

CODEND

DATA ITEMS////

! INTAKE

1 0.0          ! INTAKE ALTITUDE

2 0.0          ! ISA DEVIATION

3 0.0          ! MACH NO

4 0.9951       ! PRESSURE RECOVERY

! COMPRESSOR

5 -1.0         ! Z PARAMETER

6 -1.0         ! ROTATIONAL SPEED N

7 22.0         ! PRESSURE RATIO

8 0.8825       ! ISENTROPIC EFFICIENCY

9 0.0          ! ERROR SELECTION

10 3.0         ! MAP NUMBER

11 0.0         ! STATOR ANGLE RELATIVE TO DP

! PREMAS

12 0.049       ! BLEED AIR
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13 0.0 ! FLOW LOSS

14 1.0 ! PRESSURE RECOVERY

15 0.0 ! PRESSURE DROP

! PREMAS

16 0.095 ! BLEED AIR

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! BURNER

20 0.075 ! FRACTIONAL PRESSURE LOSS DP/P

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23 0.0 ! AUXILIARY WORK

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25 0.6 ! NDSPEED CN

26 0.9 ! ISENTROPIC EFFICIENCY

27 -1.0 ! PCN

28 1.0 ! COMPRESSOR NUMBER

29 5.0 ! TURBINE MAP NUMBER

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! POWER TURBINE

32313320.00! AUXILIARY WORK

33 -1.0 ! NDMF

34 0.6 ! NDSPEED CN

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35 0.9        ! ISENTROPIC EFFICIENCY

36 1.0        ! PCN

37 0.0        ! COMPRESSOR NUMBER

38 5.0        ! MAP NUMBER

39 -1.0       ! POWER LAW INDEX

40 -1.0       ! COMWORK

41 0.0        ! NGV ANGLE RELATIVE TO DP

! NOZCON

42 -1.0       ! THROAT AREA

! PERFOR

43 1.0        ! PROPELLER EFFICIENCY

44 0.0        ! SCALING INDEX

45 0.0        ! REQUIRED THRUST


-1

1 2 82.55     ! INLET MASS FLOW

6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE

-1

2 -15.0       ! OD CALCULATION; DT = 0, ISA 15 DEG

-1

-1

2 -10.0       ! OD CALCULATION; DT = 0, ISA 15 DEG

-1

-1

2 -5.0        ! OD CALCULATION; DT = 0; ISA 15 DEG

-1
```

```
-1
2 -0.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 5.0           ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 10.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 15.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 20.0          ! OD CALCULATION; DT = 0; ISA 15DEG
-1
-1
2 25.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 30.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 35.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
-1
-1
2 40.0          ! OD CALCULATION; DT = 0; ISA 15 DEG
```

-1

-1

-3

A.3

OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV

! GENERAL ELECTRIC LM2500 PLUS DESIGN POINT

! MODELLED BY ABASS KABIR, SOE, JANUARY 2013

! Turbo match programme: Off Design TET Handle simulation of GE LM 2500plus Industrial Gas turbine

////

OD SI KE VA XP

-1

-1

INTAKE S1,2 D1-4 R100

COMPRE S2,3 D5-11 R101 V5 V6

PREMAS S3,12,4 D12-15

PREMAS S4,13,5 D16-19

BURNER S5,6 D20-22 R102

MIXEES S6,13,7

TURBIN S7,8 D23-30,101,31 V24

MIXEES S8,12,9

TURBIN S9,10 D32-41 V32 V33

NOZCON S10,11,1 D41 R107

PERFOR S1,0,0 D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

```
CODEND

DATA ITEMS////

! INTAKE

1 0.0          ! INTAKE ALTITUDE

2 0.0          ! ISA DEVIATION

3 0.0          ! MACH NO

4 0.9951       ! PRESSURE RECOVERY

! COMPRESSOR

5 -1.0         ! Z PARAMETER

6 -1.0         ! ROTATIONAL SPEED N

7 22.0         ! PRESSURE RATIO

8 0.8825       ! ISENTROPIC EFFICIENCY

9 0.0          ! ERROR SELECTION

10 3.0         ! MAP NUMBER

11 0.0         ! STATOR ANGLE RELATIVE TO DP

! PREMAS

12 0.049       ! BLEED AIR

13 0.0         ! FLOW LOSS

14 1.0         ! PRESSURE RECOVERY

15 0.0         ! PRESSURE DROP

! PREMAS

16 0.095       ! BLEED AIR

17 0.0         ! FLOW LOSS

18 1.0         ! PRESSURE RECOVERY

19 0.0         ! PRESSURE DROP

! BURNER
```

20 0.075 ! FRACTIONAL PRESSURE LOSS DP/P

21 0.9995 ! COMBUSTION EFFICIENCY

22 -1.0 ! FUEL FLOW

! HP TURBINE

23 0.0 ! AUXILIARY WORK

24 -1.0 ! NDMF

25 0.6 ! NDSPEED CN

26 0.9 ! ISENTROPIC EFFICIENCY

27 -1.0 ! PCN

28 1.0 ! COMPRESSOR NUMBER

29 5.0 ! TURBINE MAP NUMBER

30 -1.0 ! POWER LOW INDEX

31 0.0 ! NGV ANGLE RELATIVE TO DP

! POWER TURBINE

32313320.0 ! AUXILIARY WORK

33 -1.0 ! NDMF

34 0.6 ! NDSPEED CN

35 0.9 ! ISENTROPIC EFFICIENCY

36 1.0 ! PCN

37 0.0 ! COMPRESSOR NUMBER

38 5.0 ! MAP NUMBER

39 -1.0 ! POWER LAW INDEX

40 -1.0 ! COMWORK

41 0.0 ! NGV ANGLE RELATIVE TO DP

! NOZCON

42 -1.0 ! THROAT AREA

```
! PERFOR
43 1.0        ! PROPELLER EFFICIENCY
44 0.0        ! SCALING INDEX
45 0.0        ! REQUIRED THRUST
-1
12 82.55      ! INLET MASS FLOW
6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE
-1
-1
6  6  1650          ! DT = 0 ; TET = 1650
-1
-1
6  6  1600          ! DT = 0 ; TET = 1600
-1
-1
6  6  1550          ! DT = 0 ; TET = 1550
-1
-1
6  6  1500          ! DT = 0 ; TET = 1500
-1
-1
6  6  1450          ! DT = 0 ; TET = 1450
-1
-1
6  6  1400          ! DT = 0 ; TET = 1400
-1
```

-1

6 6 1350   ! DT = 0 ; TET = 1350

-1

-1

6 6 1300   ! DT = 0 ; TET = 1300

-1

-1

6 6 1250   ! DT = 0 ; TET = 1250

-1

-1

6 6 1200   ! DT = 0 ; TET = 1200

-1

-1

6 6 1150   ! DT = 0 ; TET = 1150

-1

-1

6 6 1100   ! DT = 0 ; TET = 1100

-1

-1

6 6 1050   ! DT = 0 ; TET = 1050

-1

-1

6 6 1000   ! DT = 0 ; TET = 1000

-1

-3

A.4

```
! TURBOMATCH SCHEME - Windows NT version (October 1999)

 ! LIMITS: 100 Codewords, 800 Brick Data Items, 50 Station Vector

  !15 BD Items printable by any call of:-

  ! OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV

  ! GENERAL ELECTRIC LM2500 PLUS OFF DESIGN CALCULATION

  ! MODELLED BY ABASS KABIR, SOE, JANUARY 2013

! Turbo-match Programme: Ambient Temperature Influence simulation of GE LM 2500plus Gas Turbine

  ////

OD SI KE VA XP

-1

-1

INTAKE S1,2     D1-4            R100

COMPRE S2,3     D5-11           R101     V5        V6

PREMAS S3,12,4  D12-15

PREMAS S4,13,5  D16-19

BURNER S5,6     D20-22          R102

MIXEES S6,13,7

TURBIN S7,8     D23-30,101,31            V24

MIXEES S8,12,9

TURBIN S9,10    D32-41                   V32 V33

NOZCON S10,11,1 D41             R107

PERFOR  S1,0,0   D32,43-45,107,100,102,0,0,0,0,0,0,0,0,0

CODEND

DATA ITEMS////
```

```
! INTAKE
1 0.0          ! INTAKE ALTITUDE
2 0.0          ! ISA DEVIATION
3 0.0          ! MACH NO
4 0.9951       ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0         ! Z PARAMETER
6 -1.0         ! ROTATIONAL SPEED N
7 22.0         ! PRESSURE RATIO
8 0.8825       ! ISENTROPIC EFFICIENCY
9 0.0          ! ERROR SELECTION
10 3.0         ! MAP NUMBER
11 0.0         ! STATOR ANGLE RELATIVE TO DP
! PREMAS
12 0.049       ! BLEED AIR
13 0.0         ! FLOW LOSS
14 1.0         ! PRESSURE RECOVERY
15 0.0         ! PRESSURE DROP
! PREMAS
16 0.095       ! BLEED AIR
17 0.0         ! FLOW LOSS
18 1.0         ! PRESSURE RECOVERY
19 0.0         ! PRESSURE DROP
! BURNER
20 0.075       ! FRACTIONAL PRESSURE LOSS DP/P
21 0.9995        ! COMBUSTION EFFICIENCY
```

22 -1.0 ! FUEL FLOW

! HP TURBINE

23 0.0 ! AUXILIARY WORK

24 -1.0 ! NDMF

25 0.6 ! NDSPEED CN

26 0.9 ! ISENTROPIC EFFICIENCY

27 -1.0 ! PCN

28 1.0 ! COMPRESSOR NUMBER

29 5.0 ! TURBINE MAP NUMBER

30 -1.0 ! POWER LOW INDEX

31 0.0 ! NGV ANGLE RELATIVE TO DP

! POWER TURBINE

32313320.0 ! AUXILIARY WORK

33 -1.0 ! NDMF

34 0.6 ! NDSPEED CN

35 0.9 ! ISENTROPIC EFFICIENCY

36 1.0 ! PCN

37 0.0 ! COMPRESSOR NUMBER

38 5.0 ! MAP NUMBER

39 -1.0 ! POWER LAW INDEX

40 -1.0 ! COMWORK

41 0.0 ! NGV ANGLE RELATIVE TO DP

! NOZCON

42 -1.0 ! THROAT AREA

! PERFOR

43 1.0 ! PROPELLER EFFICIENCY

```
44 0.0        ! SCALING INDEX

45 0.0        ! REQUIRED THRUST

-1

1 2 82.55     ! INLET MASS FLOW

6 6 1580.0    ! COMBUSTION OUTLET TEMPERATURE

-1

-1

2   0.0              ! OD Calculation; DT = 0, ISA = 15 DEG C

-1

5  6  1650.0         ! DT = 0 ; TET = 1650

-1

-1

5  6  1600.0         ! DT = 0 ; TET = 1600

-1

-1

5  6  1550.0         ! DT = 0; TET = 1550

-1

-1

5  6  1500.0         ! DT = 0 ; TET = 1500

-1

-1

5  6  1450.0         ! DT = 0 ; TET = 1450

-1

-1

5  6  1400.0         ! DT = 0 ; TET = 1400

-1
```

```
-1
5  6  1350.0          ! DT = 0 ; TET = 1350
-1
-1
5  6  1300.0          ! DT = 0 ; TET = 1300
-1
-1
5  6  1250.0          ! DT = 0 ; TET = 1250
-1
-1
5  6  1200.0          ! DT = 0 ; TET = 1200
-1
-1
5  6  1150.0          ! DT = 0 ; TET = 1150
-1
-1
5  6  1100.0          ! DT = 0 ; TET = 1100
-1
-1
5  6  1050.0          ! DT = 0 ; TET = 1050
-1
-1
5  6  1000.0          ! DT = 0 ; TET = 1000
-1
2   -5.0          ! OD Calculation; DT = -5, ISA = 15 DEG C
-1
```

```
5  6  1650.0          ! DT = -5 ; TET = 1650

-1

-1

5  6  1600.0          ! DT = -5 ; TET = 1600

-1

-1

5  6  1550.0          ! DT = -5 ; TET = 1550

-1

-1

5  6  1500.0          ! DT = -5 ; TET = 1500

-1

-1

5  6  1450.0          ! DT = -5 ; TET = 1450

-1

-1

5  6  1400.0          ! DT = -5 ; TET = 1400

-1

-1

5  6  1350.0          ! DT = -5 ; TET = 1350

-1

-1

5  6  1300.0         ! DT = -5 ; TET = 1300

-1

-1

5  6 1250.0          ! DT = -5 ; TET = 1250

-1
```

```
-1
5  6 1200.0          ! DT = -5 ; TET = 1200
-1
-1
5  6 1150.0          ! DT = -5 ; TET = 1150
-1
-1
5  6 1100.0          ! DT = -5 ; TET = 1100
-1
-1
5  6 1050.0          ! DT = -5 ; TET = 1050
-1
-1
5  6 1000.0          ! DT = -5 ; TET = 1000
-1
2   -10.0             ! OD Calculation; DT = -10, ISA = 15 DEG C
-1
5  6  1650.0          ! DT = -10 ; TET = 1650
-1
-1
5  6  1600.0          ! DT = -10 ; TET = 1600
-1
-1
5  6  1550.0          ! DT = -10 ; TET = 1550
-1
-1
```

```
5  6  1500.0            ! DT = -10 ; TET = 1500

-1

-1

5  6  1450.0            ! DT = -10 ; TET = 1450

-1

-1

5  6  1400.0            ! DT = -10 ; TET = 1400

-1

-1

5  6  1350.0            ! DT = -10 ; TET = 1350

-1

-1

5  6  1300.0            ! DT = -10 ; TET = 1300

-1

-1

5  6 1250.0             ! DT = -10 ; TET = 1250

-1

-1

5  6 1200.0             ! DT = -10 ; TET = 1200

-1

-1

5  6 1150.0             ! DT = -10 ; TET = 1150

-1

-1

5  6 1100.0            ! DT = -10 ; TET = 1100

-1
```

```
-1
5  6 1050.0           ! DT = -10 ; TET = 1050
-1
-1
5  6 1000.0           ! DT = -10 ; TET = 1000
-1
2   5.0               ! OD Calculation; DT = 5, ISA = 15 DEG C
-1
5  6  1650.0          ! DT = 5 ; TET = 1650
-1
-1
5  6  1600.0          ! DT = 5 ; TET = 1600
-1
-1
5  6  1550.0          ! DT = 5 ; TET = 1550
-1
-1
5  6  1500.0          ! DT = 5 ; TET = 1500
-1
-1
5  6  1450.0          ! DT = 5 ; TET = 1450
-1
-1
5  6  1400.0          ! DT = 5 ; TET = 1400
-1
-1
```

```
5  6  1350.0          ! DT = 5 ; TET = 1350
-1
-1
5  6  1300.0          ! DT = 5 ; TET = 1300
-1
-1
5  6  1250.0          ! DT = 5 ; TET = 1250
-1
-1
5  6  1200.0          ! DT = 5 ; TET = 1200
-1
-1
5  6  1150.0          ! DT = 5 ; TET = 1150
-1
-1
5  6  1100.0          ! DT = 5 ; TET = 1100
-1
-1
5  6  1050.0          ! DT = 5 ; TET = 1050
-1
-1
5  6  1000.0           ! DT = 5 ; TET = 1000
-1
-3
```

A.5

************** TURBOMATCH WORKSHOPS **********

***************** EXERCISE 4 ***********

REF_TITLE: Performance simulation modelling of
HITACHI H25 ENGINE using the TURBOMATCH scheme

```
!Turbomatch Code: Simulation of HITACHI H25 INDUSTRIAL ENGINE
Ambient temperature influence = TAMB = BD(2)
////
OD SI KE VA FP
-1
-1
INTAKE  S1,2      D1-4          R100
COMPRE  S2,3      D5-11         R102    V5   V6
PREMAS  S3,4,15   D12-15
BURNER  S4,5      D16-18        R104
MIXEES  S5,15,6
TURBIN  S6,7      D19-26,102,27         V20
NOZCON  S7,8,1    D28           R110
PERFOR  S1,0,0    D19,30-32,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0            ! INTAKE DATA : ALTITUDE
2 0.0            ! DEV FROM STANDART TEMP
3 0.0            ! MA-NUMBER
4 -1.0           ! PRESSURE RECOVERY

5 0.85           ! COMP : Z
6 1.0            ! RELATIVE ROTATIONAL SPEED
7 14.7           ! PRESSURE RATIO
8 0.80           ! ISENTROPIC EFFICIENCY
9 0.0            ! ERROR SWITCH
10 3.0           ! MAP-NUMBER
11 0.0           ! STATOR ANGLE RELATIVE TO DP

12 0.85          ! Premas: Cooling bypass: LAMBDA W
13 0.0           ! DELTA W
14 1.0           ! LAMBDA P
15 0.0           ! DELTP

16 0.05          ! BURNER: PRESSURE LOSS
17 0.9995        ! COMB. EFF.
18 -1.0          ! FUEL FLOW

19 27500000.0    ! TURBINE DATA: AUXWORK
20 0.8           ! DESIGN NON DIM FLOW / MAX
21 0.6           ! DESIGN NON DIM SPEED
22 0.89          ! ISENTROPIC EFF
23 -1.0          ! ROT SPEED OF PT
24 1.0           ! NUMBER OF COMPRESSOR DRIVEN
25 3.0           ! MAP NUMBER
26 1000.0        ! POWER LAW INDEX; POWER OUTPUT = CONSTANT*PCN**3.0
27  0.0          ! NGV ANGLE RELATIVE TO DP
```

```
28 -1.0             ! FIXED CONVERGENT NOZZLE

29 -1.0             ! POWER OUTPUT
30 -1.0             ! PROPELLER EFF
31 0.0              ! SCALING SWITCH
32 0.0              ! REQUIRED THRUST at Design point

-1
1 2 88.0            ! item 2 at station 1 = Mass flow(kg/s)
5 6 1548.0            ! Item 6 at station 5 = TET(K)
-1
-1
5   6   1000.0  !  --New  OD  Calculation;  DT=TAMB(STANDARD)-TAMB(ACTUAL)=0;   TET  =
1000.0K
-1
-1
5  6  1100.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
5  6  1200.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
5  6  1300.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
5  6  1400.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
5  6  1500.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2    -5.0   ! --New OD Calculation; DT=-5.0; TET = 1500.0K
-1
5  6  1500.0        ! OD Calculation; DT=-5; TET = 1500.0K
-1
-1
5  6  1400.0        ! OD Calculation; DT=-5; TET = 1400.0K
-1
-1
5  6  1300.0        ! OD Calculation; DT=-5; TET = 1300.0K
-1
-1
5  6  1200.0        ! OD Calculation; DT=-5; TET = 1200.0K
-1
-1
5  6  1100.0        ! OD Calculation; DT=-5; TET = 1100.0K
-1
-1
5  6  1000.0        ! OD Calculation; DT=-5; TET = 1000.0K
-1
2  -10.0  ! --New OD Calculation; DT=-10; TET = 1000.0K
-1
5  6  1000.0        ! OD Calculation; DT=-10; TET = 1000.0K
-1
-1
5  6  1100.0        ! OD Calculation; DT=-10; TET = 1100.0K
-1
-1
```

```
5  6  1200.0         ! OD Calculation; DT=-10; TET = 1200.0K
-1
-1
5  6  1300.0         ! OD Calculation; DT=-10; TET = 1300.0K
-1
-1
5  6  1400.0         ! OD Calculation; DT=-10;TET = 1400.0K
-1
-1
5  6  1500.0         ! OD Calculation; DT=-10; TET = 1500.0K
-1
2  5.0   ! --New OD Calculation; DT=5;  TET = 1500.0K
-1
5  6  1500.0         ! OD Calculation; DT=5;  TET = 1500.0K
-1
-1
5  6  1400.0         ! OD Calculation; DT=5;  TET = 1400.0K
-1
-1
5  6  1300.0         ! OD Calculation; DT=5;  TET = 1300.0K
-1
-1
5  6  1200.0         ! OD Calculation; DT=5;  TET = 1200.0K
-1
-1
5  6  1100.0         ! OD Calculation; DT=5;  TET = 1100.0K
-1
-1
5  6  1000.0         ! OD Calculation; DT=5;  TET = 1000.0K
-1
2  10.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
5  6  1000.0         ! OD Calculation; DT=10;  TET = 1000.0K
-1
-1
5  6  1050.0         ! OD Calculation; DT=10;  TET = 1050.0K
-1
-1
5  6  1100.0         ! OD Calculation; DT=10;  TET = 1100.0K
-1
-1
5  6  1150.0         ! OD Calculation; DT=10;  TET = 1150.0K
-1
-1
5  6  1200.0         ! OD Calculation; DT=10;  TET = 1200.0K
-1
-1
5  6  1300.0         ! OD Calculation; DT=10;  TET = 1300.0K
-1
-1
5  6  1400.0         ! OD Calculation; DT=10;  TET = 1400.0K
-1
-1
5  6  1500.0         ! OD Calculation; DT=10;  TET = 1500.0K
-1
-3
```

A.6

************** TURBOMATCH WORKSHOPS **********
***************** EXERCISE 4 ***********

REF_TITLE: Performance simulation modelling of
HITACHI H25 ENGINE using the TURBOMATCH scheme

```
!Turbomatch Code: Simulation of HITACHI H25 ENGINE INDUSTRIAL ENGINE
Ambient temperature influence = TAMB = BD(2)
////
OD SI KE VA XP
-1
-1
INTAKE  S1,2      D1-4          R100
COMPRE  S2,3      D5-11          R102     V5
PREMAS  S3,4,15    D12-15
BURNER  S4,5      D16-18         R104     W5,6
MIXEES  S5,15,6
TURBIN  S6,7      D19-26,102,27          V20
NOZCON  S7,8,1    D28           R110
PERFOR  S1,0,0    D19,30-32,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0           ! INTAKE DATA : ALTITUDE
2 0.0           ! DEV FROM STANDART TEMP
3 0.0           ! MA-NUMBER
4 -1.0          ! PRESSURE RECOVERY

5 0.85          ! COMP : Z
6 1.0           ! RELATIVE ROTATIONAL SPEED
7 14.7          ! PRESSURE RATIO
8 0.80          ! ISENTROPIC EFFICIENCY
9 0.0           ! ERROR SWITCH
10 3.0          ! MAP-NUMBER
11 0.0          ! STATOR ANGLE RELATIVE TO DP

12 0.85         ! Premas: Cooling bypass: LAMBDA W
13 0.0          ! DELTA W
14 1.0          ! LAMBDA P
15 0.0          ! DELTP

16 0.05         ! BURNER: PRESSURE LOSS
17 0.9995       ! COMB. EFF.
18 -1.0         ! FUEL FLOW

19 27500000.0   ! TURBINE DATA: AUXWORK
20 0.8          ! DESIGN NON DIM FLOW / MAX
21 0.6          ! DESIGN NON DIM SPEED
22 0.89         ! ISENTROPIC EFF
23 -1.0         ! ROT SPEED OF PT
24 1.0          ! NUMBER OF COMPRESSOR DRIVEN
25 3.0          ! MAP NUMBER
```

```
26 1000.0           ! POWER LAW INDEX; POWER OUTPUT = CONSTANT*PCN**3.0
27  0.0          ! NGV ANGLE RELATIVE TO DP

28 -1.0          ! FIXED CONVERGENT NOZZLE

29 -1.0          ! POWER OUTPUT
30 -1.0          ! PROPELLER EFF
31 0.0           ! SCALING SWITCH
32 0.0           ! REQUIRED THRUST at Design point

-1
1 2 88.0          ! item 2 at station 1 = Mass flow(kg/s)
5 6 1548.0         ! Item 6 at station 5 = TET(K)

-1
6 0.9           ! PCN=90%
-1
-1
6  0.8          ! PCN=80%
-1
-1
6  0.7          ! PCN=70%
-1
-1
6  0.6          ! PCN=60%
-1
-1
6  0.5          ! PCN=50%
-1
-1
6  0.4          ! PCN=40%

-1
-3
```

A.7

```
!TURBOMATCH SCHEME DESIGN FILE
!ENGINE TYPE: HITACHI H25
////
OD SI KE VA XP
-1
-1
INTAKE  S1-2    D1-4                 R100
ARITHY          D310-317
ARITHY          D320-327
COMPRE  S2-3    D5-11                R101     V5 V6
PREMAS  S3,11,4  D12-15
BURNER  S4-5    D16-18               R102
MIXEES  S5,11,6
TURBIN  S6-7    D19-26,101,27                 V19 V20
NOZCON  S7-8,1   D28                 R103
PERFOR  S1,0,0   D19,29-31,103,100,102,0,0,0,0,0,0
```

```
CODEND

BRICK DATA////
! INTAKE
1   0.0     ! INTAKE ALTITUDE
2   0.0     ! DEVIATION FROM ISA CONDITIONS: Tamb=288.15K, Pamb=1.01325bar
3   0.0     ! MACH NUMBER
4   0.99    ! TOTAL PRESSURE RECOVERY

! COMPRESSOR
5   0.85    ! SURGE MARGIN                    [Z=[PR-PR(choke)]/[PR-PR(surge)]
6   1.0     ! DESIGN SPEED PCN                  [PCN=RPM/RPM(DP)]
7   14.7    ! DESIGN PRESSURE RATIO
8   0.80    ! DESIGN ISENTROPIC EFFICIENCY
9   0.0     ! ERROR SWITCH
10  3.0     ! COMPRESSOR MAP NUMBER
11  0.0     ! STATOR ANGLE RELATIVE TO DESIGN POINT

! PREMAS
12  0.150  ! COOLING BYPASS: LAMBDA W
13  0.0    ! DELTA W
14  1.0    ! LAMBDA P
15  0.0    ! DELTA P

! BURNER
16  0.05   ! TOTAL PRESSURE LOSS IN THE BURNER
17  0.9995   ! COMBUSTION EFFICIENCY
18 -1.0    ! FUEL FLOW                        [-1 = TET SPECIFIED. SEE SV DATA]

! TURBIN
19    27500000.0 ! DESIGN GENERATOR TURBINE FROM TURBINE DATA
20    0.8        ! DESIGN NON DIM MASS FLOW
21    0.6        ! DESIGN NON DIM SPEED
22    0.89        ! ISENTROPIC EFFICIENCY
23   -1.0        ! RELATIVE ROTATIONAL SPEED
24    1.0        ! NUMBER OF COMPRESSOR DRIVEN
25    3.0        ! TURBINE MAP NUMBER
26    1000.0      ! POWER INDEX
27    0.0        ! NGV ANGLE RELATIVE TO DESIGN POINT

! NOZCON
28    -1.0       ! FIXED AREA

! PERFOR
29    1.0        ! SHAFT EFFICIENCY
30    0.0        ! SCALING INDEX
31    0.0        ! REQUIRED DESIGN POWER

310 3.0        ! COMPRESSOR #1,BD(820)=BD(820)*BD(317)
311 -1.0
312 820.0      ! ETASF FOR COMPRESSOR#1
313 -1.0
314 820.0
315 -1.0
316 317.0
317 1.0        ! COMPRESSOR#1 DETERIORATION OF 0% IN EFFICIENCY

320 3.0        ! COMPRESSOR#1,BD(830)=BD(830)*BD(327)
```

```
321 -1.0
322 830.0      ! WASF FOR COMPRESSOR#1
323 -1.0
324 830.0
325 -1.0
326 327.0
327  1.0          ! COMPRESSOR#1 DETERIORATION OF 0% IN NON-DIMENSIONAL MASS
FLOW


-1
1 2 88.0        ! MASS FLOW (kg/s)
5 6 1548          ! TET (K)
-1
317 0.950      !5.0% DETERIORATION OF COMPRESSOR EFFICIENCY:COMPRESSOR 2
327 0.98      !2% DETERIORATION OF COMPRESSOR MASS FLOW:COMPRESSOR 2
-1
-1
```