Cranfield University

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The Optimisation of the Usage of Gas Turbine Generation Sets for Oil and Gas Production Using Genetic Algorithms

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Supervisor: Dr K W Ramsden

March 2010

Executive Summary

The privatisations of the energy supply industries world-wide has meant that emphasis is now on how to profitably compete. In this environment the development of effective models for optimisation of power plant are of increasing importance, particularly operational strategies for off-design conditions, and particularly for gas turbine engines.

Maximisation of plant profitability necessitates proper and integrated evaluation of many factors, the most important of which are: availability and price of fuel, system efficiency and performance, life cycle costing of plant and machinery, present and future generation of revenue, likely future market dynamics.

A major contribution of this work is the application of the proposed method to simultaneously maximise both total profit and usage availability of a typical combination of gas turbines engines used for power generation in oil and gas production. The method allows the user, for example, the opportunity to select locally appropriate daily and seasonal power demands and ambient conditions.

Through a genetic algorithm optimisation technique, an additional powerful feature of the method is that it allows the user to choose an optimised operating combination of their existing gas turbine equipment. Both individual engine power setting and number of engines can be varied. Alternatively, the user can apply the code to select the best combination of new and/or replacement equipment to achieve best economic performance and highest availability.

The number of variables involved in the optimisation process is, of course, very large. It is, therefore, difficult to find the optimal configuration. To address this problem, the first phase of this study is limited to the analysis of the performance of industrial gas turbine engines. The primary aim is to identify the key parameters in the determination of off-design performance. The second aim for the first phase is to identify those tasks suitable for automation. The Gas Turbine Library (Turbomatch) developed at Cranfield University includes simulation codes for many different industrial gas turbines and processes. The optimiser developed as part of this research has been linked with that library.

The second phase of this project is to develop an economic model for gas turbines analysing off-design performance. The model includes a life cycle cost assessment including: capital cost, maintenance and operating costs, fuel cost, emission and other taxes and disposal cost. By including total revenue it has been possible to develop a model that allows maximisation of total profit under variable operating conditions.

The third phase of the project presents an automated optimisation tool based on a listing of the Turbomatch simulation code and a genetic algorithm technique. The tool uses an evaluation of the fitness value of the objective function and takes into account the optimisation constraints. Two case studies considered where real data obtained from oil field in Libya are used to illustrate the use of the new code to maximising the profit.

ACKNOWLEDGEMENTS

My ultimate thanks to Allah who created me and gave me strength and knowledge. I feel lucky for being a student of Cranfield University and meeting very interesting people. It was wonderful experience, which has really broadened my mind.

I would like to express my greatest and sincere thanks to my supervisor Dr. K W Rmsden for his guidance and continuous support in every aspect, throughout the course of the investigation. Without his consultation and assistance, I would have hardly completed successfully this work.

Also, I would like to thank all of the staff at thermal power group and in particular Prof. P. Pilidis for their support and assistance.

Finally, I dedicated this work to my wife and my children Hanna, Ahmed, Ali and Omar who always stood by me and supporting me during this work.

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MENCLATURE

SYMBOLS

P	Total Pressure
p	Static Pressure
T	Total Temperature
t	Static Temperature
R	Gas Constant
W or m	Mass Flow
ρ	Density
Γ	Non- Dimensional Mass Flow
η	Efficiency
Cp	Specific Heat (at constant pressure)
Cv	Specific Heat (at constant volume)
γ = Cp/Cv	Specific Heat Ratio
C	Cost
F	Fuel
D	Number of Days

ABBREVIATIONS AND TERMS

Amb	Ambient
AMSL	Above Means See Level
ISA	International Standard Atmospheric Conditions
DP	Design Point
GE	General Electric
GPA	Gas Path Analysis
GT	Gas Turbine
GA	Genetic Algorithm
NDMF	Non- dimensional Mass Flow
OD	Off- Design
PR	Pressure Ratio
SCGT	Single Cycle Gas Turbine
SFC	Specific Fuel Consumption
SMR	Square Mean Root
TET	Turbine Entry Temperatur
IC	Intercooling
HX	Heat Exchanger
SC	Simple Cycle
CC	Combined Cycle
NPV	Net Present Value
TR	Total Revenue
SSS	Spark Spred Sheet
TP	Total Profit
LCC	Life Cycle Cost
C_{CAP}	Capital Cost
C_{FUEL}	Fuel Cost
LMP	Larson Miller Parameter
ADC	Accounting Depreciation Charge [£/Period]
LHV	Low Heating Value
EGT	Exhaust Gas Temperature
FAR	Fuel-to-air ratio
GT	Gas Turbine
MW	Mega Watts
V	Velocity
EGT	Exhaust Gas Temperature
HPT	High Pressure Turbine
O&M	Operations and Maintenance
NOx	Nitrogen Oxides
HR	Heat Rate
I/O	Input and Output
MWh	MegaWatt-Hour
CO	Carbon Dioxide
IT	Income Tax
BD	Book Depreciation
tr	Tax Rate

TI	Taxable Income
TR	Total Revenue
PC	Production Cost
DT	Depreciation Tax
AV	Ad Valorem Taxes and Insurance
INT	Interest Expenses
FCV	Fuel Calorific Value
ISO	International Organization of Standardization
NG	Natural Gas
C failure	Cost of Failure
$C_{FUEL}, (T)$	Cost of fuel During Time Period T
$C_{OM}\ (T)$	Operations and Maintenance cost
$C_{DEP}\ (T)$	Cost of Depreciation
$C_{EMI}\ (T)$	Emission tax during period time T
$C_{FIN}\ (T)$	Financial Cost
$HR(t)$	Heat rate of the Power Plant (MJ/MWh)
CDP	compressor delivery pressure
FOD	Foreign Object Damage
SPS	Shaft Power Speed
ISF	Index of Compressor Sensitivity to Fouling
TMR	Turbomatch Result File
DCFR	Discounted Cash Flow Rate of Return

1 INTRODUCTION

1.1 BACKGROUND

Continuous growth in electricity supply industry demand due to advances of science and technology has completely geared the industry into a more complex environment that no longer operates under the conventional philosophy of supply follows demand.

The non storage characteristics of electricity together with the increasing fuel costs worldwide further call for the need to operate the systems more economically without compromising on supply stability and reliability. In order to satisfy all these requirements, power systems short term scheduling is based on two key tasks, namely unit commitment and economic dispatch. This combinatorial optimisation task is aimed at meeting the load demand at each period of time in a least cost manner while satisfying both local and global constraints such as minimum up and down time, ramp rate limits, spinning reserve and other energy requirements.

Unit commitment is a system which, given a definite load profile, attempts to determine when the scheduled generating units in a system should be started and stopped to meet the load at all times, subject to the given operating constraints. The core of the unit commitment problem is that the total generation must equal the forecast two hour demands for electricity.

Economic dispatch is a method for determining the power outputs of each of the scheduled generating units at a given time so that the system meets its load requirement in the most economic manner.

Though the problem may look simple, it is typically extended in a number of ways due to the large variations in the operating conditions such as the power demand and ambient temperature between weekdays and between peak and off peak hours. As a result, utility companies need to decide carefully on these two tasks while satisfying other operational limits of generation such as ramp rate limits, uptime and downtime constraints as well as reserve and energy requirements. The problem becomes almost impossibly complex when one considers the huge number of generating units in the system, and the astronomical number of combinations and permutations of start-ups and shut-downs.

This study aims to use a genetic algorithm to develop an optimisation tool for the study of a portfolio optimisation simulation, and to present the analyses and results obtained. Maximum total profit is maintained by the selection of the best combination of feasible generating units, or where an existing power station has to be re-powering the plant requiring least cost is selected to provide the generating power required for the time it is required.

The gas turbine is an engine that works on the principle of the thermodynamic cycle described by the American George Brayton (1830-1892), though it is also known as the Joule cycle and was patented by an Englishmen John Barber in (1791). As with many other engines the purpose of the gas turbine is to convert heat of combustion into mechanical work. The gas turbine has three basic sections.

1. Compression; - ambient air is drawn in and compressed in a compressor. The process of compression heats the air.
2. Combustion; - the heated and compressed air is mixed with fuel and the mix ignited in an expansion or combustion chamber.
3. Expansion; the enthalpy of the combustion exhaust transformed into mechanical work by the turbine section. The expansion of the gases is used to move the turbine blades connected to rotors and engine shaft.

The gas turbine has many applications: it can act as the driving force for electrical generators, pumps, and mechanical equipment. In the aero-industry the exhaust gases are used to produce the thrust driving the plane.

Because industrial gas turbines can be found all over the world, operating in a wide range of industries and inhospitable environments it has been necessary to develop procedures for their selection and operation to ensure maximum cost effectiveness. Such procedures should, of course, maximise profit over the expected useful life of these engines.

This investigation focused on the economic performance of industrial gas turbine engines operating at off-design conditions.

To obtain maximum total profit from a given gas turbine plant, the selection procedure should include consideration of what is currently available on the market, including all those parameters which impact on the ability of the turbine to meet the specified operational strategies, energy demands and ambient conditions (these may well vary with time and will include the capacities of the gas turbines and necessary auxiliary equipment).

For complicated systems, of which the gas turbine power plant is one, it was traditional to place great emphasis on capital cost and reliability. However, in a deregulated market other considerations are also important: annual cost and online availability are of particular and important concern to the equipment owner.

Finding optimal solutions to real world design applications is usually an iterative process limited by the available resources. These limits may be the maximal number of possible computational and experimental design evaluations or the limited time of the design engineers. The development of optimal designs can be achieved by human designers or by exploiting automated optimisation techniques.

Human designers exploit their accumulated domain knowledge while automated optimisation algorithms search by analysing design evaluations resulting from a systematic parameter variation. For a large number of problems human design is the method of choice, however as the decision space becomes large and the associated processes more complex, the systematic procedures of automated optimisation become an interesting alternative. In addition, automated optimization is able to utilize most advantageously the opportunities offered by the increasingly massive and complex parallel computing and multicore architecture, and may result in alternative designs and eventually lead to new design philosophies.

In power generation field many parameters and aspects are interrelated with each other and there are a huge number of possible alternatives options; automated optimisation techniques offer a way to deal with such a problem and offer the possibility of optimum decisions.

Electricity cannot be stored cost-effectively, each day with the varying power demand it is necessary to start-up and shut-down power generating units. The problem for the companies generating electricity and those providing the power transmission systems are how best to meet the cyclically varying demand for electricity.

1.2 Study Rationale

According to the literature review the sum of the money invested in gas turbine power plants is invariably very large and mistakes during procurement or operation will not only be very costly, they could have serious long-term effects on the ability of the plant to generate adequate power in a timely manner.

In a deregulated market there is considerable pressure on power plant operators to develop the best and most advanced operational strategies to maximise profitability.

As an example to demonstrate the importance of this study, an oil field power plant producing about 136 MW/h, the seasonal production is 278,000 MWh. This value represents £8.9 million of revenue based on £32/MWh and the average total cost or the life cycle was about £8 million / season. Of this, about 80 % comes from the fuel consumption and 5% variable maintenance cost. The latter is directly related to the operational strategies. Accordingly for a 1% saving in the fuel cost by optimising the operational strategy for the fleet (which engine on and what is the power setting for each segment of time) will led to a saving of about £ 6 million over 25 years only (i.e. for the fuel at 1 % saving). Further to the fuel cost the maintenance and operating cost could be reduced significantly by considering the number of running units and the creep life for each. This can be achieved through careful control of power setting and the compressor fouling.

Against this background, new methodologies for gas turbine power plant optimisation are very desirable to enhance the correct choice of operational decision making while considering variable ambient and operating conditions. The outcome will then be to get closer to maximising plant profitability.

1.3 Objectives

The main objective of this work has been to develop an optimisation tool based on Genetic Algorithms and a gas turbine performance code (Turbomatch) which will allow an optimum selection and operation of a combination of gas turbines that leads to maximum total profit. Typical changes in both ambient conditions and power demand are taken into account.

1.4 Methodology

The methodology adopted for the research is as follows:

- To utilise a reliable gas turbine performance prediction program
- To establish the best combination of gas turbine power plants and their operation that leads to highest profit.
- To enhance the design point economic model for application to operation at off-design conditions.
- To establish a new model for the solution of the optimisation problem
- To link the model with the performance prediction model

1.5 Contribution

The major contribution of this study is to provide gas turbine users with the opportunity to compare the actual engine performance with a performance prediction program. This new program can predict, using a genetic algorithm (GA) optimiser, the combined engines operation that achieves least cost and highest availability.

Alternatively, the code provides the means by which the user can select the most cost effective engine combination when purchasing new or replacing existing engines so that the choice leads to least cost and best availability.

Indeed, the code presents, for the first time, an optimisation tool available for the user to suit a prescribed power demand profile while considering the ambient conditions.

1.6 Thesis Structure

As this thesis touches on a number of areas, it has been divided in such a manner as to afford maximum understanding of all elements of the study. To achieve this goal, this thesis has been divided into the following parts:

Chapter one gives a general introduction about this thesis, includes a general overview followed by the motivation for the study and, objective and methodology and then the contribution for this work.

Chapter two presents a literature review of the work undertaken previously in this field. It is focused on three key areas of interest, which were identified to be closely linked to the objectives of this work. These three areas are the performance and simulation for the industrial gas turbines at design point and off design conditions, gas turbine power plant economic and power generation optimisation techniques.

Chapter three discusses the performance of industrial gas turbine engines by introducing an operational model for the industrial gas turbine engine at design and off design conditions whine the power setting and ambient temperature are variable daily and seasonally and also investigate the behaviours of the engine at these variable parameters and also this chapter will present engines library for many different gas turbines by using Turbomatch simulating code.

Chapter four performs the economic analysis for each engine individually and for the whole power plant and this includes the total revenue, capital cost, maintenance and operating cost, fuel cost, emission tax and other financial costs.

Chapter five presents a brief description of genetic algorithm and its working mechanisms and the validation of the code used in the present study the secure GA171. It also provides a study of the parameterization of the GA variables to efficiently search a highly multimodal parameter space for a global maximum.

Chapter six presents the optimisation scheme and its implementation; this chapter also discusses the application of the new code to a variety of case studies

Chapter seven summarises the work performed and presents the recommendations for further future work in the areas studied.

2 LITERATURE REVIEW

2.1 Introduction

This Chapter summarises relevant literature to the project and, where appropriate, introduces a little theory. A short introduction is required to allow the reader to navigate the various threads.

This review has been divided into three main parts; the first part of the chapter presents a general background of the gas turbine and its behaviour at design and off design point and the common deterioration mechanisms that affect the engine performance.

The second part review the economic aspects and the life cycle cost for the stated problem considering the dynamic of the electricity market. The remaining part comments the different optimisation techniques and the selected optimisation algorithms for carrying out this study.

Finally the chapter concludes with the presentation of the rationale for the use of automated optimisation methods as part of common industrial design practice followed by a comprehensive survey of GAs, as a powerful tool for optimisation design containing GAs, methods, constraint handling procedures and GAs parameterisation techniques.

2.2 Industrial Gas Turbines Performance and Simulation

Over recent decades there has been a substantial increase in health and safety regulation and environmental legislation. This occurred simultaneously with pressure on fuel prices. The result is an increased demand more efficient generators producing fewer emissions.

One response has been the success of the design engineers in continuously increasing the efficiency and performance of gas turbine engines. In parallel engine operators were concerned with the operational efficiency and time periods between necessary servicing of the engine.

However, major issues that still need addressing is engine performance at off-design conditions and degradation of performance with increased service hours.

In practice performance degradation is mainly due to the day-to-day physical conditions in which the turbine is sited. That is component performance deteriorates due to fouling, which is a function not only of engine site location but of maintenance and operation methodology. These and other aspects of performance (and component) deterioration due to fouling, and also the consequences on engine performance, engine life and other economic issues, will be discussed in this and following chapters.

There are several ways of modelling and simulating gas turbines and diagnosing possible degradations and faults. Such modelling provides better information, which eventually helps with selection decisions and leads to improved operation after the gas turbine has been installed.

This part of work intended to review the literature on modelling and simulation work for the industrial gas turbines that has been carried out specifically for oil and gas applications.

2.2.1 Gas Turbine Performance Simulation

The possibility of estimating performance parameters and cycle details, for any operating conditions encountered in field operation is of fundamental importance to any technique of performance monitoring. Engine performance computer models are used for this purpose. Such models are based on a conceptual division of the engine into its components, according to the kind of thermodynamic process occurring in the combustor and the turbine.

A common feature of all available component-based computer simulation techniques is the requirement of component maps. The reliability of the predictions is highly dependent on the accuracy of these maps. Such maps are not easily obtained by the users as they are proprietary to the manufacturers. A number of techniques have been proposed by various authors to overcome this problem. They are mainly based on similarity considerations [1].

To build a performance model the gas turbine, or other machine, it is usual to view it as an assembly of separate individual components (also called modules). The individual components are identified according to the thermodynamic process they perform. Compatibility and matching of components imposes important conditions on the model, for example, temperatures at interfaces, speeds of the various components, power balance between turbines and compressors, etc.

To predict engine performance using a 1D model, it is enough to assume uniform flow, circumferentially and radially at any position along the engine. The working fluid is assumed to be a perfect gas with properties depending on temperature only. The values of the thermodynamic properties of the working fluid define the engine cycle at any operating point.

2.2.1.1 Design Point Performance

Central to the concept of the engine design process is design point performance. A given specification for a given engine configuration will determine the component performance levels and cycle parameters. However, before analysis of any other operating conditions is possible the design point performance must be defined [2]. In this chapter the performance input, which cannot be divorced from component design, is described.

The condition at which the engine spends most of its operating life is usually chosen as the engine design point. For industrial units this would normally be the ISO base load. However an alternative that is sometimes used is to choose an important high power condition. Whichever is chosen as the design point, the engine will be designed for optimum performance under those conditions. This project assumes that the component design points are for the same operating condition as the engine design point in the concept design phase.

Overall engine performance is defined by a number of key parameters which are used to assess how suitable a given engine design is for the given application. They may also be used to compare a number of possible alternative engine designs. These engine performance parameters include: Output power, specific power, exhaust gas power, specific fuel consumption, exhaust mass flow rate, exhaust temperature and thermal efficiency.

Brooks [3] has discussed cycle characteristics of several GE gas turbines including the thermodynamic principles of one and two shaft gas turbines, factors affecting performance and methods to enhance gas turbine output.

2.2.1.2 Off Design Point Performance

Any movement away from design point is normally referred to off-design performance. This movement will be due to internal or external alteration. Internal alteration is caused by component degradation while external alteration is caused by deviation of ambient conditions. Here geometry is fixed and operating conditions are changing. Influence of the ambient condition in the gas turbine performance from ISO like pressure, temperature and altitude. To see the effect of theses changes on engine performance ambient temperature, the following procedure has been deliberated [2]:

External:

- Influence of ambient temperature
- Influence of altitude
- Influence of power setting

Internal:

- Influence of compressor efficiency degradation
- Influence of compressor flow capacity degradation
- Influence of turbine degradation

The operating condition is changing even in the same location from day to night. The off-design operation could be one or many combination of the above. This will help to know the output and efficiency of turbine in each operating condition. The following combination has been adapted:

The Influence of Ambient Temperature and Site Elevation

Any changes to the mass flow rate of the air entering a gas turbine will change its performance, including air temperature, humidity and pressure all of which affect air density [4]. For example, with increase in elevation air temperature decreases but so does air pressure, and the net effect is that the density of the ambient air decreases with height above sea level.

The performance of gas turbine engines will thus depend on ambient conditions which will vary from day to day and from place to place [4]. For comparative purpose the

International Standards Organization (ISO) has established standard conditions which are universally accepted and used.

Gas turbine performance decrease in hot days due to the following consequences:

- Reduction in air density.
- Reduction in mass flow.
- Reduction in pressure ratio.
- Reduction in power output due high temp, low pressure and low mass flow.
- Increase in SFC due to high temp and power output decrease.
- Overall Efficiency decrease due to all of the above.

The pressure will decrease and the temperature will increase in all stations due to ambient temperature increase. And the opposite is correct for ambient temperature decrease.

A case study by Erdem and Sevilgen, [5] demonstrated the effect of ambient temperature on the efficiency and electric power output of a gas turbine. They showed that monthly temperature variations caused a loss of output from about 1.7% to 7.2%. It can be seen that higher ambient temperatures decreases electricity production. But such temperatures will also increase fuel consumption per unit of electricity produced. These researchers showed that at an ambient temperature of 10^0C, a power augmentation from about 0.4% to about 7.5% is obtained, and that fuel consumed per kWh is $0.308Nm^3$.

Gorji and Fouladi [6] in their study of ambient temperature effects on gas turbine power plant, stated that the with temperature increase from 5 ^{0}C to 35 ^{0}C (i.e. winter to summer), the amount of heat consumption decreases 8.5% and compressor work consumption increases 10.5% whereas the turbine work does not change so much (about 0.1%) and output work decreases by 13.8%. since the reduction of net work output is more than the reduction of fuel-air ratio, the results show that, by increasing ambient temperature from 5 ^{0}C to 35 ^{0}C, the amount of specific fuel consumption (SFC) will increase 5.7% and the cycle thermal efficiency decreases 5.9%.

The high quantity of air required by the engine in a typical compressor is 500kg of air for each unit of horse power produced by the engine [7]. Previous publications have demonstrated that optimal conditions of the air can produce high levels of power production from the gas turbine with low fuel consumption [8], [9].

The different applications of industrial gas turbine involve different conditions of the air due to atmospheric conditions [10]. For example, the increment of a centigrade degree from the ambient temperature can produce that the output power decreases 2%, see figure 2.1. [11]. The altitude also plays an important factor in the engine performance. The pressure in the inlet of the compressor is reduced in engines located at high altitudes, see figure 2.2.

In general, industrial gas turbines for power generation operate in low altitudes (below 800m AMSL) However, there are cases where these engines are in operation at high altitudes. For example this is the case of: mines facilities, gas & oil pump stations and power plants located at high altitudes [11], [12].

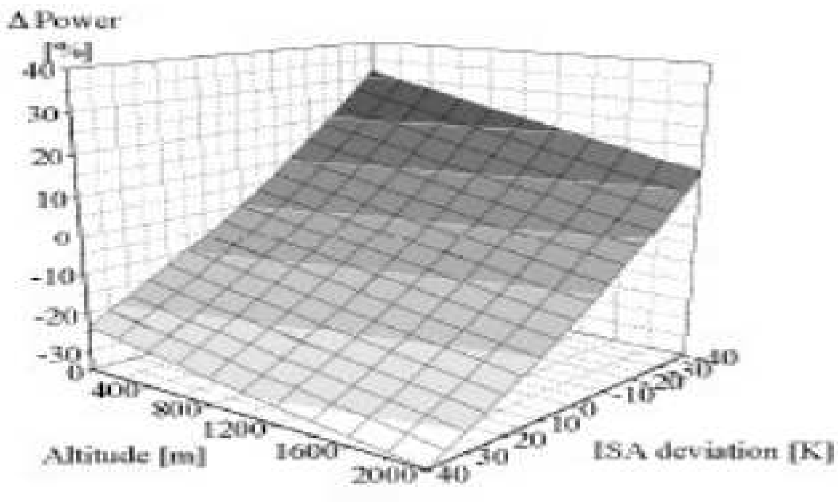


Figure 2-1 Output power of a gas turbine according to altitude and ambient temperature [11].

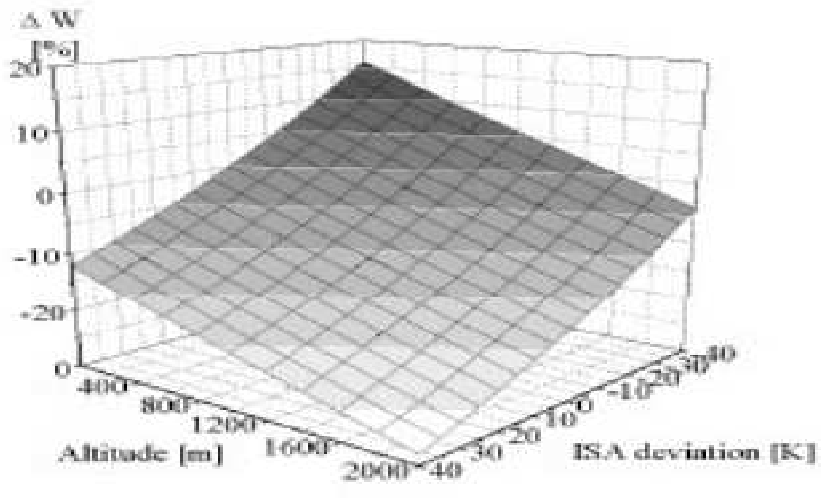


Figure 2-2 Mass flow of a gas turbine according to altitude and ambient temperature [11].

- **Influence of power setting**

The influence of ambient temperature on overall efficiency is affected by the power setting (TET). If the engine is operating on part load, the drop of efficiency will be much higher than if turbine is running on base load. But the power output is directly proportional to TET.

2.2.2 Industrial Gas Turbines Performance Deterioration

The deterioration of industrial gas turbines has been studied since 1914 when the ideal cycle of Brayton was modified to represent the real conditions. At the end of the forties was observed that the gas turbine deterioration affected the power production and increased the fuel consumption [13].

During the sixties the improvements in different sections of the gas turbine increased the engine efficiency by 17% with pressure ratio of 7:1 and temperatures of 815ºC [14]. The increment of the fuel price in the seventies obligated to the gas turbine users to study new ways to reduce the costs of operation. During this period the deterioration mechanisms that affect the engine performance were classified [15].

The boom of industrial gas turbines was in the eighties with the installation of Combined Cycles in power plants. The results of new technologies at the end of this decade increased the engine efficiency to 42% [7]. Today it is possible to find public documents published by OEM’s where the gas turbine operates with efficiency of 45%

and temperatures of 1371ºC [14]. However, these results are produced in special conditions (ISA conditions) and when the engine condition is new, because the gas turbine is a machine that presents a quick degradation [9].

Maintenance is intended to minimize risk of breakdown and improve reliability. It can also repair a system and/or upgrade performance. The longer a system runs, the more performance degradation increases and the probability of breakdown and interruption of production increases.

Preventive maintenance protects the system from unnecessary degradation, and helps restore system performance and reliability. For a gas turbine unit the degradation is largely due to the degradation of components and depends on the unit's history.

Diakunchak has pointed out that the harsh environment within a gas turbine means that all gas turbines experiences loss in performance with time [16]. No matter what care is taken to ensure good inlet filtration and clean fuel, the conditions inside the engine are such that flow path components will become corroded, damaged, eroded, fouled, etc, and engine performance will degrade and get progressively worse with increased operating time.

Diakunchak estimated that a simple cycle natural gas turbine engine operating 8000h a year with 46.5MW output, suffered a 3% annual decrease in power and an increase of 1% in heat rate. A degradation of performance that would cost $1.5 million dollars (US) over three years of operation [16].

Performance degradation is not only affected by the air, water and fuel which physically enter the turbine, it is also affected by starting cycle and power settings. In fact, the Brook [3] has pointed out that the majority of power lost in a gas turbine is because of compressor degradation, and that degradation of gas turbine performance could be classified as recoverable and non-recoverable.

Recoverable loss is the deterioration of performance that can be recovered by compressor cleaning (online and offline water wash). Losses associated with compressor fouling are often recoverable and can often be rectified by water washing or mechanically cleaning the compressor blades and vanes.

Performance degradation is not solely a function of the hours of operation, it also depends on the operating mode [16], which means that the mode of operation must considered when modelling performance degradation. Diakunchak [16] describes as important the presence of contaminants, of fouling, the efficacy of filters, the presence of corrosion and erosion, engine operation, damage repair and faulty maintenance practices.

The rate at which a turbine degrades will depend on how it has been operated, in particular during start-up when the engine experiences the most severe hot end thermal gradients. For a short time at before the control system regulates the fuel and air flows the combustor exit temperature exceeds that during normal operation. Thus, oxidation and corrosion is most severe at this time and add significantly to the engine aging process. The more often an engine is subject to start up, possibly due to emergency trips, the more rapidly it will degrade. Also if the turbine is operated for long periods at its peak rating this will degrade performance more rapidly than an engine operating at or below base load rating.

Before expensive remedial actions are taken, the extent of the performance degradation must be assessed and simple actions such as online and offline water washing considered. Economic considerations play an important role in determining the optimal frequency at which a gas turbine is, for example, water washed. Washing too frequently is a waste of resources, but if the process is left too long, the resulting loss of performance will cause a loss of revenue that outweighs the cost of washing. Remedial water washing is recommended by Diakunchak [16] whenever the mass flow rate falls by 2 to 3%.

2.2.2.1 Types of deterioration in gas turbines

The gas turbine components are affected by the wear over the lifetime of operation. These problems are presented in the blade aerodynamics and internal mechanical properties [17]. The literature review showed that there is limited information about the deterioration mechanisms in the gas turbines. This is due to the marketing used for the OEM’s in the public domain [11]. In addition, the high costs and difficulty that involves experimental tests have limited the number of investigations in this topic [18].

The deterioration mechanisms are classified by the gas turbine application and the type of damage caused in the engine [19]. The deterioration mechanism of industrial gas turbines can be classified according to the type of the damage in three sections [7], [13], [20].

- Recoverable damage involves light maintenance such as cleaning or washing.
- Major-Recoverable damage involves maintenance such as welding or coating process.
- Non-Recoverable damage requires the replacement of the part.

The non-recoverable damage is attributed when the engine is in operation with some part already damaged. This is also presented when the engine operates at lower efficiency and it produces excess of fuel consumption and increases the temperature in the turbine inlet that can produce internal damage in the components [21]. For that reason in the last decade, the development of technologies to monitor the engine health has given the opportunity to the engine users to solve the problem before the damage becomes non-recoverable. These preventive actions have demonstrated that reduces the maintenance cost and unexpected shut downs [13].

The common deterioration mechanisms presented in gas turbines are divided into six categories [17].

- Fouling (is caused by particle deposition on the airfoils and annulus surfaces. Deposition is due to the inertia forces acting on the particles.)
- Corrosion and hot-corrosion (it is caused by chemical reactions between the contaminant and the component material).
- High-temperature-oxidation (it is caused by chemical reactions between metal components and oxygen).
- Erosion (it is caused by the result of abrasive components that removes the component material from the surfaces.

- Foreign Object Damage (FOD) (it is caused by the ingestion of large objects into the flow path. They are the results of internal pieces broken or ice formation in the inlet).

- Abrasion, rubbing and wearing (it is caused by the contact between two surfaces in movement, generally one in rotation and other static).

These problems are difficult to detect when the engine is in operation. In some cases, the effects of the engine degradation can be detected when the engine decreases the output power or increases the fuel consumption [17]. However, these parameters do not give enough evidence to find the source of the problem, because this is due to changing from the operation conditions [22].

2.2.2.2 Compressor degradation

Compressor fouling is produced due to the ingestion of dust mixed with the air. This mechanism decreases the compressor isentropic efficiency. Fouling and erosion have been demonstrated to affect the thermal efficiency and output power of the engine [23]. The deposition of the particles in critical areas can change the geometry of the airfoils and then the flow condition is modified [24]. In addition, the accumulation of dust reduces the tip clearance and increases the roughness of the surface roughness [17]. These changes in the blades affect the compressor delivery pressure (CDP) and reduce the mass flow.

Howell and Calvert [25] & Aker and Saravanamuttoo [26] calculated the impact of degradation mechanisms in the compressor performance based on the output power of the engine. Gulen, Griffin, and Paolucci [27] reported that fouling decreased by 5% the output power due to mass flow reduction. Figure 2.3, shows a typical performance degradation due to compressor fouling [28]

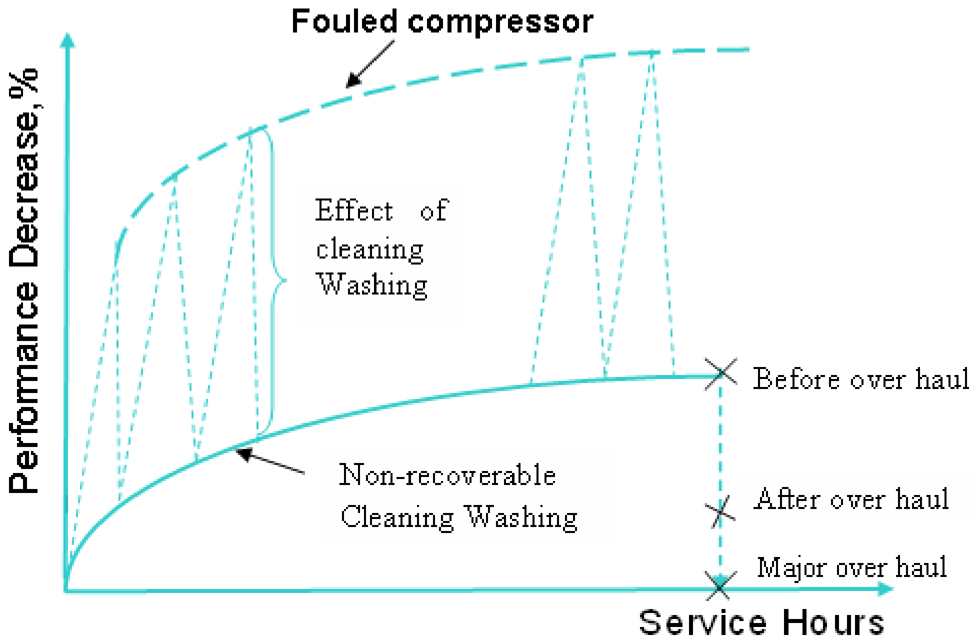


Figure 2-3 Show typical Performance degradation due to fouling [1]

2.2.2.3 Combustion chamber degradation

The combustion chamber is one of the sections with a low level of degradation. The operation time of the combustion chamber has an irrelevant effect to degrade this section [29]. However, small variations in the combustion process such as increment of the fuel ratio can affect the components from the turbine section.

For that reason, it is necessary to control the process of combustion. In addition, an incorrect combustion can produce ash that is deposited in the fuel injectors. This problem produces fluctuations in the flame and high local temperatures. The alteration in the temperature profiles increases the possibility of secondary flows, reduces the turbine efficiency and damages the turbine blades [17].

2.2.2.4 Turbine degradation

Erosion is a common problem present in this section that modifies the profile of the blades. Fouling is also present in the turbine blades due to ash adhesion on the surfaces [13].

It has been demonstrated that fouling on the turbine blades reduces the turbine isentropic efficiency by 1% and the output by 3.7% [29]. As it was mentioned in the previous section, the high temperatures and incomplete fuel burning affect the turbine section due to overheating. This effect is responsible for hot corrosion that modifies the shape of the leading edge from the blades [17].

2.2.2.5 Engine performance monitoring

Due to the fuel prices crisis in the last decades, many gas turbine users have introduced the use of technologies for monitoring the engine performance. The data obtained from engine monitoring have estimated the deterioration of the gas turbine.

To determine when an engine is operating inefficiently, G.F Aker and Saravanamuttoo [26] mentioned that the Engine Performance Monitoring has to be involved closely. Most important gas path parameters, such as compressor delivery pressure (CDP) and turbine entry temperature (TET), have to be monitored in order to determine the deviations of these parameters from their baseline value specified by engine manufacturer. The deviations from these baselines are then plotted as a function of time to form "trendplots." Significant deviation from the baseline on these trendplots is used as a trigger for maintenance action.

In order to be effective, such of trend analysis monitoring system must be accurate enough. Although engine measurement system provides the information of engine gas path performance parameters, there are many issues to be dealt with before engine health is predicted and maintenance schedules can be suggested. For example, the engine measurements vary with engine operating conditions and ambient condition. To make correct engine health analysis, the measured performance may have to be corrected with the help of accurate engine performance model. Otherwise, the results produced with the engine model may be misleading [30]

This information is used to schedule the preventive maintenance and to extend the period of the optimal production [15], Haq and Saravanamuttoo [31]. For example Pinelli reported a gas turbine model AVIO TG20 that reduced in the first month 4.1% of the power production due to degradation. Similar result was reported by Gulen, Griffin, and Paolucci [27] where two single shaft gas turbines in operation in CCGT reduced 5% the output power in the first month.

Mund [11] mentioned that the single shaft configuration has a high sensitivity in the output power when the aerodynamics of the flow changes. For that reason, it is necessary in power plant applications the periodic maintenance in the engine, Zwebek [7].

The typical overall maintenance period for industrial engines is approximately between 3000 to 4000 hours of operation. At the end of this period it is possible to find that the engine has lost 20% of the production capacity [13].

2.2.3 Compressor Fouling

The gas turbine operators have identified the presence of fouling in the engine since 1936 [32]. However, since the sixties the problem of fouling in industrial gas turbines has been considered an important cause to degrade the compressors performance and to be an evil inherent product of the operation [33] & [34]

The Middle East crisis in the seventies produced that the fuel prices increased 70%. The high cost of fuel produced the sufficient incentive to gas turbine operators to look for new technologies to abate the mechanism of degradation to obtain optimal efficiencies for long periods [33].

The studies in this area have demonstrated that the fouling mechanism represents approximately 80% of the compressor losses. A reduction of 5% from the mass flow can reduce 13% the output power and increase 5.5% the heat rate [35] & [36]. These losses represent millions of USD in power and fuel consumption [29] & [35].

Fouling is found in all engines due to the big quantities of air ingested. A typical gas turbine in operation in a residential location ingests 1.5 kg of solid contaminants per day [32]. For example, an engine of 7.5MW in an environment with particles concentration of 1 ppm can ingest 5 kg of dust in a single day [19].

This problem could turn worse if the engine operates in a much polluted environment such as mining or oil field areas where the ingestion of foreign particles can rise up to 39kg per day [35]. It is necessary to pay careful attention to the inlet of the engine, because the ingestion of high quantities of particles reduces also the engine life [37].

Sedigh, E, and Saravanamuttoo [38], indicated that a 1 % reduction in axial compressor efficiency can account for a 1.5% increase in heat rate for a given power output. If fouling continues, eventually compressor surge may take place. Surge can severely damage a compressor and in the worst case irreparably damage an entire gas turbine engine.

In general, the industrial gas turbines are installed with inlet filters that stop the pass of particles [39]. The size of particles stopped by the filters can be in the range of μ m [40].

2.2.4 The Mechanism of Fouling

The flow in an axial flow compressor is a complex and three-dimensional phenomenon. As suggested by Mattingly [41], to help the understanding of the basic phenomena the flow field can be regarded as two-dimensional which is less complicated than three-dimensional flow.

The mechanism of entrainment of particles by a surface of a body situated in the stream of the air-aerosol mixture was described by Fuks, [42]. It is also, A. P. Tarabrin et al, [39] noted that the deposition of particles on the surface of a blade takes place under the action of inertia forces acting on the particles and forcing them to move across the curved stream lines (the particle trajectory deviates from the stream lines). Particles of dust colliding with the blade can stick to the blade surface.

On the other hand, when the particles move in the flow path of compressor the centrifugal inertia forces make them move to the periphery of the compressor passage.

The coefficient of entrainment or the separation factor E according to Fuks, [42], is determined as a ratio:

$$E = \frac{h}{L} \qquad [2.1]$$

Where:

h – Number of particles colliding with the surface of the body

L – Number of particles which could fall on the body surface if the stream lines were not deviated by the body

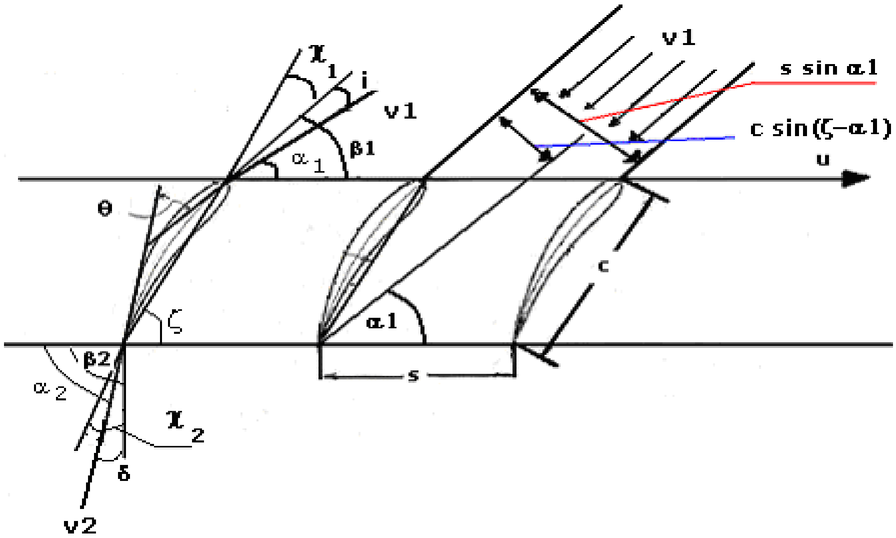


Figure 2-4 Cascade of profiles of axial compressor [39]

Then

$$\mathbf{E}_C = \frac{h}{s \sin \alpha_1} = \frac{h}{c \sin(\varsigma - \alpha_1)} \times \frac{c}{s} \times \frac{\sin(\varsigma - \alpha_1)}{\sin \alpha_1} \qquad [2.2]$$

As the pressure ratio of a compressor stage increases the flow turning angle and (c/s) of the cascade also increases. Therefore, the high-head stage is more sensitive to fouling then the low-head one.

As Trabin [39] suggested, the inertial deposition of the particles takes position on the windward side of the blade facing the flow. The deposition of the particles is also present on the leeward side of the blade profile as a result of the flow swirls and turbulence. The results of an investigation of the deposit formation on gas turbine compressor blading performed by Olhovsky [43] have shown that the fouling affects the first 5-6 compressor stages. The blade fouling becomes less severe in the later stages of the compressor. Figures 2.5 and 2.6 shows that fouling is present on both the concave and the convex sides, but the convex side of the rotor and stator blades, except the IGV, is more affected by the deposit formations, than concave one [39].

Figure 2-5 Compressor rotor bales fouling (Sulzer type 3, Nafoora oil field)

Figure 2-6The hydrocarbon deposit in compressor IGV and blades (Sulzer type 3, Nafoora oil field)

2.2.5 Fouling contaminant source

The atmospheric air ingested by the gas turbine contains certain amounts of contaminants. These contaminants are the product of soft aerosols formed by small particles of dirt, dust, pollen, insects, oil vapour, sea water salt, water vapour, sticky industrial chemicals, un-burnt hydrocarbons, soot particles, etc. Brooks [3] &, Upton [34]

The performance deterioration of the compressor is due to these particles that can cause in the blades a temporary problem (fouling) or a permanent problem (erosion) [44]. The main source of the fouling problem in the compressor has the origin from the particles mixed with the atmospheric air. The fouling layer on the blade surface is formed with 80% of the dust that is found in the filters [29].

The layer of fouling presents in many cases a combination of contaminants with residues of oil or water mist [17]. The concentration of particles increases under unfavourable conditions such as sand storms or chemical polluted clouds and the mechanism of fouling is accelerated [35]. For that reason, it is important to consider the environment conditions, layout of the plant and maintenance schedule in order to reduce the probability of compressor fouling [45].

The particles that are deposited on the compressor blade surface are in the range of micrometers [17], [13] & [37]. The x-ray analysis for a fouling sample demonstrated that the layer of fouling is a mix of different components, see figure 2.7 [46]

Two groups of components were identified from this analysis. The first group was water-insoluble solids represented by the presence of silicon, and the organic materials were represented by the presence of carbon and oxygen. The second group was water soluble substances that cause corrosion. They were hydroscopic and contained chlorides to promote corrosion.

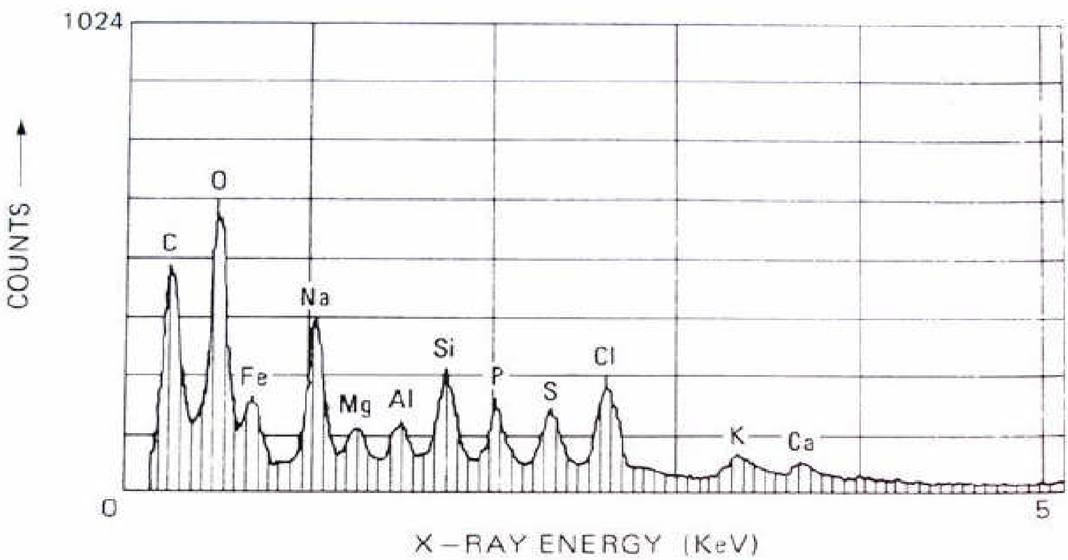


Figure 2-7 EDX spectrum of layer deposit on the surface of compressor blade [110].

2.2.5.1 Sources of external contaminants

Table 2.1 proposes a general classification from the contaminants according to geographical location of the engine [10]. For marine applications, it has been demonstrated that salt is the main source of compressor fouling [10], [47] & [48].

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

Table 2-1 Gas turbine location and typical contaminants [10]

New technologies of filtration have demonstrated to stop the particles from the air stream. However, their effectiveness is random due to changes from the environment conditions. The common particles retained from the filters are summarized in Table 2.2.

These particles are found mixed between them and seldom were found isolated in the filters [49] and [50].

TYPE	TYPE OF PARTICLE	SIZE (μM)
F1	Sand	20 ~ 2,000
F2	Ground-Dust	1 ~ 300
F3	Oil Smokes (oil & gas plants)	0.02 ~ 1
F4	Fly Ash	1 ~ 200
F5	Salt Particles in Mist	Less than 10
F6	Salt Particles on Spray	More than 10
F7	Insects Swarms	More than1,000
F8	Smog*	Less than 2
F9	Clouds & Fog	2 ~ 60
F10	Rain	More than 60
F11	Fume*	Less than 1
F12	Clay	Less than 2
F13	Rosin smoke	0.01 ~ 1
F14	Fertilizer	10 ~ 1,000
F15	Coal Dust	1 ~ 100
F16	Metallurgical Dusts and Fumes (welding smoke)	0.001 ~ 100
F17	Ammonium	0.1 ~ 3
F18	Cement Dust	3 ~ 100
F19	Carbon Black	0.01 ~ 0.3
F20	Contact Sulphuric Mist	0.3 ~ 3
F21	Pulverized Coal	3 ~ 600
F22	Paint Pigments	0.1 ~ 5
F23	Plant Spores	10 ~ 30
F24	Pollens	10 ~ 100
F25	Snow & Hail	More than1x10^4

Table 2-2 Common particle size and concentration in atmospheric air [49] and [50]

The ambient conditions such as temperature, pressure and humidity play an important role in the engine performance and they have to be considered in the study of fouling. The ambient temperature is considered into three ranges: Hot (50 to 30ºC), Warm (29 to 15ºC) and Cold (15 to -20ºC).

The humidity is considered into three ranges: Dry (less 10%), Medium (around 50%), and Wet (more than 75%) [51]. The combination of the possible environmental scenarios between the temperature and humidity result in nine cases, see table 2-3.

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

Table 2-3 General environment scenarios of industrial gas turbines in operation

The type of fouling can change due to the season time in the same location. This is influenced by the ambient temperature and the concentration of particles. For example, the ambient temperature in the northern hemisphere can change dramatically between one season and another. The altitude is an ambient parameter that also modifies the engine performance. However, the influence of this parameter in the mechanism of fouling is nil.

2.2.5.2 Sources of internal contaminants

The internal causes that produce compressor fouling are due to non-maintenance or incorrect operation from auxiliary equipments [52]. For example, the presence of oil residues due to lakes seals is commonly found in the compressor. The cooling system by fog in the inlet of the engine can carry salt or water impurities into the compressor [24] & [13]. The filter panels are affected by erosion and corrosion and this increases the presence of FOD inside of the engine [13]. In a sample of fouling the presence of steel and aluminium particles was found due to the components rubbings from the brush seals and bearings [53]. The wear of graphite bushing from the compressor guide vanes has been demonstrated to be also a source of compressor fouling [54].

2.2.5.3 Steam and vapours as source of fouling

The presence of oil and vapours inside of the compressor increases the adherence of particles on the blade surface [55]. The oil vapours are produced by oil lakes from internal components of the engine [24]. The deposition of oil and particles on the rear compressor blades formed a hard layer due to the temperature.

This layer is only possible to remove by hand in the overall engine maintenance [56]. The chemical vapours mixed in the air are the product of polluted environments. For example, in marine applications the diesel vapour is produced by the auxiliary engines localized close to the engine inlet [56]. The natural ambient agents accelerate the adhesion process as such as heavy fog, rain and excessive humidity [57].

2.2.6 Effects of Fouling on Engine Performance

When individual components degrade they often do so at different rates which could result in components mismatch, and will lead to losses in overall gas turbine performance. Generally the effects of degradation due to fouling are:

1- Compressor component deterioration leads to performance and efficiency reduction

2- Power output reduction caused by deterioration of the ingested air mass flow

3- Reduction in overall thermal efficiency

4- Where constant power is required, degradation leads to increased fuel consumption and reduction in turbine blade creep life – due to higher turbine entry temperatures.

5- Due to all the above, economic losses will be incurred: additional fuel, repairs and more frequent replacement of components (especially the expensive hot section) and the necessity of employing additional maintenance technicians.

6- There will be loss of production because of the reduction in power supply.

7- With heavy fouling the likelihood of compressor surge and subsequent engine damage is increased.

8- Unbalanced components due to corrosion and/or fouling could lead to unstable and critical engine operation and even vibration breakup.

9- The complete or partial blockage caused by even fine particles entering the cooling passages of the turbine can cause hot section overheating.

Zwebek [13] and Seddigh and Saravanamuttoo [38] reported that in real applications 1% of reduction in the compressor efficiency increased the heat rate by 1.5% to produce the same output power, see figure 2.8.

In the case of the compressor configuration, fouling has a higher impact in axial compressors than in centrifugal compressors [40] & [38]

The map of the axial compressor shows the reduction of the surge margin due to fouling [29] & [38]. The speed of the engine also could hide the fouling effect in gas turbines. This is because the movement from the design point represented in the compressor map could be easy confused with a different compressor line of operation [38]. The case is illustrated when the design point of the compressor is located in a higher shaft speed point and lower efficiency.

When compressor fouling occurs, the compressor efficiency increases due to the reduction of the shaft speed and then effect of fouling is difficult to detect [17].

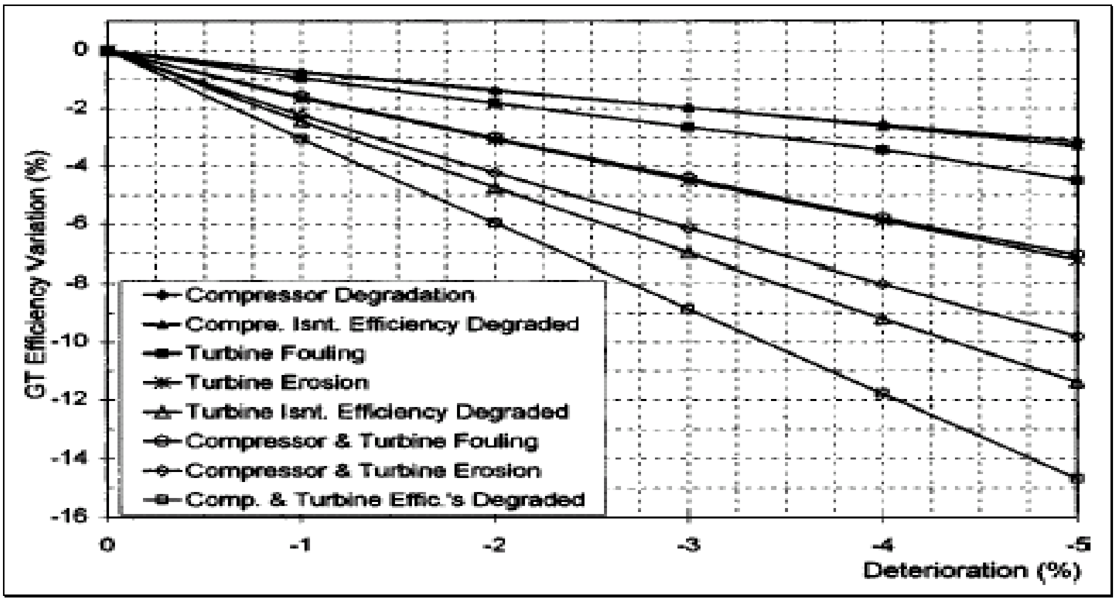


Figure 2-8 Gas turbine efficiency based on deterioration in specific sections [13]

The importance of the engine configuration to evaluate the influence of fouling was analyzed by Caguiat, Zipkin, and Patterson [48]. The first case is an engine configuration of two shafts (GE LM2500), figure 2.9. The normal operation condition was linked to a specific value of the compressor delivery pressure (CDP) and the gas generator turbine (GGT) was linked to a specific value of the shaft power speed (SPS).

When the CDP was reduced due to fouling, the turbine entry pressure (TEP) was also reduced. This affected the GGT and then the SPS was also reduced. The automatic control had to increase the fuel consumption to increase the power from the GGT and to obtain the correct SPS. In this case, it is possible to detect the problem in the compressor based on the speed and fuel consumption of the engine.

The second case was the operation of a single shaft engine (Allison 501K), see figure 2.10. The CDP was decreased due to fouling and then the SPS also decreased. The automatic control increased the SFC and hence the turbine entry temperature (TET) increased to obtain the correct SHP. In this case, it is difficult to detect the problem directly because it requires to check the engine performance with two different loads. However, in the real application, it is not possible to change the load.

The detection of fouling in a multiple axial compressor is also a difficult, because the early stage modifies the rear stages. The analysis in this case is a complex study based on the blade aerodynamics to detect the stage affected [58]. For that reason detecting fouling inside of the compressor is a difficult work and is generally based on the operation experience [11], [35] & [59].

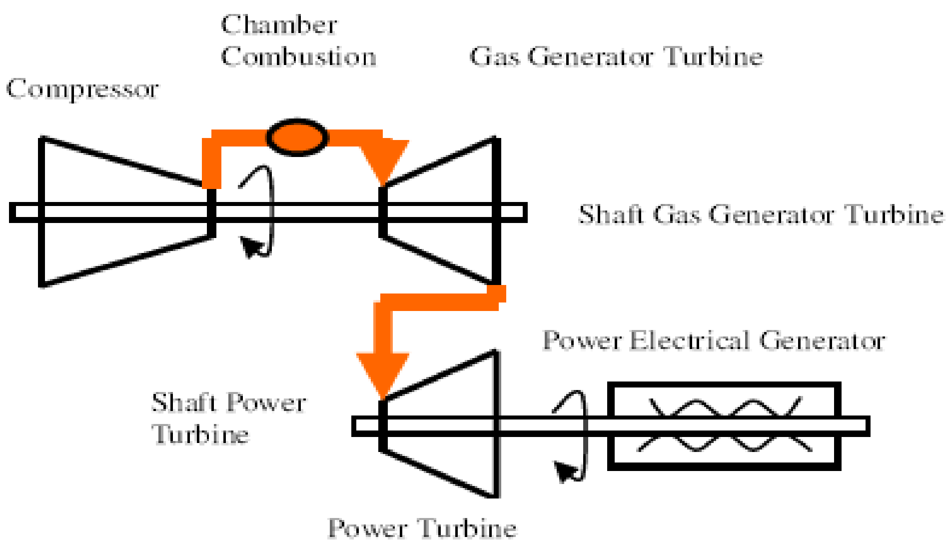


Figure 2-9 Gas turbine two-shaft configuration

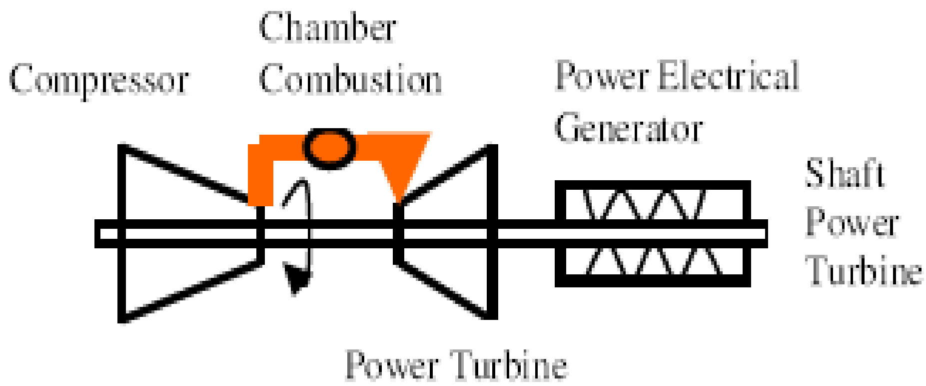


Figure 2-10 Gas turbine single-shaft configuration

2.2.6.1 The effect of Fouling on air distortion and compressor surge

Reduction of the inlet mass flow rate is a major consequence of compressor fouling, particularly if it occurs at the first stage because the first stage is not operating at its design point and this will have follow on effects and problems will accumulate.

This change in the air flow condition will decrease the efficiency and the stall region margin from the compressor [60], [53] & [58]. This reduction is due to the increment of the surface roughness that changes the boundary layer thickness and reduces the aerodynamic properties from the blade [17].

This change is represented in a compressor map when the operation point is moving close to the surge line [23], [60], [43] & [40]. According to Mustafa [35] the condition of operation close to the surge line is a dangerous situation for the compressor.

2.2.6.2 The effect of compressor fouling on blade integrity

Blade failure is a direct consequence of such mechanisms such as erosion, corrosion and FODs. Fouling will have a direct effect is on component performance, but it is not a main reason for blade failure. The direct effect of fouling is rotating stall and subsequent compressor surge, which can affect blade integrity.

Heavy fouling on moving surfaces will often cause imbalance and vibration problems. Where there is excessive humidity which allows fouling layers to adhere to each other and build up, corrosive contaminants such as salts, acids and aggressive gases such as NOx and SOx, can become trapped and this will lead to corrosion and pitting of the blades. This can lead to stress concentrations and cause reduction in blade fatigue life.

Also dirt and fine particles can get in through the small gaps and/or clearances such as occur in bearing and seals, these can build up and result in unstable operation of the disc and blades.

2.2.6.3 Effect of compressor fouling on emission

Today with current concern about environmental pollution the emissions from gas turbines are a serious consideration and a major factor to be considered in plant maintenance.

Increased turbine entry temperatures, due to fouling will cause higher turbine section temperatures. A major consequence is an increase in the quantity of NOx generated at higher firing temperature in order to maintain power. However if increase of turbine temperature is limited by the manufacturer there will be a loss of power.

2.2.6.4 Economic consequences of fouling

The costs associated with fouling are significant in several area:

- **Daily Operation Losses**

These losses will include the additional of fuel consumed to maintain the power level. Where increase of turbine entry temperature is prevented or limited by a control system installed during manufacture there will be a loss of power with consequent production losses; this could be of major importance in the power generation and oil and gas industries.

- **Maintenance and Repair Costs**

The increase in turbine inlet temperature necessary to maintain power lost will significantly reduce turbine blade creep life. Also fouling can block or partly block the cooling passages causing overheating problems in the turbine section and increasing turbine blades temperature. An increase of turbine entry temperature by 22°K has been estimated as being responsible for a decrease in the high pressure turbine blade by a factor of 3.2 or 69% [61]. The early and repeated replacement of expensive hot section components would cause huge economic costs. Fouling can also result in corrosion problems in the compressor and the early replacement of compressor components. In section 2.2.2 it was mentioned that an increase in heat rate by 1% and a reduction in power output by 3% will cause an increase in the operating costs of a 46.5 MW engine of about $1.5 million dollars (US) over of 3 years [16].

One estimate of the total losses incurred due to fouling for two engines, one of 26MW and the other 225MW, each working 8000h a year for two engines, were $500,000 (US) and $5000,000 (US) respectively. [29]. However the costs incurred due to fouling could be substantially reduced at relatively small cost by employing suitable and well-designed compressor cleaning operation. That is an operator who uses the correct washing system including elements such as the washing skid, the correct nozzle and injection system design, the correct choice of cleaning detergents, washing frequency and the correct washing procedure [62].

2.2.7 Sensitivity of Engine Size to the Fouling

There are many publications, for example, by Aker and Saravanamuttoo [26], Sedigh and Saravanamuttoo [38], Tarabrin [39] and many others, describe the sensitivity of the engine to the fouling. Most of them, however, confirm that an engine of smaller size shows more deterioration due to fouling than that a larger one.

It is very useful to develop effective methods to predict the degradation of engine performance in the presence of fouling. Some mathematical simulations of fouling in multistage compressors have recently appeared.

As mentioned above, field operating data revealed that the first stage of a compressor fouls the most, constituting 40-50% of the total fouling effect. In addition, the degree of fouling diminishes from front to the rear stages. A stage stacking technique introduced by Howeli and Calvert [63] used a mathematical model to calculate the compressor map through the performance of every stage.

In order to determine the sensitivity of a compressor to fouling, Sedigh and Saravanamuttoo [38] suggested that the following index can be used to compare compressor stage performance degradation due to fouling. The non-dimensional index is the ratio of the specific power output of the engine and the enthalpy rise for a stage.

$$\frac{P^0}{m^0 C_P \Delta T_{stg}} \qquad [2.3]$$

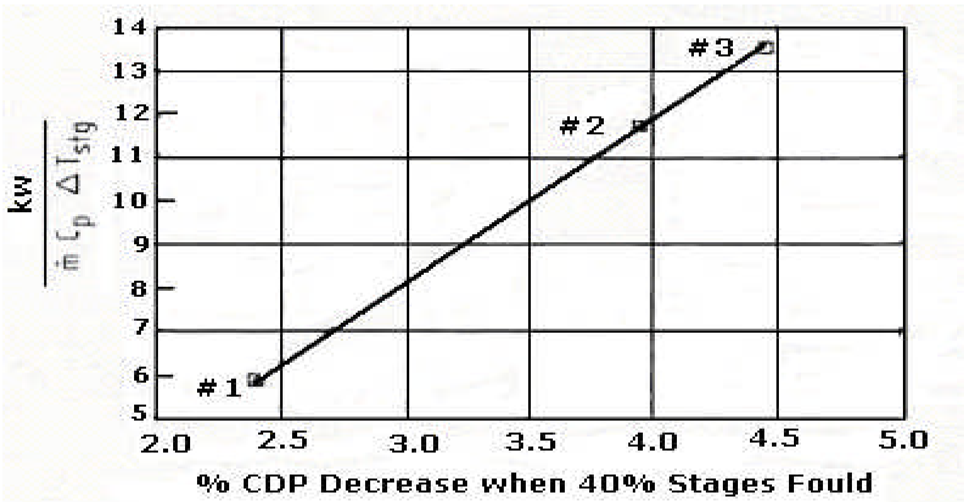


Figure 2-11Variations of the index with percentage CDP decrease for the three engines when 40 percent of stages fouled [38]

Figure 2.11, shows the variation of this index against the percentage decrease in CDP for three different size engines, due to 40 percent fouling of the stages. The variation of this index with the sensitivity of an engine to fouling is linear and consistent. This index incorporates important specification data for an engine, i.e., the engine size manifested by its design power output, its design mass flow rate and its stage loading. [38].

Tarabrin [39] has established another index of compressor sensitivity to fouling (ISF), and suggested that, the following equation will reflect qualitatively the sensitivity of the axial compressor to fouling under similar environmental conditions.

$$\mathbf{ISF} = \frac{\mathbf{m^{\circ} c_P \Delta T^{*} stg}}{\left[\mathbf{1} - \left(\frac{\mathbf{D_h}}{\mathbf{D_t}}\right)^2\right]\mathbf{D_t}^{\mathbf{3}}}\mathbf{10^{-6}} \qquad [2.4]$$

Tarabin [39] also noted that a compressor of a smaller size is more sensitive to fouling than that of a larger size. Finally, he noted that sensitivity to fouling of an axial compressor increases when the stage pressure ratio increases.

Where:

ISF = Index of sensitivity to fouling

m^{o} = Mass flow rate (kg/sec)

C_p = Specific Heat (Joules/KgK)

ΔT stage= Average total temperature Rise/Stage,

Dh/Dt = Hub/Tip ratio for the first stage

D_t = Tip Diameter of axial compressor, first stage.

Assuming equal air filtration quality and environmental conditions, engines with a higher ISF will give a greater reduction in mass flow, pressure and efficiency than engines with a lower ISF.

2.2.8 Fouling degradation rate

Experience of gas turbines shows that fouling increases most quickly at the beginning of service life, then the rate at which fouling build up occurs decreases to a more or less constant rate. Figure 2-12 illustrates typical efficiency degradation due to fouling over the life of a gas turbine.

An empirical relation of Tarabin et al. [39] is ;

$$\Delta \text{Power} = a\,[1 - e^{-b}] \qquad [2.5]$$

a = 0.07

b = 0.005

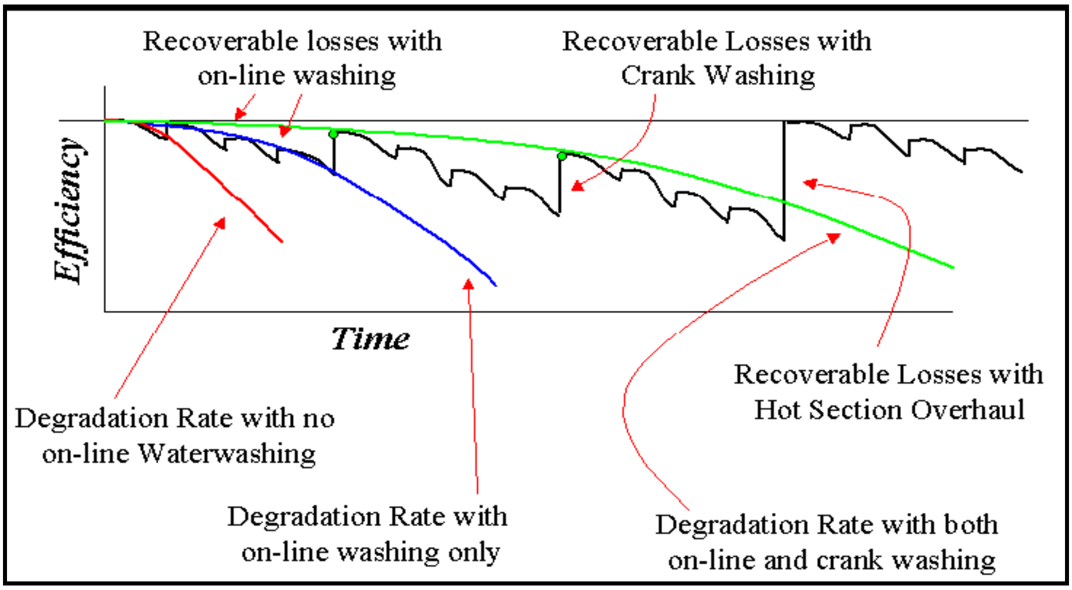


Figure 2-12 Fouling degradation rates [17]

2.2.9 Protection against degradation

The following steps are considered in minimising for protecting gas turbine degradation.

- A frequent maintenance strategy.
- Selection and regular maintenance of intake air filtration.
- Frequent on-line compressor washing.

- Thorough treatment of fuel to reduce hot section fouling.
- Following the manufactures instructions and recommendation regarding operation and maintenance procedures.

2.3 Economic and Life Cycle Cost

The field of gas turbine power generation economics has developed over the past thirty years from the simple calculation of unit generating cost in mills per kWh to the complex analysis of the operation and costs of complete systems of generating units.

It is fundamental that this development has paralleled the development of improvement methods of reliable and economic operation of power system, because the function of economic analysis is to maximise the total profits over a long term period; and this requires that the economic analysis faithfully represent actual operations, technical as well as financial [65]

Gay et al. [66] published a study (Plant Playback) describing the application of plant optimisation software to assess the benefit of plant upgrade analysis for a combined-cycle power plant. This was based on the characteristics of the power plant, its operational profile (including likely variations in loads and ambient conditions, contract stipulations, etc.) electricity sales and cost of fuel, and on configuration and performance.

2.3.1 Economic Evaluation Methods

2.3.1.1 Discount Cash Flow Rate of Return Method

The “Discounted Cash Flow Rate of Return” is the name given to the discount rate which results in the future cash flows equalling the initial investment. The best choice project is the one with the highest discounted cash flow rate of return. However, if investment risks are ignored, all that projects with a discounted cash flow rate of return that exceeds the cost of money are considered worth-while [67].

The “Discount Cash Flow Rate of Return Method” is a widely used economic evaluation model used by competitive business enterprises which examines every cash flow alternative for the duration of the evaluation. Different worth rates are used to discount each of the alternative the cash flows.

2.3.1.2 Investment Pay-Back Method

Another evaluation method, the "Investment Pay-Back Method" is used by both utilities and free market enterprises for scoping analysis. The bay back method is simply calculated as the number of years required for the net benefit to equal the initial investment. Business enterprises in a free market system use this as screening tool to examine a variety of alternatives. After narrowing down the pool of alternatives to a manageable size (typically 5 to 10) the enterprise may then conduct discounted cash flow rate of return analysis on the most promising candidates. In the regulated utility industry, the pay back method is widely used on small discretionary investments, particularly for spare parts or retrofit activities of utility [67].

2.3.1.3 Minimum Revenue Requirements Method

The "Minimum Revenue Requirements Method" is the economic evaluation method most widely used by regulated utilities, because the rate of return on any investment is determined based on what the regulator allows. That return is a weighted average return on bonds, where the calculated interest is based on bond ratings and the equity return allowed by the regulating commission.

It is possible to calculate the return to bond holders and return on equity as well as depreciation and taxes incurred for a number of alternative strategies. The best alternative is usually considered to be the one that necessitates the lowest revenue requirements. Thus, projects that have the lowest present worth revenue requirements are preferred [66].

Revenue requirements are the sum of the annual fixed charges on a new investment and the annual costs of fuel, operation and maintenance. Stoll [76] introduced a mathematical technique to levelise the cost of energy which includes a number of factors such as annual fixed charge rates for the power generation plant.

2.3.2 Fixed Charge Rate

The concept of fixed charge rates is widely used in the utility industry. Fixed charge rate is defined as the annual owning costs of an investment as a percent of the investment. A typical value in the power generation industry might be 20% /year. When an investment in utility plant is made and placed into service, the owning cost to the utility includes the following:

- Interest on bonds used to partially finance the project
- Equity return requirements of the stockholders
- Income taxes
- Insurance and property taxes
- Depreciation charge on the investment

2.3.3 Annual expenses

To a certain extent annual costs depend on decisions made at the design and construction stages of any system. The Operating and Maintenance (O&M) costs depend significantly on fuel prices (in fact fuel contributes about 70% of the total O&M costs). It is possible that actions taken to reduce capital and installation costs may lead to increased O&M costs, with an overall negative impact on the total economic performance of the project. Typical O&M costs for a co-generation plant (at 2004 prices) are 0.0035£/kWh, 0.005-0.0115£/kWh and 0.008-0.016£/kWh for steam cycles, gas turbines and reciprocating engines, respectively [68], [69], [70].

2.4 Optimisation Method Review

2.4.1 Evolutionary Algorithms

One of the most widely used numerical methods for optimising the solution of engineering problems involves the computation of sensitivity gradients. These methods have been used to optimise engineering performance over a wide range of problems. Gradient based methods (GBM) generally require a smooth design surface and either an initial guestimate of the solution close enough to the global optimum or minimum to ensure proper convergence, or the existence of only one local optimum in the region of interest.

In contrast to GBM, design space search methods such as genetic algorithms (GA) offer an alternative approach with a number of attractive features. The basic idea associated with the GA approach was inspired by the natural selection and evolution theory proposed by Charles Darwin.

The problem to be optimised is parameterised into a set of decision variables or genes. Each set of genes that fully defines one design is called an individual or a chromosome. A set of chromosomes is called a population or a generation. Each complete design or chromosome is evaluated using a fitness function that determines survivability of that particular chromosome. For example, in gas turbine applications, the genes may be a series of performance parameters associated with a cost function that is to be optimised. The fitness function takes as input all the performance and economic parameters and returns the fitness (cost function)

During solution advance (or "evolution" using GA terminology) each chromosome is ranked according to its fitness. The higher-ranking chromosomes are selected to continue to the next generation usually multiple times while the lower ranking chromosomes are not selected at all. The selected chromosomes in the next generation are manipulated using different operators: crossover (where new individuals are generated by combining existing selected chromosomes, crossover is performed by swapping parts of two existing chromosomes to produce two new chromosomes) or mutation (where all chromosomes in the population are checked bit by bit and the bit values are randomly reversed according to a specific rate) to create the final set of chromosomes for the new generation. These new chromosomes are then evaluated for fitness and the process continues from generation to generation steadily improving the design.

Constraints can be included in GA optimisation either directly into the fitness function or by pre-processing the candidate design so that if a candidate violates a constraint, its fitness is set to a value that ensures it does not survive to the next evolution level. Because GA optimisation is not gradient-based it works well in non-smooth design spaces containing more than one local optima Goldberg [71], Davis, [72] and Beasley, et al. [73], [74].

Additional useful studies which survey recent activities in the area of genetic algorithm or evolutionary algorithm research including the presentation of model problems useful for evaluating GA performance are given in Deb, [75] Jiménez, et al. [76} and Van Veldhuizen and Lamont.[77]

Other applications involving GA search methods have been in the area of multi-objective or multidiscipline optimisation, because GA optimisation techniques offer the are not limited to traditional single design point solutions, but have the capacity to compute so-called “pareto optimal sets”.

The GA used in the present project is described below including details of each of the operators (selection, pass-through, random average crossover, and mutation). The GA was tested using a hill-climbing problem and its convergence efficiency assessed as a function of the design space characteristics and the control parameter specification.

The motivation for using GAs will depend upon the nature of the application. If the optimisation problem under investigation is well behaved, then the obvious choice will be the use of conventional deterministic techniques. However, these optimisation techniques face major difficulties: when the objective function is discontinuous, or contains points at which gradients are undefined, or contains variables that interact in highly non-linear ways, see figure 2.13. Table 2.4 tabulated the different between GA and other traditional methods.

In these situations, heuristic methods like GAs are a powerful alternative for exploring search spaces and finding good solutions that cannot be detected by conventional numerical techniques. [78]

GENETIC ALGORITHMS	**TRADITIONAL METHODS**
Search a population of points in parallel.	Operate on single points
Do not require derivative or auxiliary information.	Generally requires derivative information
Use probabilistic transition rules	Generally use deterministic transition rules
Work on an encoding of the parameter set (except in real valued representations)	Work on the parameter set itself

Table 2-4 GA versus traditional methods

2.4.2 Limitations of GA

Despite its superiority in search capabilities, GAs have drawbacks such as premature convergence, poor local tuning and long computer time for convergence.

Premature convergence: This phenomenon occurs when some super individuals acquire more and more representatives. These super individuals can originate from local minima and could be far away from desired global minima.

This phenomenon can be overcome by implementing methods such as dynamic mapping of the objective function.

Poor local tuning: GAs is very efficient to detect the global search space. However, binary GAs is less efficient and encounters difficulties to reach the absolute global minima.

The problem of poor local tuning can be tackle by implementing a traditional method “a hill climber” once the area of global optima is reached.

For some cases GAs takes long time to converge, and this problem can be solved by using parallel GAs which is rather easy to parallelise GAs unlike many traditional optimisation methods [80]

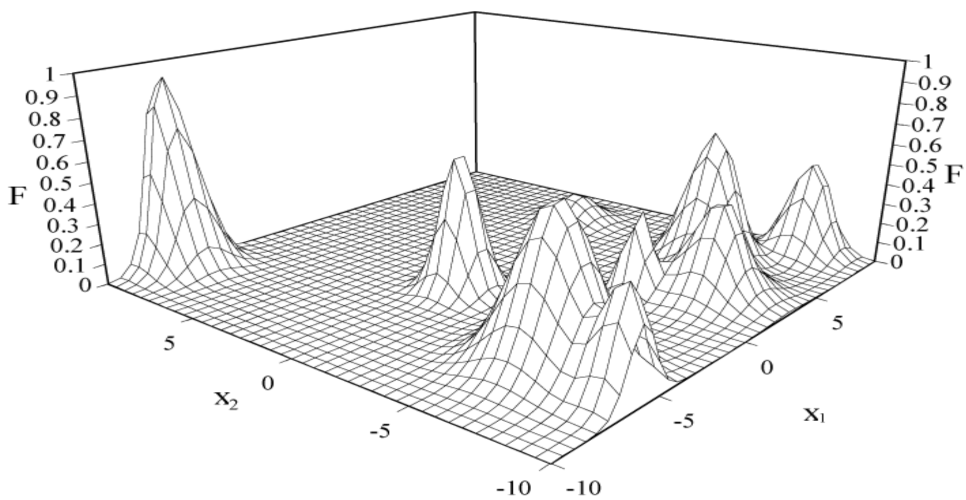


Figure 2-13 Multiple local optima

2.4.3 *Power plant optimisation*

Kikuo Fujita et al [81] have applied a GA based optimal planning method to energy plant configuration, to determine the types, models, and numbers of items of equipment, to satisfy required energy demand and to minimise the plant facility cost and input energy cost. They applied simple a GA using binary coding for plant configurations.

Richard Knight et al [82] developed an optimisation tool for gas turbine power generation economic optimisation based on GA, the economic improvements provided by the application of the tool are dependent on the particular plant configuration, the accuracy of the component models used during the modeling process, and the economic scenarios selected for plant operation, indications from the case studies reported in this paper suggest significant benefits are achievable.

For comparison a GA procedure was applied to a system sufficiently simple that the best configuration could also be found by inspection. This allowed the results to be compared and confirmed the performance of GA the methodology used [83].

2.5 Summary

From this literature review, several economics and technical aspects related to industrial gas turbine power plant selection were described. Considering these technical and economics parameters optimisation techniques have been investigated to evaluate the most suitable one in order to maximise the power plant total profit or minimise the total costs. It is concluded that the optimisation of power plants costs can benefit from using GA,s in order to maximise the profit by help the user to select the most cost effective equipment to buy or help the user to set the required loads for an existing units in order to minimise the life cycle cost.

It is also clear from the literature review that, there are many different aspects interrelated with each others that should be taken into account when optimising the industrial gas turbine power plants as the process of cost optimisation at off design conditions is not a linear process.

The literature review shows that the ongoing power plant optimisation research seeks not only to shorten the design time but also to find new methods of minimising the number of human interventions within the design process.

The advantage of employing the genetic algorithm methods, when compared with gradient based optimisation approaches are their robustness, their suitability for parallel computing, their efficiency when the number of variables is high and the objective function has many local minima and the fact that they are general and therefore do not require modification for a specific problem.

3 GAS TURBINE POWER PLANT OPERATIONAL MODELLING

3.1 Introduction

The technical objective of this work is to study the behaviour of the industrial gas turbine engine at off design conditions such as the different ambient temperatures and the power setting.

This will facilitate an investigation of the key parameters that have the major effects on the operational strategies in order to maximise the total profit for the power plant while satisfying the other constraints which will be the total power demand in this study. Also to investigate how this power plant is sensitive to each of them, as these parameters required by the optimisation tool as variable parameters.

3.2 Engine Modelling Using Turbomatch

Cranfield School of Engineering has developed a gas-turbine performance simulation suite of programs called Turbomatch [84]. This can be used to explore gas turbine design, behaviour and performance and the estimation of design point and off design point performance. The output from Turbomatch includes power, thrust, fuel and specific fuel consumption [84].

Turbomatch is based on pre-programmed routines known as “bricks” each of which simulate the performance of one of the component parts of the turbine. To model the gas turbine’s off-design point performance it is necessary to determine the engine configuration and build it from the component parameters. So-called station numbers are used to connect components. At each interface or station, the gas state is described by a station vector which contains eight quantities. BRICK requires this is done by the use of “codewords”

Inputs:

1. The inlet station vector

2. BRICK DATA=BD(K) to be input(Efficiencies, pressure ratio, pressure loss factors, etc)

3. Engine Vector Data (Compressor work, generated in COMPRE and be used in TURBIN) [84].

Outputs:

1. The outlet Station Vector

2. Engine Vector Results (Thrust, Power, Compressor work), [84].

3. Inputting the experimented results in a Turbomatch input file will require

Making a model of the engine and building it in a modular fashion by using

Various pre-programmed units called Bricks.

INTAKE: Intake

COMPRE: Compressor

PREMAS: Splitter, Bleed, By-pass duct

DUCTER: Ducts

MIXEES: Mixing of two flows

BURNER: Combustor

TURBIN: Turbine

NOZCON: Convergent nozzle

PERFOR: Final calculation of performance

The modelling of LM2500+ has been attached as Appendix 1 of this research work.

3.3 Modelling of "GE LM2500+" Gas Turbine Engine"

For the simple cycle, GE LM2500+, [86] has been selected for study. It is an industrial and cogeneration two shaft gas turbine. This machine is used in a wide variety of power generation as simple or combined cycle but it can also be used as mechanical drive. Figure 3-1

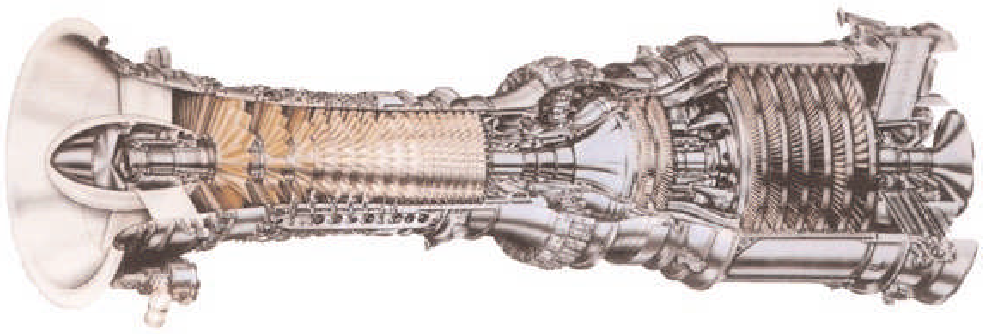


Figure 3-1 GE LM2500+ Gas turbine engine [1]

3.3.1 Design and Configuration

The application needed for this part is simple cycle power generation with the following characteristics from the literature (Table 3.1):

Frequency	**60 Hz**
Power Output	**27.6 MW**
Pressure Ratio	**23.1: 1.0**
Turbine Speed	**3600 rpm**
Exhaust Temperature	**762.00 ^{o}K (922.496 ^{o}F)**
Exhaust Flow	**80.5 Kg/s**
Heat Rate, MJ/KW-HR	**9.16**

Table 3-1 Engine data

It would be favourable to have all operating line points but the design point is the only published values available. To match the design point, Turbomatch model has been built for different engines as shown below in Turbomatch file result sheet based on Figure 3-2

The simulation procedure in Turbomatch manual has been followed to do modelling. The objective was to achieve results as similar as possible to those taken from the literature. Some minor differences taken into account between the real engine and the model such as turbine cooling air between burner and turbine which is not the case in real engine [84]

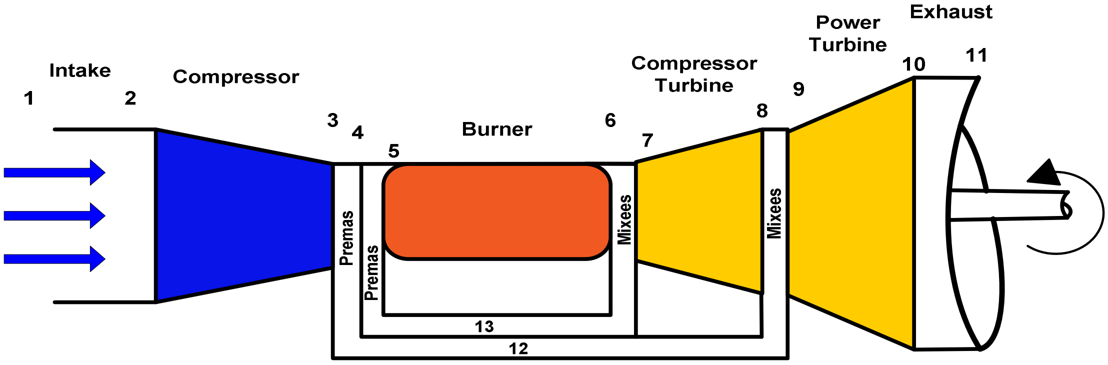


Figure 3-2 Schematic engine layout for GE LM2500+

Description	Value	Remark	Simulation	Deviation %
Ambient temperature T1 , *K*	288.15	Given	288.15	0.0
Ambient pressure P1, *kPa*	101.32	Given	101.32	0.0
Comp inlet pressure P2, *kPa*	100.82	Given	100.82	0.0
Inlet mass flow, *kg/s*	80.50	Given	80.5	0.0
Power out put, *MW*	27.60	Given	27.60	0.0
Comp. exit temperature T3, *K*	760.61	Given	754.07	0.85
Comp. exit pressure P3, *kPa*	2297.54	Calculated	2298.68	0.049
Turbine entry temp. *K*	1505	Calibrated	1505	0.0
Comp. isentropic efficiency	0.873	Calibrated		
CT isentropic efficiency	0.896	Calibrated		
PT isentropic efficiency	0.854	Calibrated		
Combustor efficiency	1.00	Asumed		
Combustor pressure loss	0.075	Asumed		
Power turbine cooling air	0.025	Calibrated		
Compressor turbine cooling air	0.075	Calibrated		
PT exit temperature T10 *K*	762.00	Given	758.20	0.49
PT exit pressure P10, *kPa*	102.88	Calculated	102.95	0.068
Exhaust exit temperature T11, *K*	758.00	Given	767.87	1.30
Exit mass flow, *kg/s*	82.29	Given	82.19	0.12
Fuel Flow, *kg/s*	1.63	Calculated	1.628	-0.122

Table 3-2 Components characteristic input data

While some parameters have been reasonably assumed such as combustor efficiency, some other parameters have been slightly calibrated like TET to match the result from literature. Table 3-2 shows the difference between engine published data and the results obtained from Turbomatcm and also from a hand calculation see Appendix A.

3.3.2 Design Point Performance

To predict the design point of the turbine requires values to be assigned to the various brick data and station vectors listed above. Appendix 1 shows how the various bricks, station numbers, brick data engine vector results and variable geometry are arranged. Appendix 2 demonstrates the running of the input file. However, to obtain a design point which matches all the engine's specified parameters (e.g. efficiency) it is necessary to make some initial intelligent guesses followed buy iterations and adaptations [86].

3.3.2.1 Components Efficiencies

Unfortunately commercial secrecy means that the gas turbine manufacturers will not release details of the efficiencies of engine components. This information has been determined from research undertaken on the LM2500 engine [87]. In addition some assumptions had to be made, and curves of typical components efficiencies from reference books [2].

- **Compressor efficiency**

Gas turbine compressors are not ideal isentropic engines and to determine compressor efficiency, it was necessary to find the compressor stage polytropic efficiency. This was found to be about 90% using available curves which plotted the axial compressor polytropic efficiency against stage loading. The stage loading was determined from the overall pressure ratio and the compressor number of stages assuming an average stage pitchline loading of 0.35, and that the curve to use was high technology [2].

The compressor efficiency is found by substituting the polytropic efficiency into the formula for isentropic efficiency together with the ratio of the major specific heats for air and the pressure ratio:

$$\eta_{ise.C} = \frac{\left[(r)^{\gamma-1/\gamma} \right] - 1}{\left[(r)^{\gamma-1/\gamma \times \eta_{poly}} \right] - 1} \qquad [3.1]$$

- **High pressure and power turbine efficiency**

The turbine isentropic efficiencies both turbines were found from charts of isentropic efficiencies as a function of the expansion ratio [2]. The efficiency was found to be 89% and this was not too different from the value of either turbines mentioned in [87].

- **Combustion section**

It was assumed that the typical combustion system efficiency was 99.5%, and the combustion chamber pressure drop was 7%.

- **Intake system**

The intake system total pressure drop was assumed to be 0.5%.

- **Percentages of bleed air**

The percentages of bleed air from the rear compressor stages had to be assumed as this is information held by the manufacturer and not released. They are not yet accurately known and were assumed to be:

  - 2.5% for the power turbine blades cooling.
  - 8% for the high pressure turbine blades cooling.

The simulation of design point and off design for different ambient conditions using the Turbomatch software was carried out on the General Electric LM2500 plus industrial engine. The simulation was performed in order to obtain representative engine behaviour with changes in a number of parameters and to be able to analyse the simulation results.

3.3.2.2 Engine cycle calculation

To identify the pressures and temperatures at different engine component stations a number of engine cycle calculations were performed using data available from the GE official website, other reference material and certain assumptions. The main parameters of interest were the turbine entry temperature and compressor efficiency.

3.3.2.3 Design point simulation results

The output file results can be listed as shown in the table below:

Power	27.6 MW
Fuel flow	1.6 Kg/s
SFC	59 mg/Kw.s
Exhaust temperature	1057^0K
Thermal efficiency	39.3 %

Table 3-3 Output file results

Figure 3.3 shows the plots of specific fuel consumption Vs specific power output for the engine studied. These plots gave a comprehensive representation of what took place when the firing temperature and the pressure ratio were allowed to vary.

The curves shows optimum specific fuel consumption and specific power output for different firing temperature and pressure ratios. It should be noted that the optimum efficiency that is the lowest specific fuel consumption, for a given firing temperature tended to occur at higher pressure ratio than the optimum specific power output.

Some of design point characteristics can be obtained from manufacturers and some are not published for the competition such as TET. Manufacturers are using either computer matching simulation or statistical averaging of production engines. So design point can certainly not be realised [2].

Appendix A. Shows the input and output data files used for the design point analysis

In the case of gas turbine engine, for an optimised design, as already mentioned earlier, the obvious requirements are a low specific fuel consumption coupled with a high specific power output. These can only be achieved by employing a high pressure ratios and high turbine entry temperature, see figure 3-3. Further improvements in gas turbine plant efficiency can be obtained by adding different components such as intercooler, heat exchanger or both to the original cycle, but these improvements are going to be at more cost.

There are some other economic aspects should be taken in consideration during the optimisation of the plant, these are; the first cost, the life cycle cost, running and maintenance costs, electricity price, fuel cost, emission tax and other costs which will discuss in details later. As shown in this figure the specific power increasing with increases the turbine entry temperature and also with increases the pressure ratio.

Selection of gas turbine parameters that have major influences on the optimisation process is the most important issue because it is the key start for a good optimisation.

Figure 3-3 shows that the SFC of gas turbine engine decreasing with the increase of turbine entry temperature (TET), the higher (TET), the higher efficiency of the gas turbine and the higher heat content of gas turbine exhaust (in case of the combined cycle) and this will increase the efficiency of the combined cycle which is significantly dependent on the heat content. However, TET cannot be increased indefinitely because of temperature limitations of the materials and also the cooling system for the turbine blades.

Figure 3-3 stated that increasing the pressure ratio decrease the SFC (increases the cycle efficiency). However, as pressure ratio changes, the specific power also changes.

Thermal efficiency and specific power both are strongly influenced by pressure ratio and turbine entry temperature. However, the maximum specific power and the maximum thermal efficiency do not occur at the same pressure ratio. So, in designing gas turbines, the design pressure ratio must be a compromise between the maximum thermal efficiency (minimum specific fuel consumption) and the maximum specific power in order to decide whether we need a small engine (high specific power) or an efficient engine (low specific fuel consumption) and this is directly relates to the capital and annual cost for the engine as it will discus in details later [88].

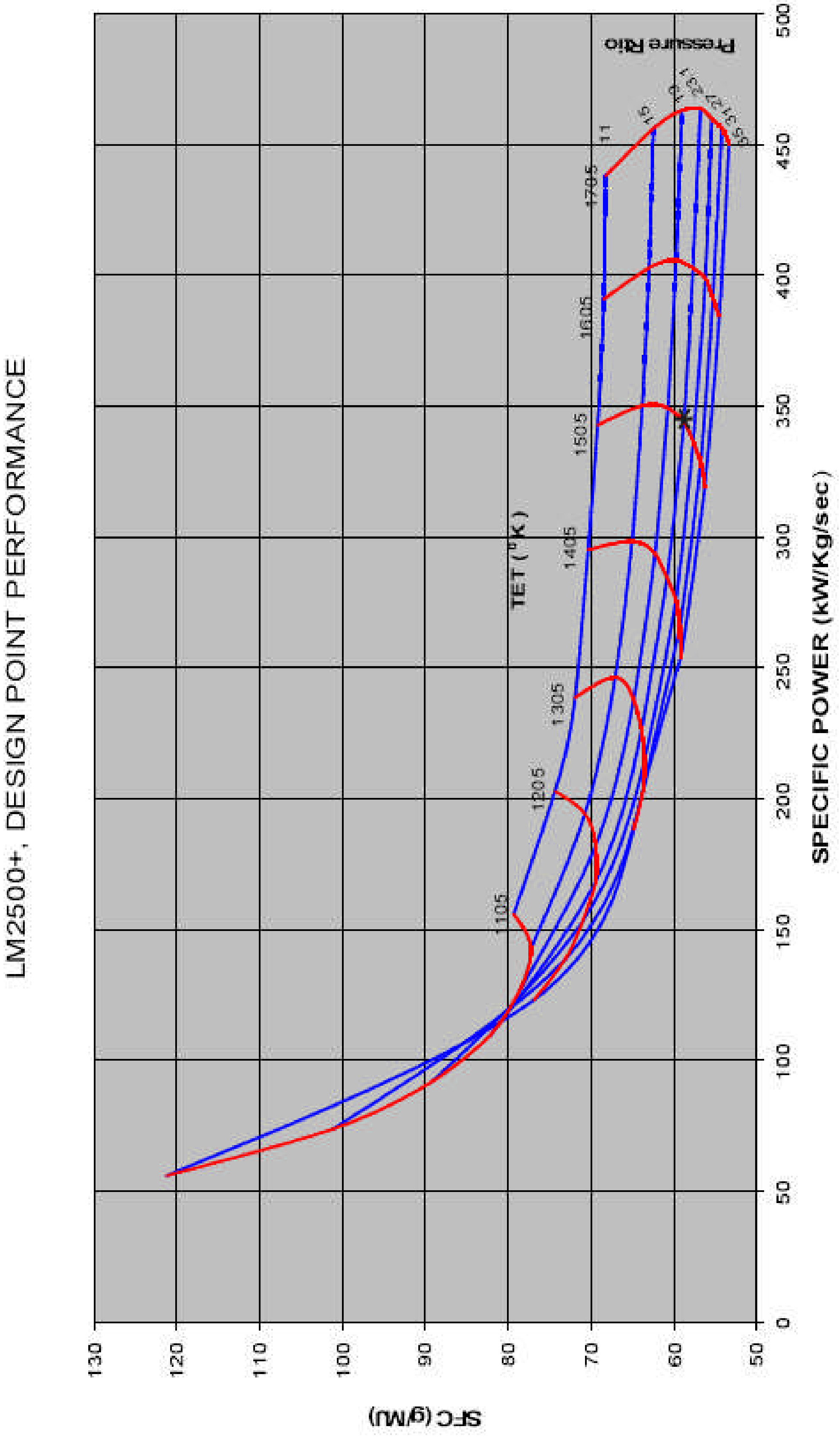


Figure3-3 GE LM2500+ Design Point Performance

Figure 3-3 GE LM2500+ Design Point Performance

3.3.3 Off-Design Performance

This section investigates the change in steady state performance of the engine as operational conditions change. Having determined the operational conditions it is possible to obtain various performance curves for the engine, these include specific thrust or power, specific fuel consumption, thermal power, etc.

As has been discussed previously the temperature, density and relative humidity of the inlet air all affect the performance of a air-breathing gas turbine. The effects of these parameters on off design performance will be studied separately [2].

3.3.3.1 Influence of Ambient Temperature

It is well known that as the temperature of air increases its density decreases and so will the air mass flow rate into a gas turbine, which has a consequent effect on its performance. In an ideal cycle when inlet ambient air temperature increases more work is done during compression (because entry temperature is proportional to ambient temperature) but the work done during expansion remains constant. The useful work is the difference between them which decreases, (figure 3.4).

Ideally, the pressure ratio of the cycle is unchanged and because thermal efficiency is a function of pressure ratio only, it also is unchanged.

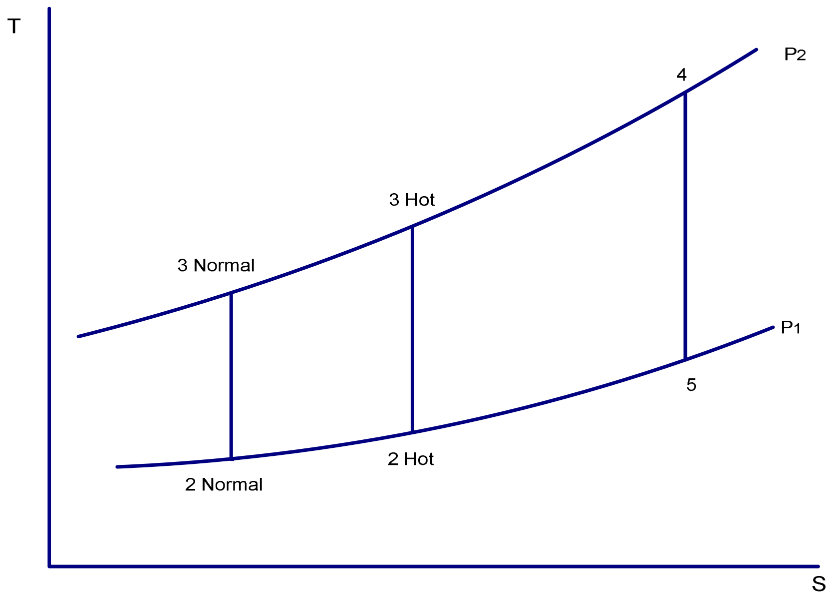


Figure 3-4 Effect of increased *Ta* on the ideal cycle

However, in a real cycle thermal efficiency will depend on the temperature ratio T_{04}/ T_{02}, so the thermal efficiency will fall along with the useful work. This can be seen by considering the engine operating at a constant shaft speed; as ambient temperature rises $(N/\sqrt{T_{02}})$ will fall, as will the compressor pressure and temperature ratio figure 3.5.

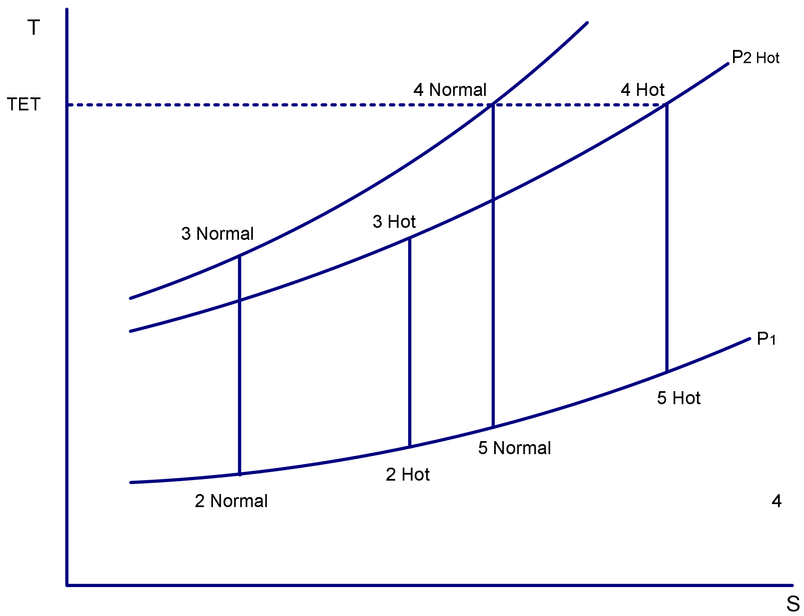


Figure 3-5 Change of the ideal cycle due to *Ta*

If TET is maintained constant, an increase in ambient temperature will result in a drop of fuel flow. However, for a given fuel flow rate (constant rate of heat input) and a given mass flow rate (of constant specific heat) an increase in inlet temperature will mean an increase in outlet temperature, that is if ambient temperature increases, $T_{02\ hot}$ increases.

The reduction in PR is not sufficient for $T_{03\ hot} > T_{03\ normal}$. This results in falling heat input, fuel flow and mass flow.

How change in ambient temperature over the range -25^{0}C to +55^{0}C affected engine performance was investigated. The graphs of power output and efficiency as functions of ambient temperature at different values of the TET were plotted in figures 3-6 and 3-7

As would be expected there is a substantial and more or less linear decrease in engine output power as ambient temperature increases from -15^{0}C to 50^{0}C, see figure 3-6. The efficiency of the engine also falls, with the biggest fall at lower values of the TET, see figure 3-7.

The decline in output power and thermal efficiency with increase in ambient temperature can be explained as follows:

- A decrease in the density of the inlet air means a reduction in mass flow into the gas turbine and simultaneously more work is required to compress the warmer air. The opposite effects occur when the temperature of the inlet air falls.

- Thermal efficiency depends on the ratio T_3/T_1, if ambient temperature T_1 rises the thermal efficiency will fall. On the other hand a drop in the ambient temperature will increase thermal efficiency.

The seasonal ambient average maximum temperatures at the Sarir oil field are shown in figure 3.8. These can be used to obtain a reasonable estimate of the range of relative power outputs which can be obtained from the GE LM2500+ gas turbine.

Figure 3.9 shows the calculated performance corrections for efficiency, fuel flow rate, mass flow rate, power output and specific fuel consumption. With increase in ambient temperature there is an increase in the heat rate which is directly proportional to fuel flow and inversely proportional to power output. Thus there is a drop in power output as ambient temperature increase. This suggests that for an increase in ambient temperature the same fuel flow will give a lower power output from the same engine, alternatively for the same power output an increase in fuel flow rate is required. This is important because fuel is the largest single portion of the total annual running cost.

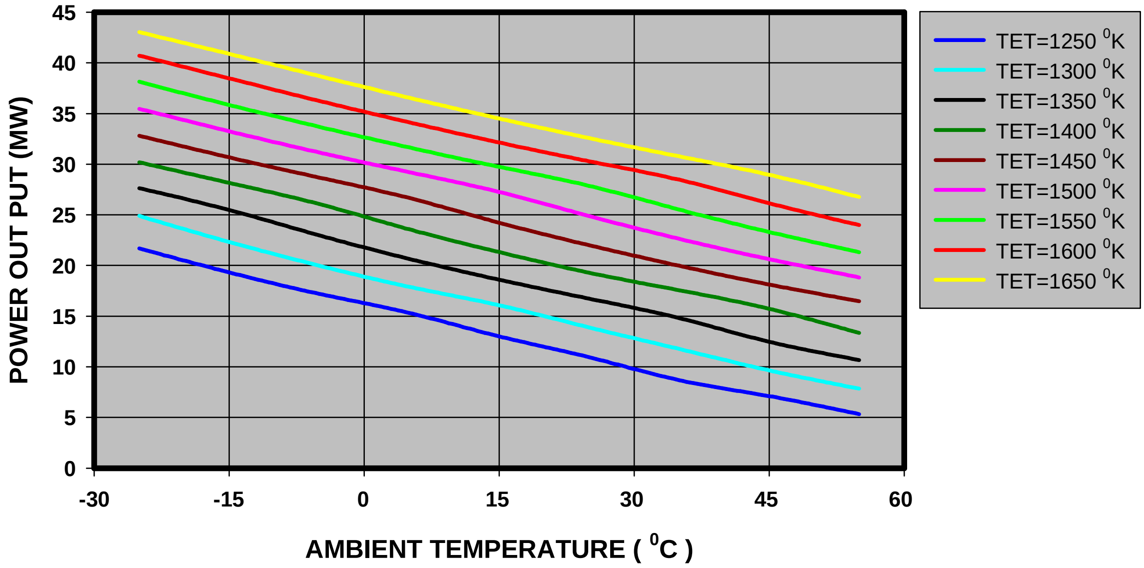


Figure 3-6 Engine power variation with ambient temperature

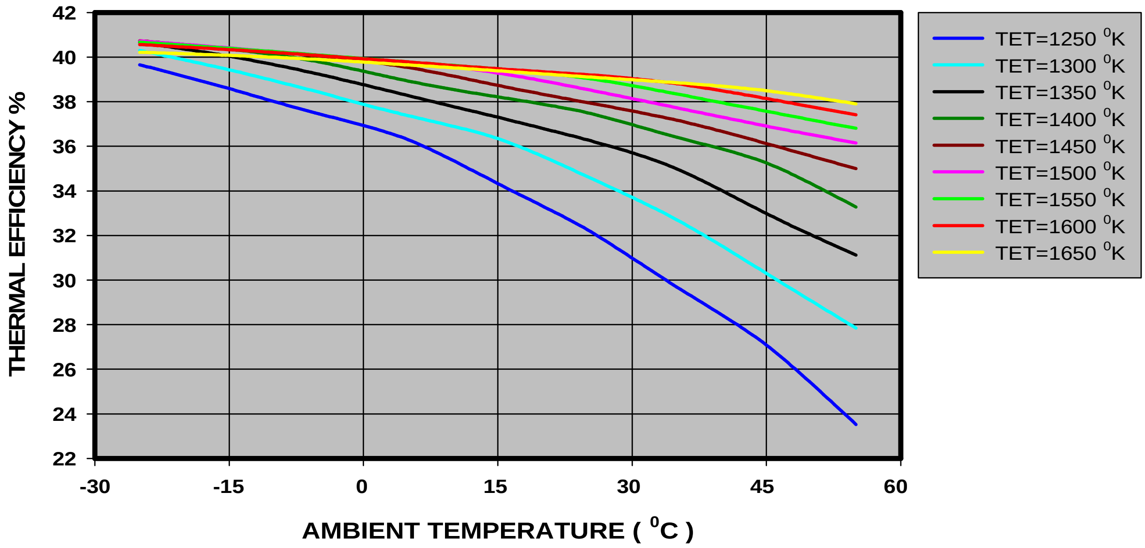


Figure 3-7 Thermal efficiency variations with ambient temperature

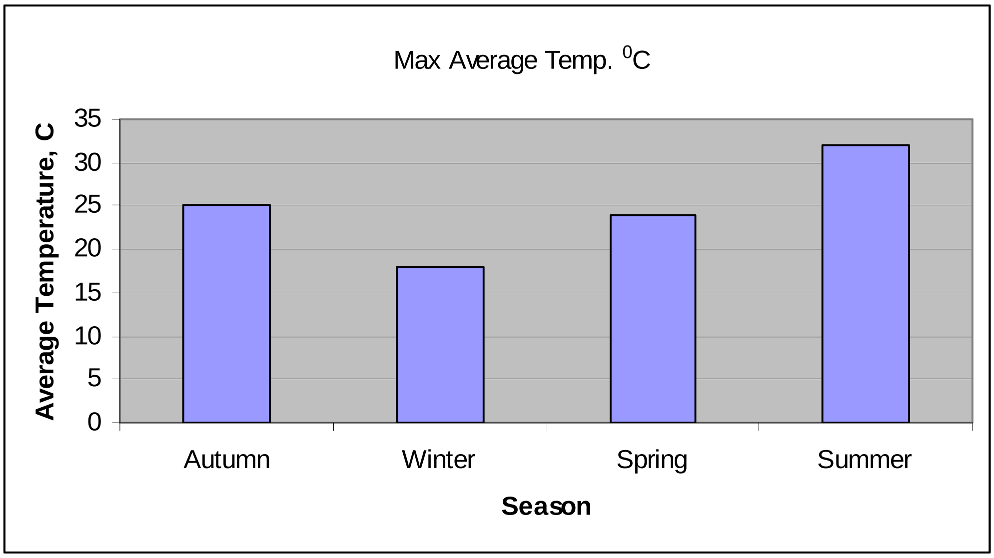


Figure 3-8 The average maximum seasonally temperature at Sarir oil field

The effect of ambient temperature on gas turbine performance can be summarised in figure 3.9, as it shows correction curves for ambient temperature and its influence on the power output, heat rate, thermal efficiency and air flow.

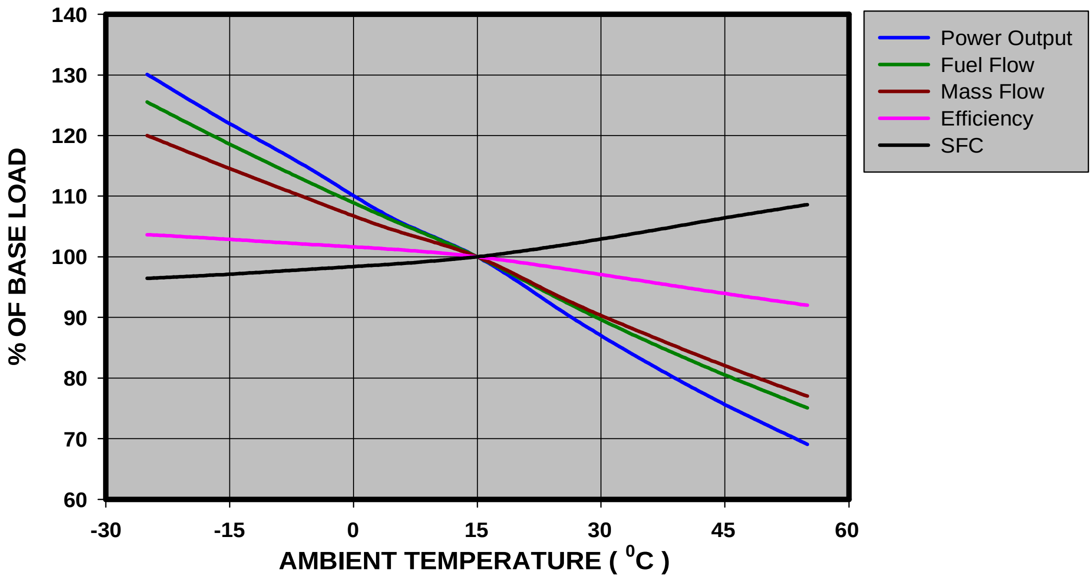


Figure 3-9 Performance corrections for ambient temperature

The pressure will decrease and the temperature will increase in all stations due to ambient temperature increase and the opposite is correct for ambient temperature decrease.

The influence of ambient temperature on overall efficiency is affected by the power setting (TET). If the engine is operating on part load, the drop of efficiency will be much higher than if turbine is running on base load. But the power output is directly proportional to TET.

Figures 3-10 and 3-11 concluded that the engine thermal efficiency decreases as power decreases (due to decreased electricity load). This is exactly the opposite of what has happened when the entry compressor total temperature has fallen. In this case T_2 and N are both constant and in order to reduce the power, pressure ratio and TET must be decreased which results in lower thermal efficiency, and more clear presentation of what is happening can be seen in the following figures.

3.3.3.2 Influence of Firing Temperature

The increase in TET generally has a positive effect on all the engine parameters. This is more obvious at higher TETs because the component losses become relatively less important. Obviously, care must always be taken in order not to exceed the thermal limitations of the turbine working components (blades, rotors etc.). Above 1300^0K a cooling system is necessary.

Figure 3-10 shows how efficiency increased with increase in the firing temperature from the design point. At low ambient temperature, -25^0C, and the engine is already working near its maximum power rating even for fairly low values of the TET. Thus an increase of TET under these conditions would be expected to produce only a marginal improvement in power output. However, at higher ambient temperatures of 45^0C to 55^0C the cycle efficiency is relatively low for lower values of the TET, so here an increase in TET would be expected to give a noticeable improvement in cycle efficiency. Figure 3-11, shows a fairly uniform increase in power with increase in TET for all ambient temperatures.

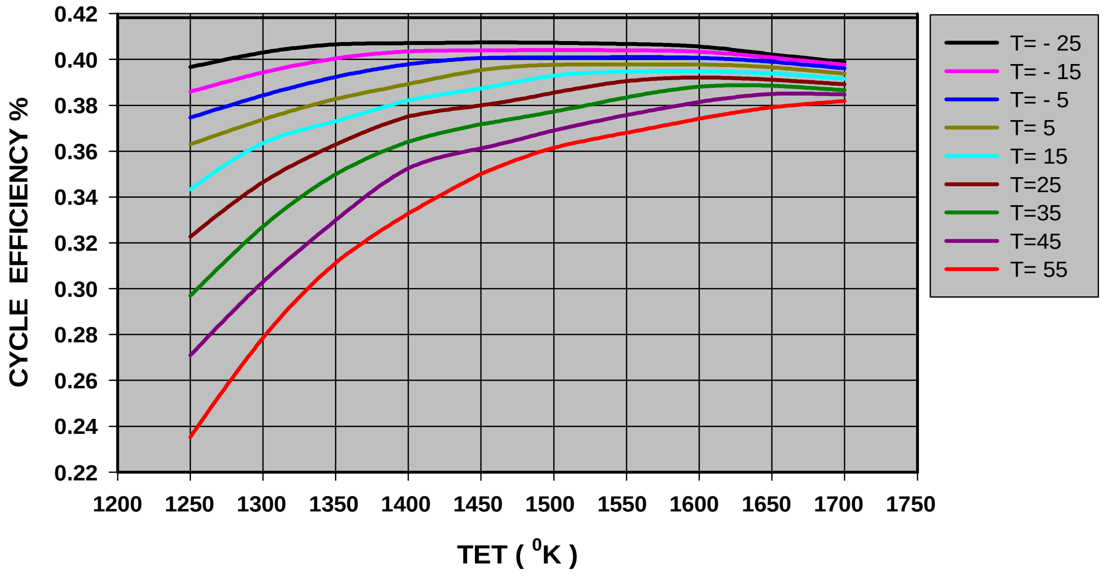


Figure 3-10 Thermal efficiency variation with TET

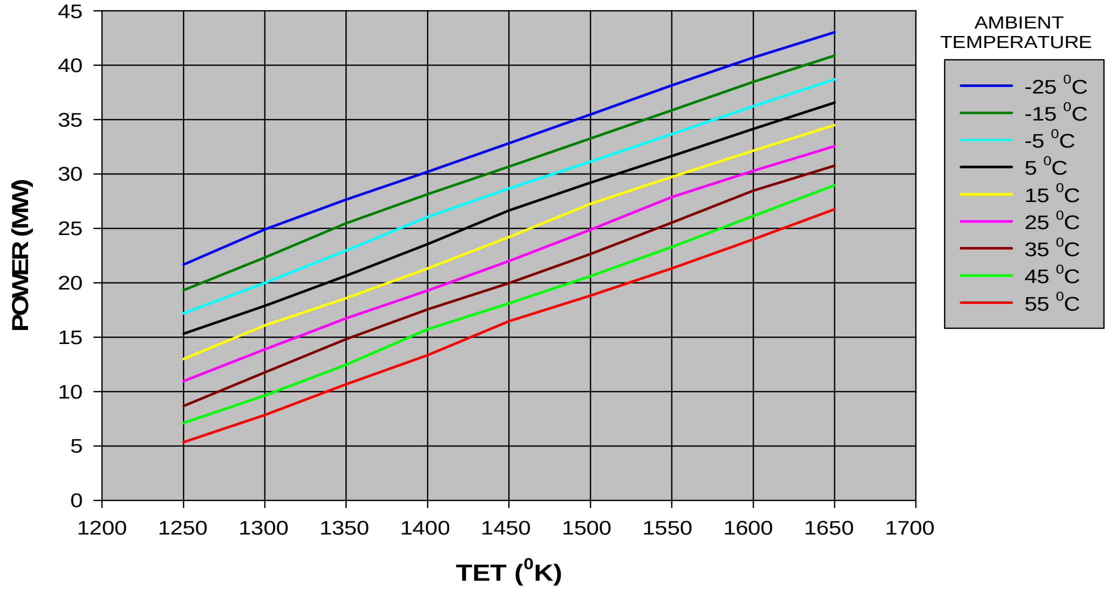


Figure 3-11 Power output variation with TET

3.3.4 Performance Degradation

3.3.4.1 Introduction

Many factors contribute to degradations of gas turbine engine performance with the passage of time including the following:

- Compressor fouling
- Sand particle erosion
- Salt erosion and corrosion
- Increased running clearances due to wear
- Bearing wear
- Fuel contamination

This study deals only with the performance degradations that take place as a result of compressor fouling.

3.3.4.2 Compressor Fouling

Compressor fouling is probably the most common cause of performance degradation. In a desert oil field environment, a significant contribution to engine performance deterioration is the fouling of compressors caused by the very small sand particles which pass through the inlet filters and mix with oil deposits. Such fouling reduces both compressor efficiency and air-flow capacity due to increasing blockage of the gas path. The effect of reduction in compressor efficiency is shown in figure 3-12.

This gradual build up of deposits over a period of typically one year following overhaul, necessitates a continuous increase in firing temperature to maintain constant output shaft power. This, of course, results in higher fuel flow as shown in figure 3-13, and seriously reduced hot section turbine blade life due to the increasing in firing temperature, see fig 3-14.

In addition, and perhaps as seriously, the build-up of deposits in the compressor changes the natural frequency of the blades. In combination, these factors can increase blade vibrations, reduce compressor surge margin and make the compressor more prone to failure. Figure 3-15 shows the effect of compressor degradation on power output and thermal efficiency. It can be seen that as compressor degradation increases from 1% to 5% the efficiency drops by 4% and the power drops by about 3%. The drop in power output and thermal efficiency is because increasing degradation leads to an increasing drop in pressure ratio.

In such circumstances a power plant might require an increase in TET to meet its installed capacity. (The effect of increase in firing temperature on the engine performance has already been discussed above.)

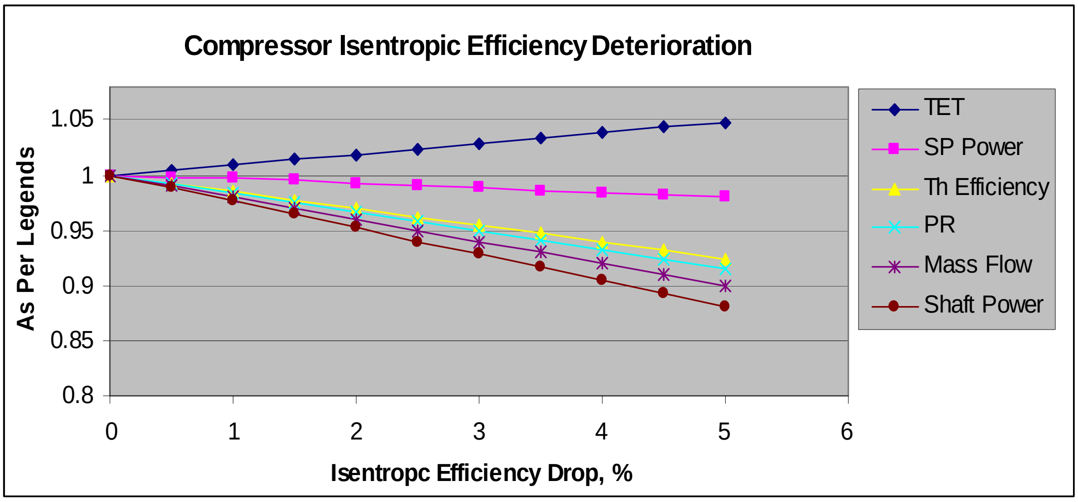


Figure 3-12 Effect of reduction in compressor efficiency

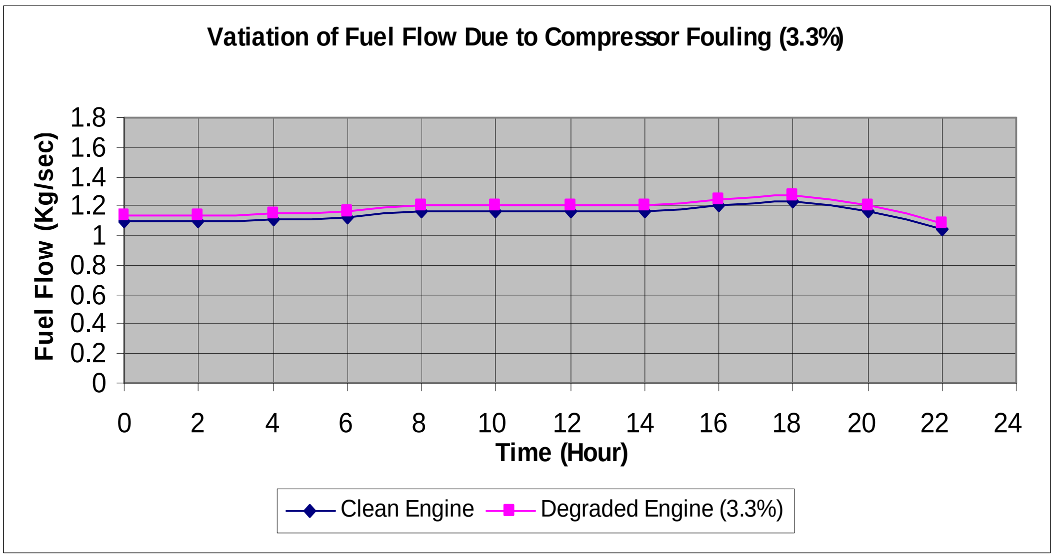


Figure 3-13 Variation of fuel flow with 3.3% compressor fouling during summer day

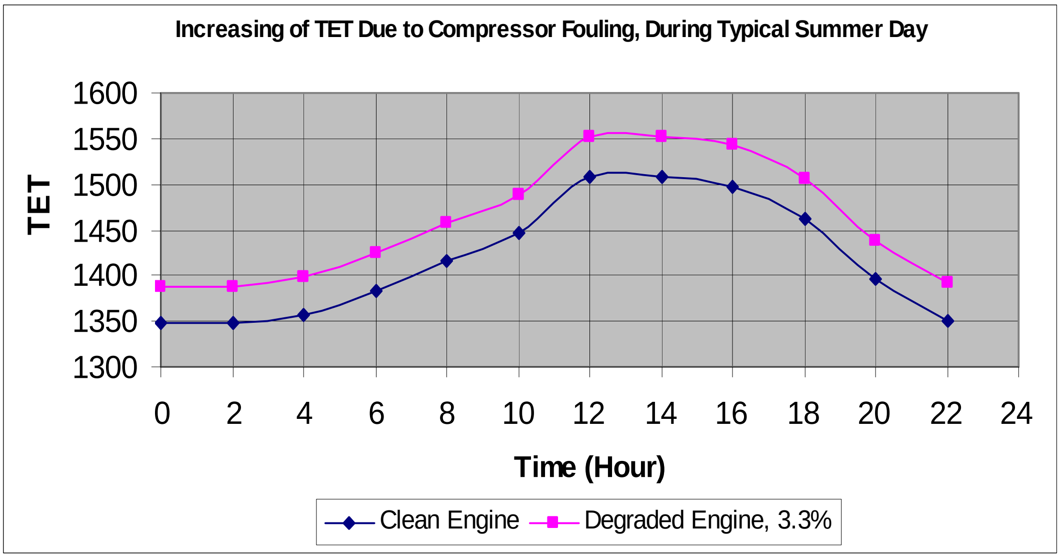


Figure 3-14 Increasing of TET due to compressor fouling during summer day

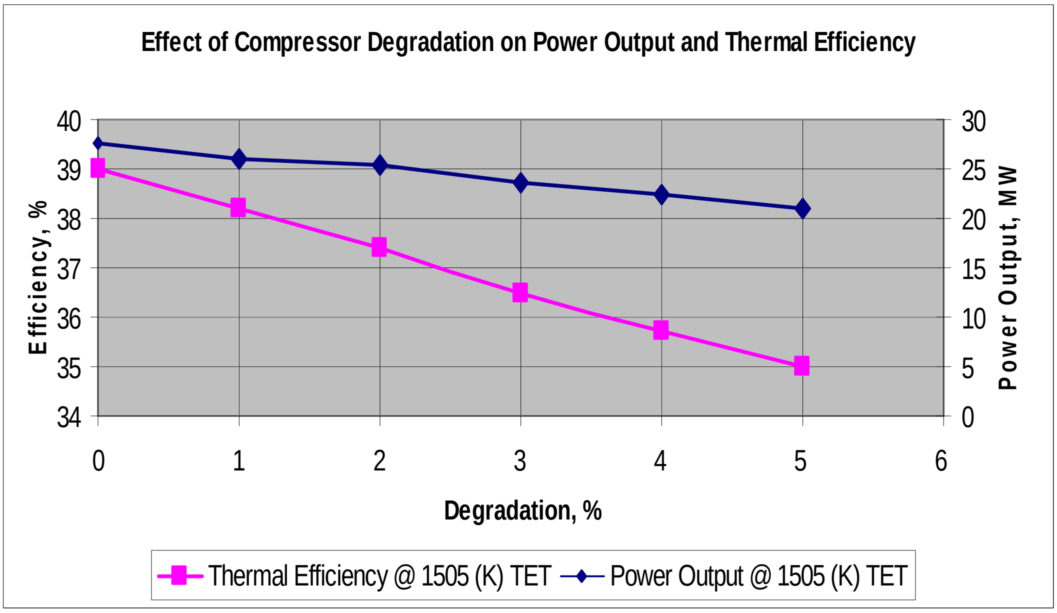


Figure 3-15 Effect of compressor degradation on power output and efficiency

Now it is possible after design point simulation completed to plot the compressor map, see figure 3-16, and to show the design point. The horizontal axis of the compressor characteristic map will represent the non-dimensional mass flow, while the vertical axis will represent the pressure ratio.

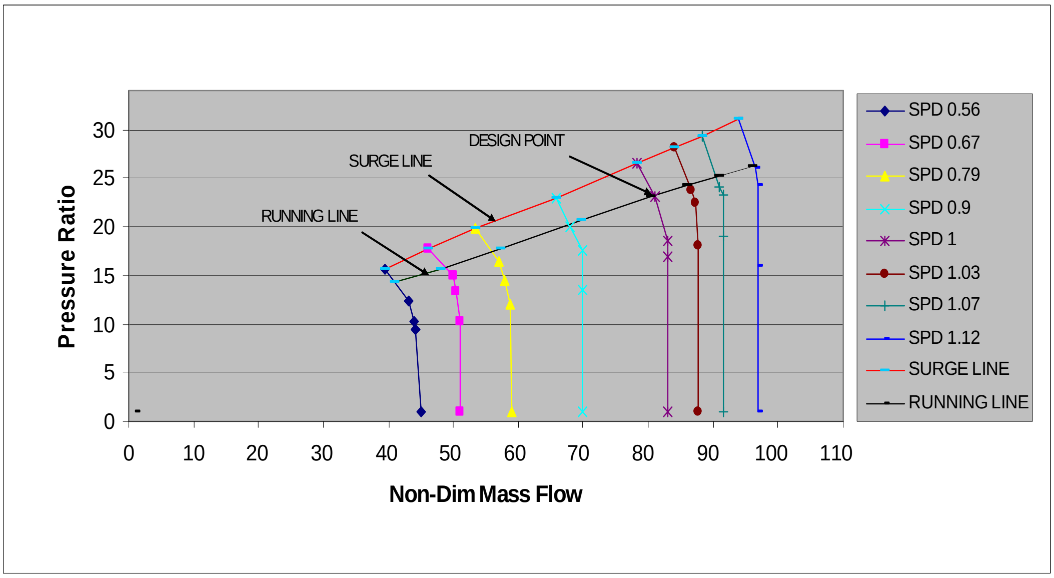


Figure 3-16 NDMF vs. pressure ratio engine compressor map

In figure 3-17, the compressor map of the LM2500 + represent the off-design operation with the running line for ambient temperature variation, and also the surge line with the various engine running speeds. The pressure ratio (PR) vs. Non-Dimensional Mass Flow (NDMF) graphs in figure 3-17 shows that the speed line moves towards the left as the compressor degrades from 1% to 5% the running line of the compressor also moves towards surge. The engine will most likely surge when transient operations are being carried out at very high level of degradation.

Also from figure 3.17 it can be noticed that the effect of increased inlet air temperature will result in reduced air density resulting in reduced mass flow which will reduce (W√T1/P) and thus resulting in unstable engine operation as a result of the operating line being closer to the surge line. In the range from 50 to 70 of non-dimensional mass flow, which is the case of increased inlet air temperature, it can be seen that the operating line approaching the surge line.

It becomes important to therefore estimate the level of degradation which has taken place in the compressor to carry out planned maintenance. This can be achieved by hot gas path condition health monitoring.

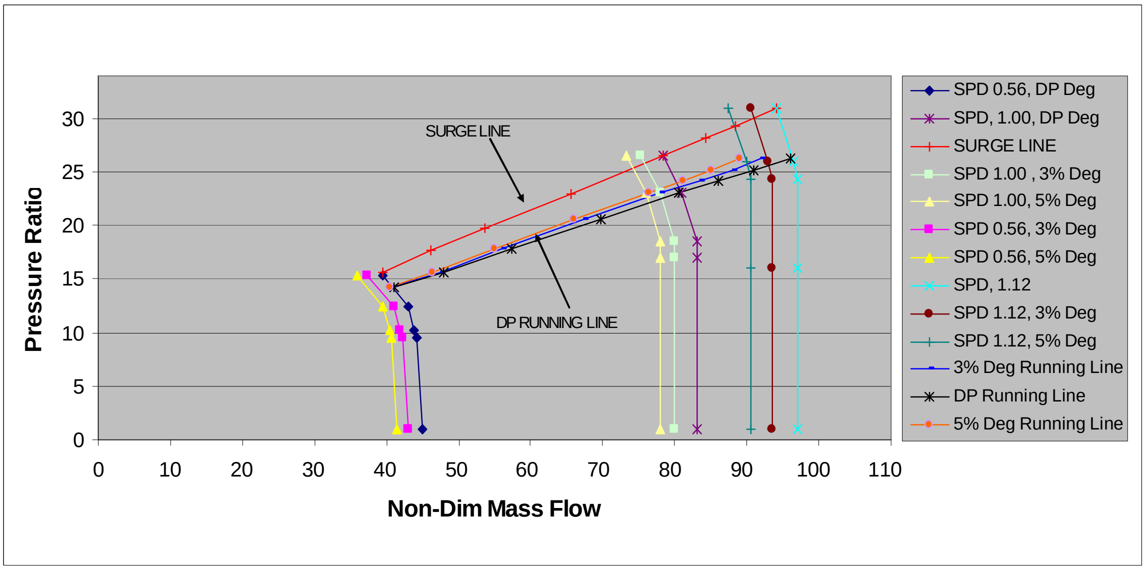


Figure 3-17 Effect of compressor degradation on compressor running lines and speed line

As mentioned above compressor fouling led to significant reduction in compressor mass flow which has a strong effect on the engine performance especially the power output which drops by about 10 % as can be seen in figure 3-18.

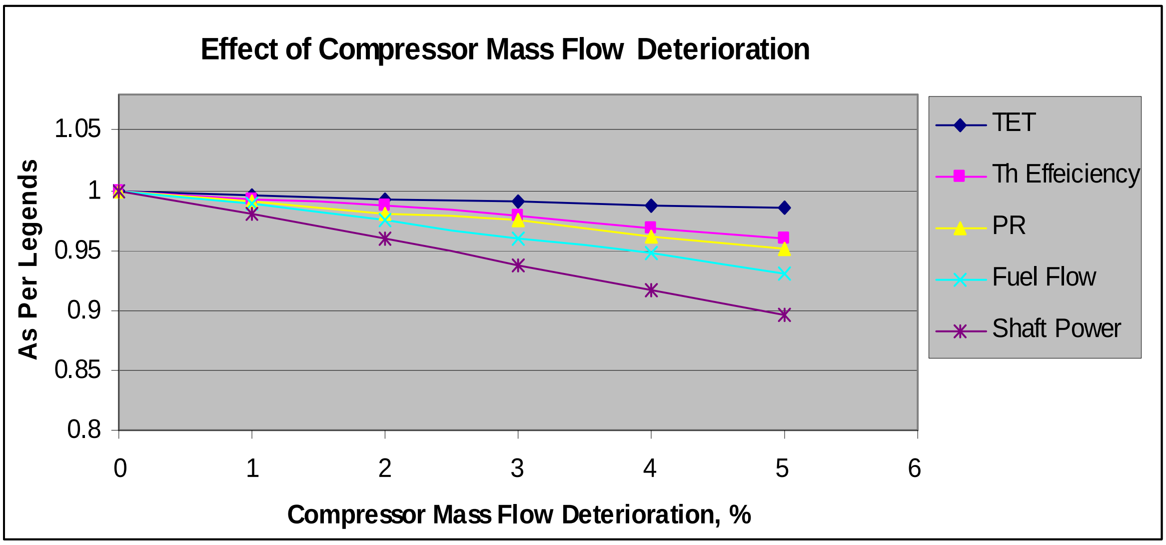


Figure 3-18 Engine performance deterioration by mass flow

3.3.5 Summary

LM2500+ Industrial gas turbine has been simulated in this part as a model to investigate its behaviour at different operating conditions as shown in figures 3-6 to 3-18 and tables 3-1 and 3-2. For the engine design point performance some parameters have been assumed, some given and some calibrated as it is shown in table 3-2.

The simulation showed that the key parameters for the ambient and the operating parameters that will effect on the total profit for the gas turbine power plant are the ambient temperature and the power setting (TET).

Altitude has also a significant effect on the total performance in case if the power is in the stage of the location selection.

For the ambient temperature as it is clear in this investigation, gas turbine likes cold wither and the choice is to reduce the temperature in hot times by using inlet cooling system which is will be a matter of economic to compare the benefit with the cost.

Figures 3-10 and 3-11 illustrated the effect of turbine entry temperature on the power and efficiency of the gas turbine and each engine has its own sensitivity to TET change.

in most power plants which is include more than one engine, the challenge is, when and how to operate the different units according to the power demand (TET) in order to minimise the annual cost from an operating point of view which is will be with the other aspects the main challenge for this study.

It is has been shown that degradation due to compressor fouling plays a significant role in reducing the performance of the engine which will impact life cycle cost as will be discussed in detail later.

Many other engines have been simulated by the same way using Turbomatch to establishing a library for wide type of engines and cycles.

4 GAS TURBINE POWER PLANT ECONOMIC EVALUATION

4.1 Introduction

A major task for this project is to develop a procedure which integrates power plant economic activities, so that economic measures of the system, such as fuel, maintenance, operational and other costs and cumulative revenues can be evaluated with time along the operating horizon. These system level economic metrics provide a basis from which to evaluate long and short term plant profitability when optimising operational performance. One approach is to integrate local economic metrics along the entire operating time horizon. Such an approach requires the accuracy and efficiency to be balanced evaluating long-term economic measures and inter-relationships. Such an approach requires numerous evaluation points.

This chapter introduces a systematic approach to evaluate the economic performance of gas turbine power plants.

4.2 Industrial Economic Analysis Methods

The financial objectives of power generation companies are to maximise earnings per share consistent with good business practices. Industrial economic analysis procedure is focused toward this objective. The method most widely used to analyse these financial obligations is the discounted cash flow rate of return method.

This discounted cash flow rate of return method (DCFR) begins by calculating revenues, subtracting expenses, and computing earnings for each year over a several year horizon period or over life of a project. Each year’s earnings are discounted back to the year that the investment is to be made. This process is repeated using several values of discount rates. The discount rate for which the cumulative present worth net revenues equals this initial investment is defined as the DCFR.

4.3 Total Profit Equation (Objective Function)

Gas turbine power plants produce power and thus generate revenue and profits for industries that deploy them. However, costs associated with the operation of gas turbines include fuel, operation and maintenance, depreciation, returns paid to investors, and more recently such costs as emission taxes. The sum of these costs is the operating costs.

The capital cost is the purchase cost of the gas turbine and auxiliary systems it also includes of cost of the space where the engine will be placed. The abandonment cost is the cost of safely disposing of the gas turbines and associated systems at the end of their lives. The sum of the capital cost, the operating cost and the abandonment cost is referred to as the total ownership cost or life cycle cost (LCC).

The difference between the total revenue (TR) and the LCC is total profit (TP) generated. The lower the LCC and the higher the revenue the greater the TP. This section of the project aims to develop an algorithm which will predict the cumulative LCC for an industrial gas turbine power plant operating at different ambient conditions and loads, see figure 4-1

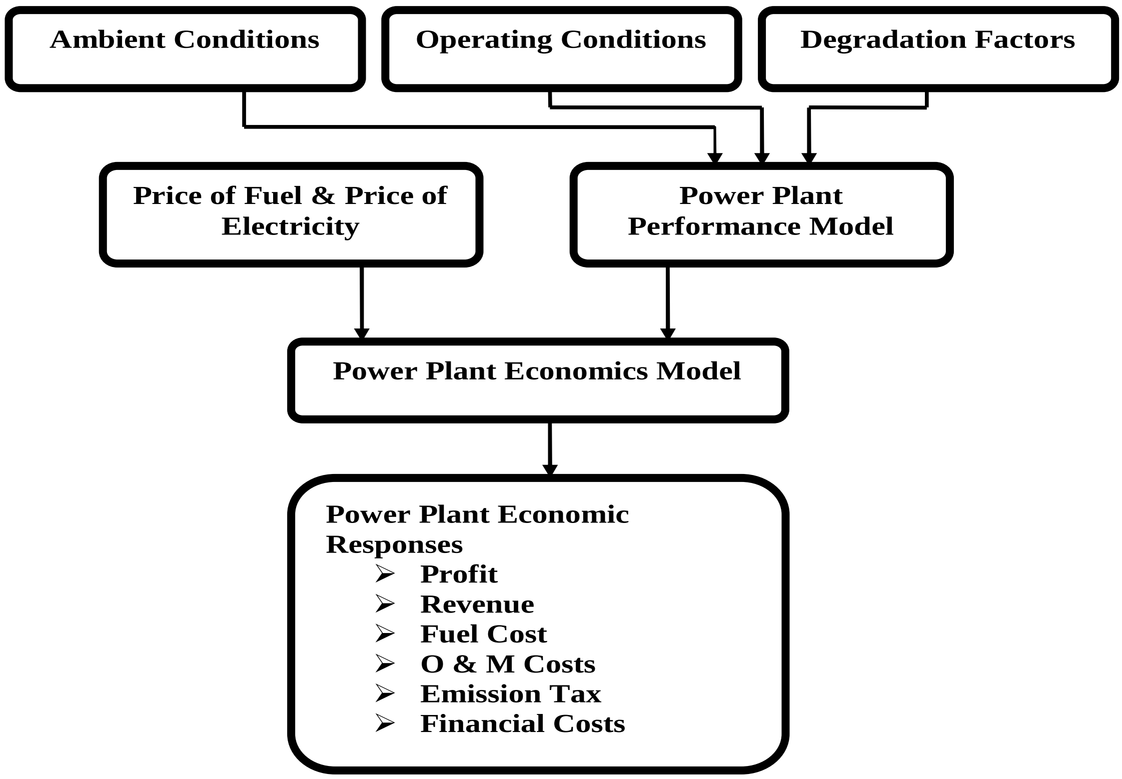


Figure 4-1 Analysis process flow for power plant economics

4.4 Power Market and Environmental Conditions

Much research effort has been expended on electricity pricing, forecasting the price of fuel, power demand and supply forecasting [89], [90]. In these considerations three major factors have emerged which are functions of the time of year: ambient conditions, the price of electricity and the price of fuel.

Theoretically in a deregulated power market, it would be expected that the prices of electricity and fuel would be stochastic or random in nature. However, seasonal demand and daily use impose short-term, seasonal and long-term patterns. For example, on a daily basis, electricity prices tend to be lower between midnight and 6 a.m. than during the day, because people use less electricity during that period.

Again the price of electricity is higher in the winter (in the UK) than that in the summer, because demand is higher in the winter. In a market driven environment, the prices of electricity and fuel are major considerations for operational planning of power plants. Weather shows strong seasonal variations which are not stochastic processes in the long term and the trends in ambient temperature and relative humidity are important factors impacting on gas turbine performance. To investigate price of electricity and fuel as a function of ambient conditions such as pressure and relative humidity is beyond the scope of this work.

Yet a simple model able to capture the consequences of seasonal variation of e.g. temperature on the dynamics of electric power market is necessary for this project. In this model the ambient temperature and the prices of electricity and fuel will show daily variance, seasonal and long-term trends. The chosen site is one of the hottest places in the world (Sarir in the Sahara desert), see figure 4-2, so the ambient conditions inserted into the model will be quite unlike those of the UK.

Daily average ambient temperature readings taken at two hourly intervals are shown in figure 4.3, which also shows the daily profile of power demand [91]. As can be seen the temperature varies from a relatively low value of about 15^0C overnight, rises fairly rapidly to about 48^0C just after midday, and then declines gradually back to 15^0C at midnight. The power demand pretty much follows the daily temperature and the daily variation of price of electricity for a typical summer day is shown in figure 4.5

Appendix 1 provides ambient temperature and power demand profiles for autumn, winter and spring. The average maximum seasonal ambient temperature at the Sarir oil field is shown in figure 4-4. The corresponding average seasonal variation in the price of electricity is shown figure 4-6. The price of electricity that has been used in this study is lower between midnight and early morning than that during the day, and the price of electricity is higher in the summer than that in the spring, autumn and winter, due to high power demand in the summer.

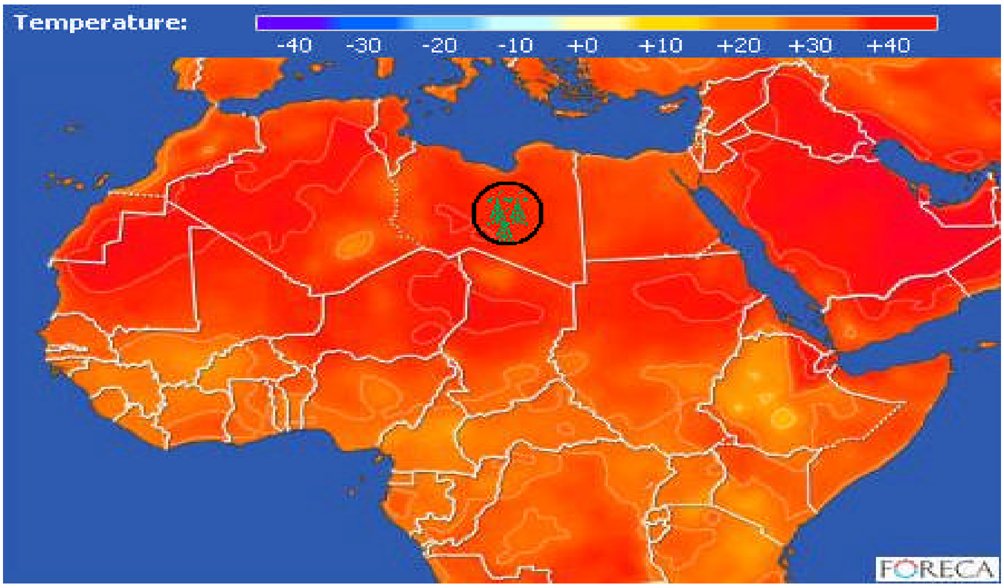


Figure 4-2 Weather Map for the Sarir Oil Field in Libya

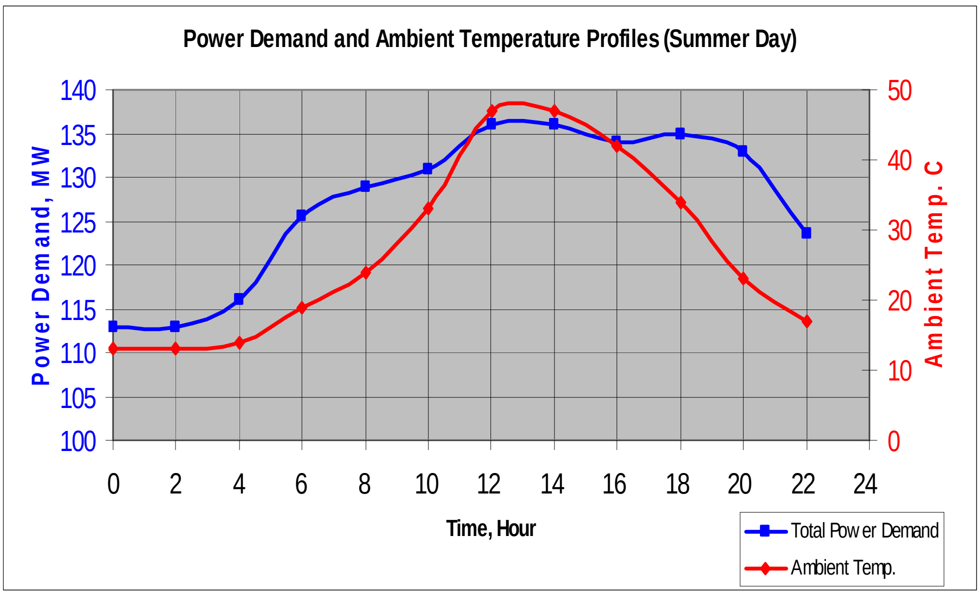


Figure 4-3 Ambient temperatures and power demand profile for summer day [91]

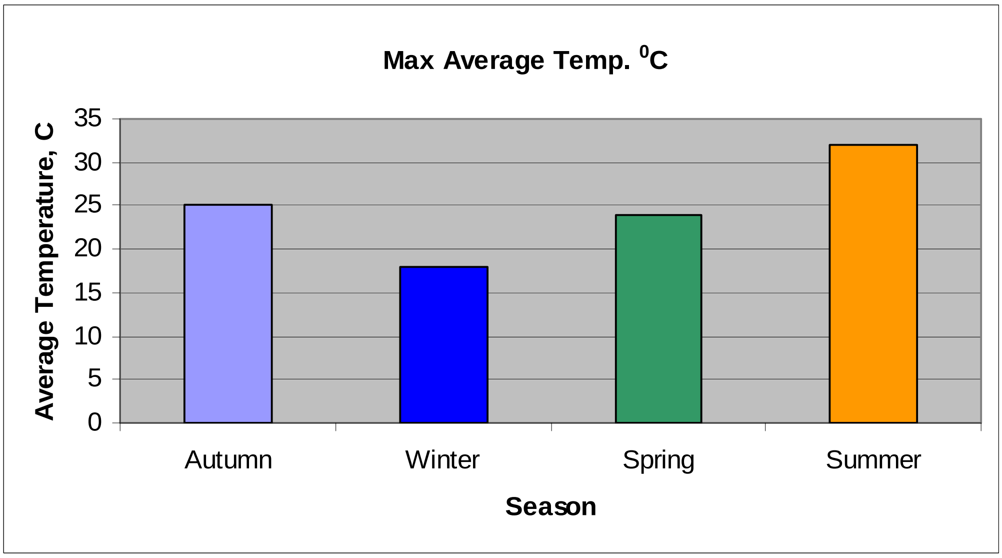


Figure 4-4 Average maximum seasonally temperature at Sarir Oil Field [91]

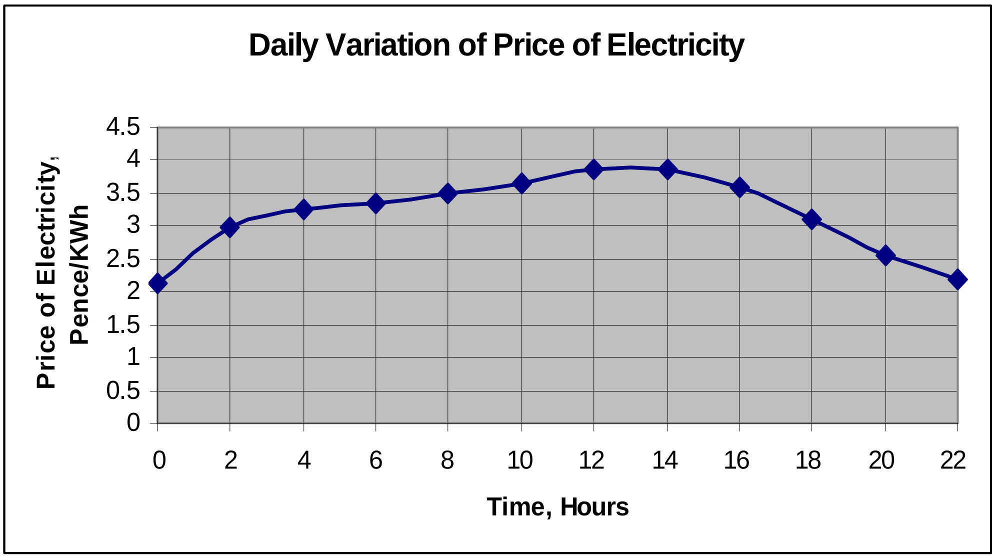


Figure 4-5 Variation of price of electricity of a typical day in the summer [92]

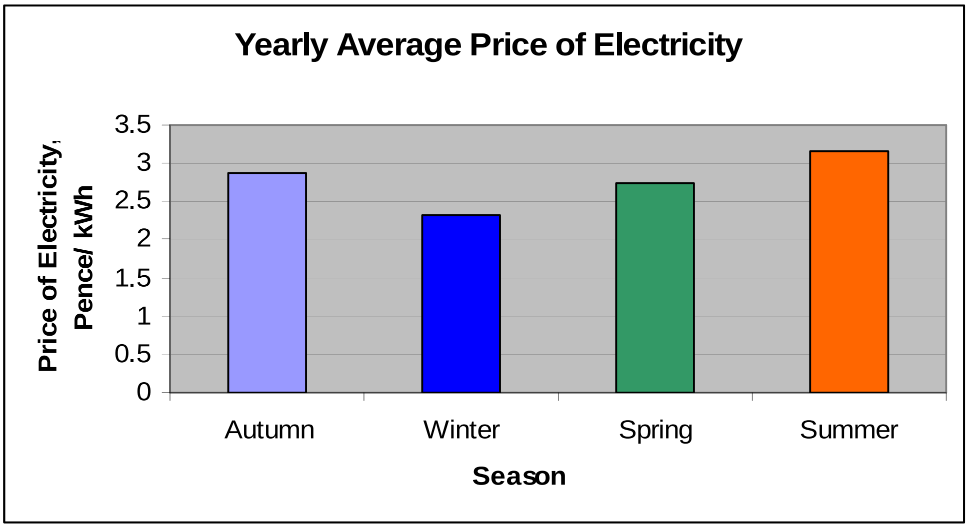


Figure 4-6 Yearly average variation of electricity price [92]

4.5 Total Revenue

Revenue is the total sum of money receivable for the sales of goods and services during a certain period of time. This also includes the total sales, exchange of assets; increase in owners equity and interest and this is calculated before any expenses can be subtracted.

$$TR = \int_T EP(t) \times P(t) dt \qquad [4.1]$$

Where:

TR – Total revenue over time T

$EP(t)$ - The projected price of electricity at time t

$P(t)$ - The electricity power output of the power plant at time t

4.5.1 Electricity Production and Price

The electric energy produced by the gas turbines is a source of income for the plant. Two assumptions are made, that the power produced is based on the power demand shown in figure 4-3 and the price per kWh is as shown in figure 4-5. For ease of calculation the total seasonal electricity produced is taken to be the sum of the electricity produced by each engine during a typical day for that season multiplied by 91.25 days [91].

$$\text{Yearly } EP = \sum_{i=1}^{n}\sum_{k=1}^{4} EP \times 12 \times 91.25 \qquad [4.2]$$

Where:

EP – Electricity produced

n – Number of units

k - Seasons

4.6 Life Cycle Assessment

As stated in section 4.3 life cycle or total ownership cost is the sum of the total fixed cost and the total variable cost. Fixed cost comprises the capital cost (cost of purchase, installation and transportation) and fixed operational and maintenance cost. Variable cost consists of fuel cost, operation, and maintenance and repair costs [65].

The summation of these costs is referred to as the operating costs. When the cost of the purchase of the gas turbine and auxiliary systems, known as the capital cost, and the cost of abandonment of the gas turbines and associated systems at the end of their lives, is added to the operating cost, the total cost is referred to as the life cycle cost or total ownership cost. The TP generated is the difference between the TR and the LCC. Thus lower the LCC and higher the revenue increases the TP.

Figure 4-7 shows that fuel costs make up over 70% of the total life cycle cost. Maintenance and repair costs will vary depending on the state of the engine at various times during its operating life, but contribute only about 11% of total costs. Figure 4-8 shows the functional tree representing the total ownership cost of electricity.

The main objective for the investigation in this part is to develop algorithms to predict the cumulative life cycle cost for industrial gas turbine power plant operating at different ambient conditions and loads.

The life cycle cost is the total cost of acquisition and ownership of a system over its life. To attain optimum lifecycle profitability, the impact of short-term generation and maintenance scheduling on long-term plant profitability needs to be considered at all times.

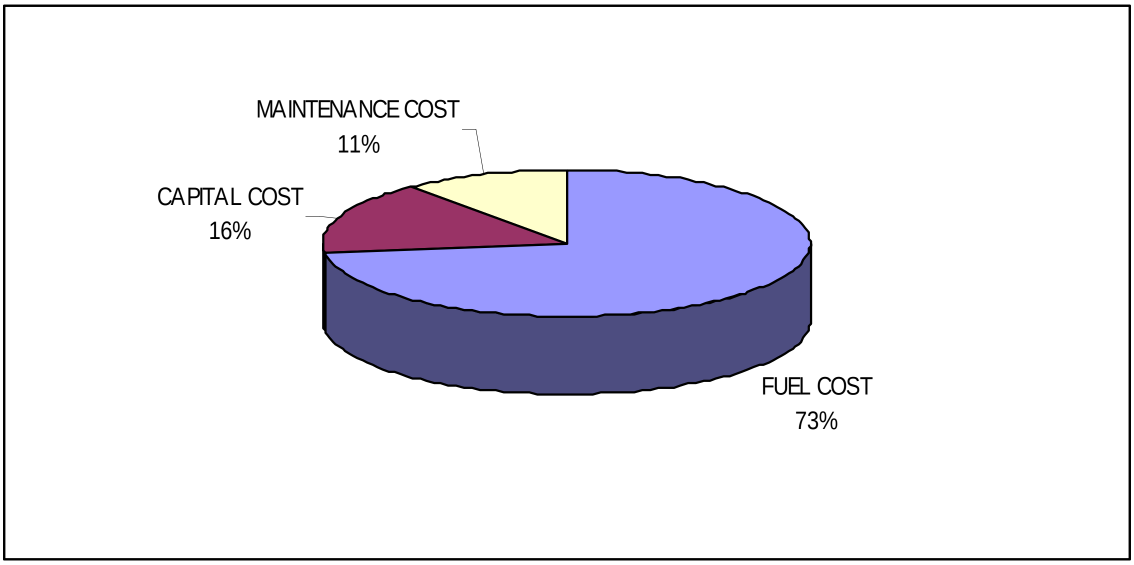


Figure 4-7 Life cycle cost brake down

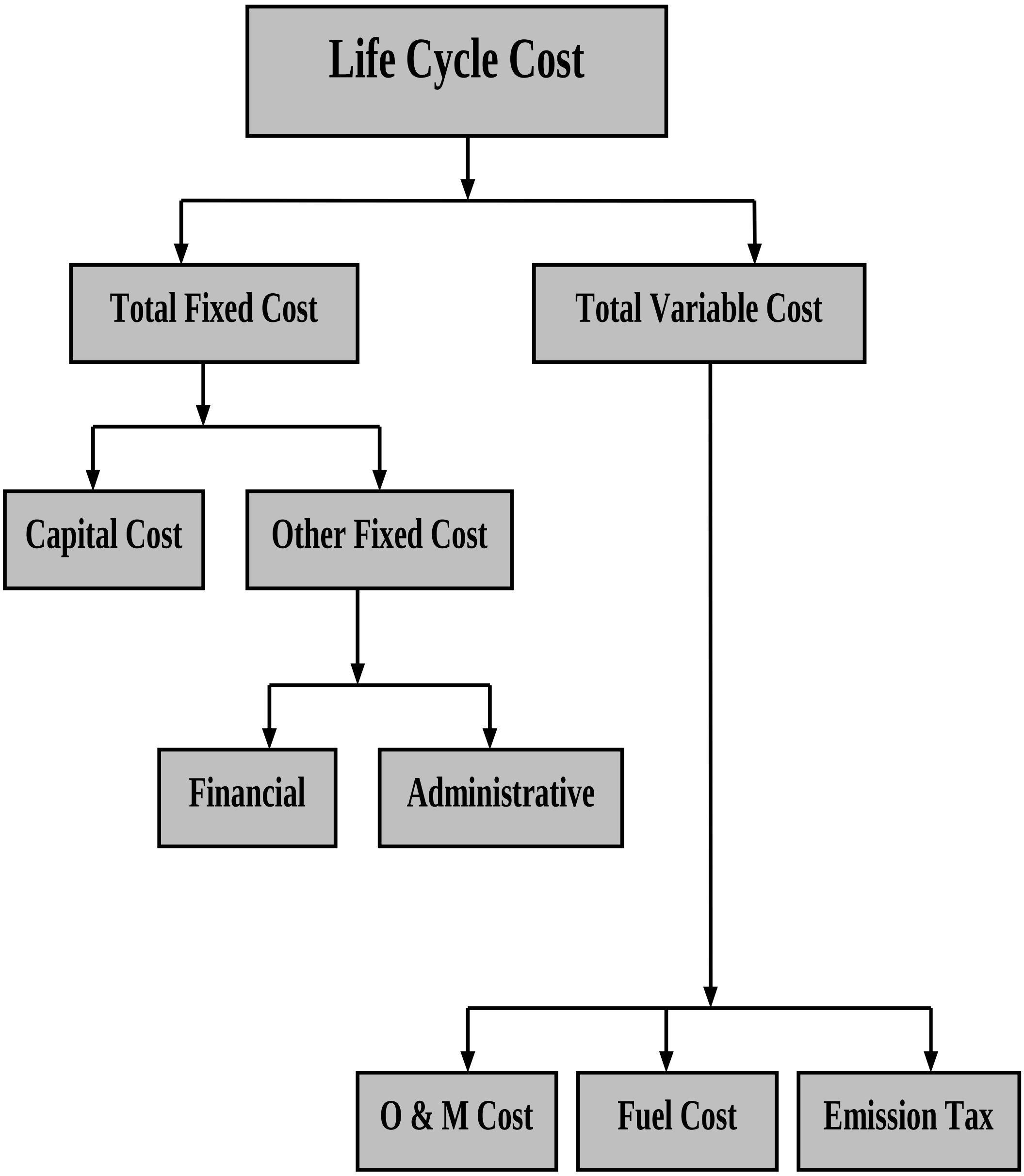


Figure 4-8 Functional tree for life cycle cost

4.6.1 Capital Cost Evaluation

The capital, initial or investment, cost will vary with the technology chosen for power generation because different technologies will have different operational efficiencies and capacities, with different costs in terms of £/Kw [65]. Table 3-1 gives an example for the capital cost breakdown for a medium power generation power plant.

4.6.1.1 Equipment Costs

Equipment costs will include price of purchase of the equipment, including any taxes, but other costs such as transportation to site or delays in delivery must also be considered.

4.6.1.2 Installation Costs

These include:

- Installation permits,
- Land acquisition and preparation,
- Building construction,
- Installation of equipment,
- Documentation and as built drawings.
- Grid connections, including reinforcement of local/national electricity networks
- First set of spare parts and any special tools needed for servicing and repair

Some of these costs may not be applicable, e.g. if the space is already available

4.6.1.3 Soft Costs (Project Engineering Cost)

Soft costs are the design and professional service fees incurred during the planning and development of a co-generation system. They will often be in the range of 15-30% of the equipment cost. Soft costs include:

- Architectural / engineering design fees.

- Construction management fees.
- Environmental studies and permitting costs.
- Special consultants and inspectors.
- Legal fees.
- Letters of credit.
- Training.

Other costs may be incurred e.g. interest paid during construction, bank fees, and debt insurance.

4.6.1.4 Unforeseen cost

Unforeseen cost is an extra cost which cannot be predicted. However, a contingency or allowance for unforeseen costs should be taken into consideration when estimating budgets. The earlier in the process the larger the contingency should be, possibly as much as 15%-20% of the total cost at the start of the design process falling to about 5% at the completion of the design process.

4.6.2 Investment costs breakdown

The initial decisions made concerning the plant and machinery to be purchased will strongly influence total power plant profit. Generally the larger the capacity of the units the less will be the capital cost of meeting a given load, also operating and maintenance costs could be less. However, operating such large units at off design conditions will be more expensive, and flexibility of use and availability will not be as good as for smaller units. These matters need to be considered in order to maximise the TP.

Vendors are usually eager to quote cost for purchasing equipment. It is more difficult, however to estimate the cost for installing the equipment at a specific site. Because installation costs can vary substantially from site to site, this study assumes installation costs can be subsumed a fixed cost.

Typically, capital costs vary from about (US)$150/kW (for larger schemes) to more than (US)$700/kW for very small units, depending on the choice of co-generation plant and auxiliaries required, see figures 4-9 and 4-10 and Appendix 3. Further details concerning prices levels will discussed later. Table 4-1 shows an approximate brake down of capital costs for medium power generation plant.

Type of Cost	% of Total
Gas turbine unit	55
Instrumentation, regulation and control	15
Auxiliary systems	5
Connection to grid	5
Civil work and/or acoustic enclosure	10
Installation and commissioning	5
Project costs	5
Total	100

Table 4-1 Brake down of capital costs for medium power generation power plant [70]

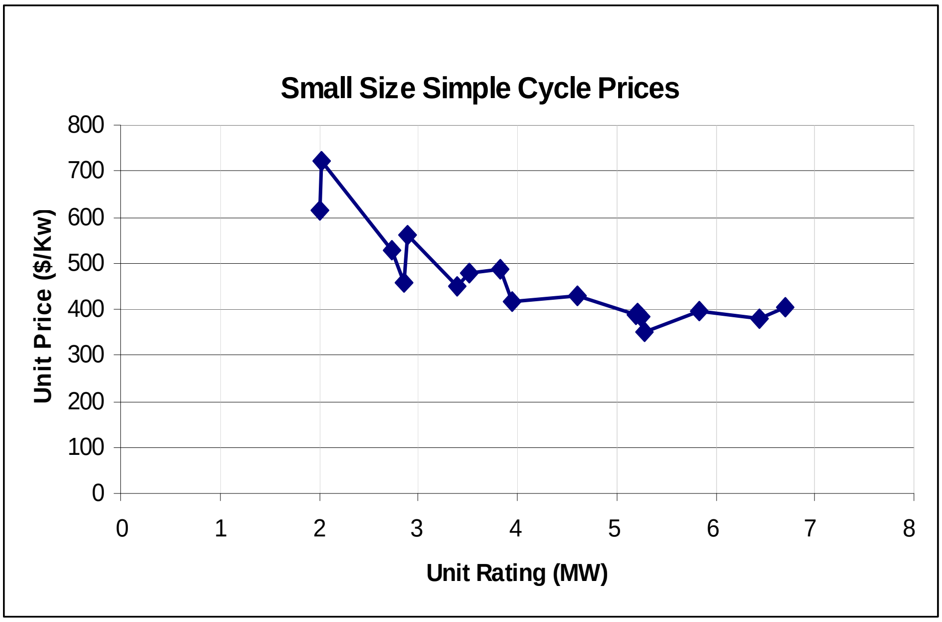


Figure 4-9 Small to Medium Size Simple cycle plants, [93]

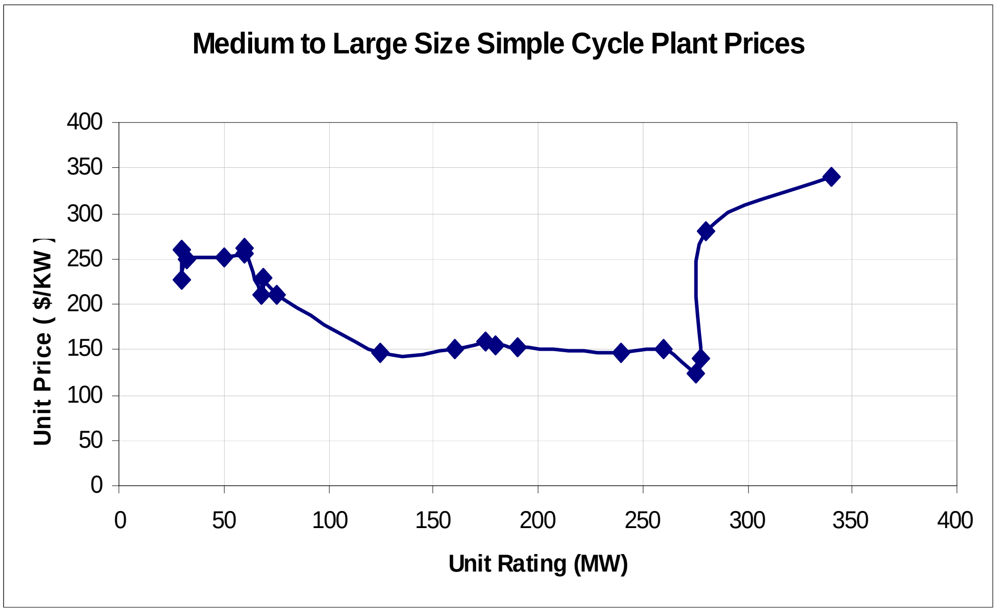


Figure 4-10 Medium to large size simple cycle plants, [93]

4.6.3 Fuel Cost and Consumption

The increase in the cost of fuel over the past years is a well recognised economic problem for users which is expected to worsen in the future. This problem has had a great impact on the operation of existing gas turbine facilities and on the design of new ones.

Generally the fuel cost is the most significant operation cost, which may reach 70% of the total operation cost as shown in figure 4-7 over a typical service life of 20 to 30 years. With the amounts of fuel cost involved in operating gas turbines, small percentage improvements in fuel consumption will bring substantial cost savings over the life of the plant as will be seen later.

Fuel contract prices have also been estimated in function of the official price of the market. This is again two different prices in this study but this time it is not in function of the time of the day but in function of the seasons [94].

It is very difficult to find which type of contract it exists between the electricity producer and the gas supplier (confidential data). Relative to this project the choice between day, week or month gas distribution contract is conceivable, however, because reasoning on the power plant earnings is on a day basis, it has been decided to have also day contract for the gas distribution. To summary, the power plant must plan the purchase of the quantity of gas it thinks to use the day after. For example, if during summer, the unit has to overcome heat waves as it was in 2003, the deciders, in case of electricity production will have to scheduled additional amount of gas to meet with the engine consumption requirements.

It has been clearly showed in figure 4-11 that very large breach exists between summer and winter prices. If prices of International Petroleum Exchange Market are followed, average amount of 20.68 p/term for the summer and 42.5 p/therm for the winter remain to be satisfactory values for the year 2004.

These two assessments have been reduced of 10 % of the initial value to introduce the commercial links which can exist between a power plant and a selling gas company trading very large quantities of gas all along the year. Therefore the two gas prices retained for the thesis are set up at

- 19 p/therm during the summer
- 38 p/therm during the winter

Figure 4-11 shows natural gas prices for years 2004 and 2005. It showed 2 types of prices from May to October what it can be called “summer price”, with low cost between 20 and 25 pence/term and from November to April, with high prices (“winter price”) varying from 30 pence/term to more than 40 pence/term. It can be noticed that July and August have the lowest natural gas prices and at the opposite January and February have the highest natural gas prices. This study will be carried out based on these prices (summer and winter).

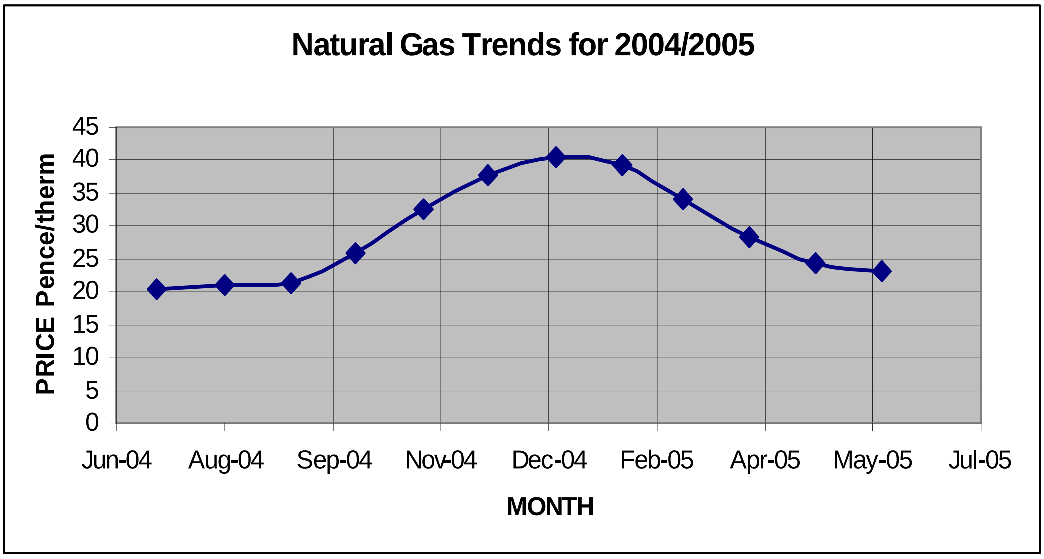


Figure 4-11 Natural gas prices trends for 2004 / 2005 [95]

Fuel Consumption:

$$Fuel\ \ Consumed = Fuel\ \ Flow \times 3600 \times Running\ \ Hours \qquad [4.3]$$

Then it should be converted to energy units called Threm as it is the unit for the natural gas price.

Knowing LHV of natural gas, the mass of fuel can be converted to Threm and the price can be found.

$$C_{FUEL}(T) = F_P \times Fuel\ \ Consumed = \left[\frac{LHV \times F_P}{105506}\right] \times Fuel\ \ Consumed \qquad [4.4]$$

C_{FUEL} (T) = cost of fuel for the period of time (T) (£/MJ)

F_P = Fuel Price, p/therm

Fuel consumed (fuel flow) given by Turbomatch and then the optimiser will use it in the objective function as an input data.

4.6.4 Operation and Maintenance Cost

Operation and maintenance costs are usually divided into fixed costs (staff, insurance, etc.) and variable costs of operation and maintenance (consumables spare parts, etc.).

Operation and maintenance costs are usually depend on factors such as quality of the operation and maintenance, type of fuel, operation cycle and operating environment. It is evident that the main influence on the maintenance costs and engine life is the condition of the turbine blades, which is significantly dependent on the engine rating.

The rating basically refers to turbine entry temperature TET, Gas turbines operating with high TETs usually have increased maintenance cost, due to creep phenomena appearing in the turbine section. These affect the blade creep life which is typically halved for every 20 ^{0}K increase in blade temperature or doubled for every 20 ^{0}K reduction in blade temperature near its rated operating point, [96]

Maintenance costs will invariably increase if heavier or dirty fuels are used or if the plant is situated in a dirty environment. In such circumstances frequent start ups are doubly disadvantageous and will substantially increase maintenance costs. In addition, they will increase thermal stresses which may also be affected by the rotational speed.

Typical fixed maintenance costs of gas turbine using natural gas is usually 2.0-2.5 £/MWh for larger plants (above 1MW) 2.0-3.2 £/MWh for small plants (under 1MW), [93].

Fixed maintenance and operation costs also can be expressed as 35% of the fuel cost.

O & M Costs = (35 x Fuel Cost) / 70 [4.5]

Remembering that fuel costs are about 70 % of the total operating expenses [96] it is quite possible that decisions made to reduce initial costs may subsequently lead to increased operation and maintenance costs, with an overall negative effect on the total economic performance of the project. Section 3.3.3 discussed reasons why the TET may be changed but in addition operating with heavier fuels or in a dirty (dusty) environment may also require a change in TET.

4.6.4.1 Creep Life Assessment

The usage of creep life of gas turbine blades will eventually require change out of the hot sections and forms a maintenance major cost and this will be discussed in this section of the project.

Creep can be defined simply as the tendency for a solid material to progressively deform when subject to long-term exposure to mechanical loads which are less than the yield strength. Creep is always more severe at high temperatures and in gas turbines creep causes slips in aerodynamics blades shape. The situation is complicated because gar turbine blades experience a whole range of temperature and loads during operation which may continue for prolonged periods of time [97].

Creep may be regarded as a four stages process, see figure 4-12. The first stage is pure elastic deformation.

The primary stage is the most common and represents the main stage for most materials subject to stress at low temperature. The strain rate is initially high, but slows with increasing strain due to work hardening.

Secondary creep: The creep strain rate is constant and a minimum for the material (due to the balance between the competing processes of strain, hardening and recovery).

Tertiary creep is a region where creep strain occurs at an accelerating rate even under constant temperature and constant load because of reduction in cross-sectional area produced by necking of the material or component.

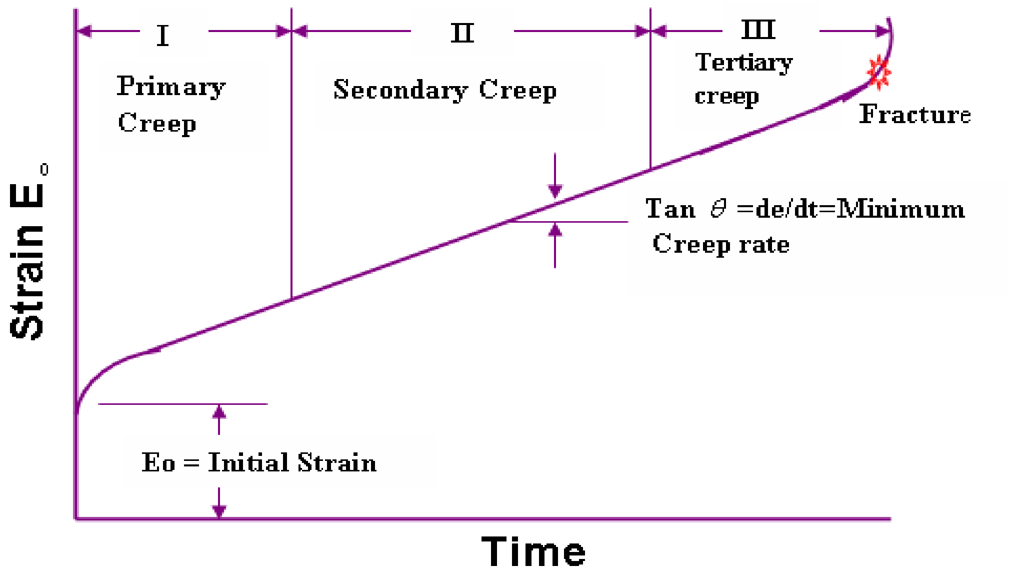


Figure 4-12 Creep stages process

4.6.4.1.1 Turbine Blade Metal Temperature

The metal temperature values were estimated using a reverse engineering process, starting by the Holland and Thake [98] method for the prediction of the cooling flow rates, and inverse path was taken to predict and built a thermal model which related the amount of cooling air with the metal temperature. In this method the percentage of cooling air was imposed and the blade metal temperature was obtained.

The Holland and Thake model includes both film and convective cooling. It is considered attractive because the cooling flow fraction (the ratio of the coolant mass flow rate to the mainstream mass flow rate) is expressed in terms of just five parameters, all of which are known, see table 4-2.

Factor	Symbol	Value
K_{cool}	**Cooling Flow Factor**	**0.045**
η_{int}	**Internal Cooling Efficiency**	**0.70**
ε_{ϕ}	**Film Cooling Effectiveness**	**0.40**
Bi_{met}	**Metal Biot Number**	**0.15**
Bi_{tbc}	**TBC Biot Number.**	**0.15**

Table 4-2 Thermal model parameters

The approach selected for estimating the blade metal temperature, will be the Holland and Thake method [98] cover in the opposite direction, this method is interesting because it expresses the cooling flow fraction in terms of a small number of parameters representative of a technology level.

The problem is to find the value of ε_0, for each blade row, which will ensure that the required value of μ is achieved. An expression for it is given by:

$$\mu = \frac{K_{cool}}{(1+b)} \left\{ \frac{\varepsilon_0 - \varepsilon_\phi \left[1-\eta_{\text{int}}(1-\varepsilon_0)\right]}{\eta_{\text{int}}(1-\varepsilon_0)} \right\} \quad [4.6]$$

Where the parameter b is given by:

$$b = Bi_{tbc} - \left(\frac{\varepsilon_0 - \varepsilon_\phi}{1-\varepsilon_0} \right) Bi_{met} \quad [4.7]$$

Where cooling technology parameters used in these equations has been showed in table 3.3.

4.6.4.1.2 Blade cooling effectiveness

$$\varepsilon_0 = \frac{T_g - T_{met}}{T_g - T_{c1}} \quad [4.8]$$

$$T_{met} = T_g - \varepsilon_0 (T_g - T_{c1}) \quad [4.9]$$

Where

- T_g is the mean temperature of the mainstream gas,
- T_{c1} is the total temperature of the coolant entering the blade passages, and
- T_{met} is the allowable external surface metal temperature, assumed constant over the blade.

T_{met} is the target of this calculation, because with it, and T_g and T_{c1} achieved by the engine simulation, the blade surface metal temperature will be known [99].

4.6.4.1.3 Calculation results

The engine that will be simulated in this present study, has a 7.5% of cooling flow fraction μ , but it is necessary to realise that the most of this air will be introduce by the first stage of NGV and the rest by the first blade stage (or high pressure turbine blades) principally. In this present study will be imposed that a fraction of 3/7.5 of this cooling air is injected inside the core air flow by the first blade stage and the other 4/7.5 air cooling fraction by the first NGV stage, assuming that the air that could be introduced by the next NGV or blade stages is negligible.

Different points of cooling fraction have been evaluated starting from 7.5% for a design situation and finishing on 0% (no cooling air). The study will be focused on the high pressure turbine blade, since that is the part of the turbine that will suffer the worse conditions from the point of view of the creep assessment; it means a worse combination of high temperature level with high stress levels

The next table (4-3) , presenting the output of the previously described approach for each of the cooling flow fraction, the value of the blade cooling effectiveness is calculated, using for that the previously listed current values of the blade cooling parameter that take part in such thermal model [100]

Cooling Flow Fraction μ	***HPT* Blade** μ	ε_o
7.50%	**3.5%**	**0.63**
6.25%	**2.91%**	**0.59**
5.00%	**2.33%**	**0.55**
3.75%	**1.75%**	**0.50**
2.50%	**1.16%**	**0.42**
1.25%	**0.58%**	**0.30**
0.00%	**0%**	**0.09**

Table 4-3 Blade cooling effectiveness outputs

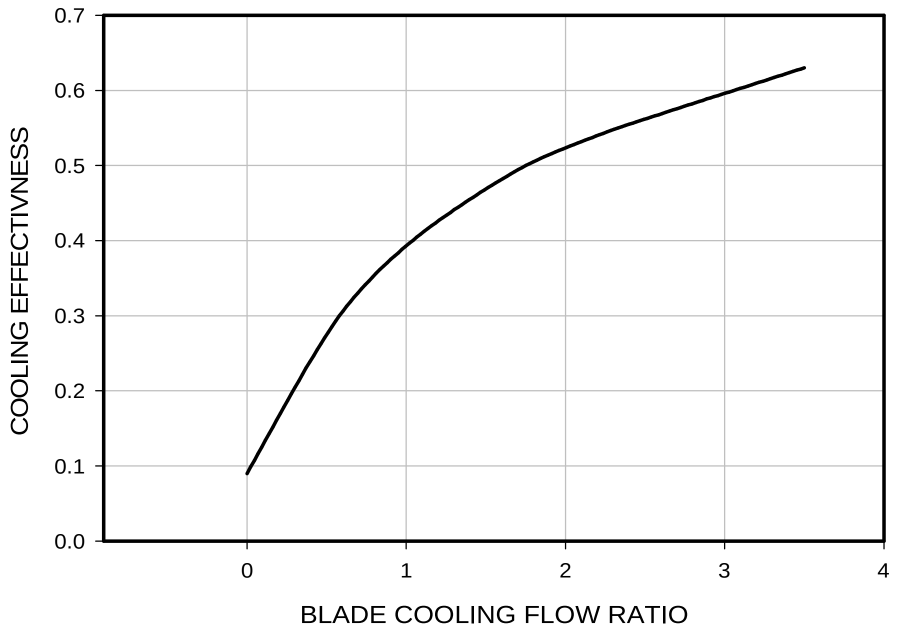


Figure 4-13 Thermal model parameters validation

Figure 4-13 shows the results of effectiveness variation against the cooling flow. It is possible to see that for the region near of the design point blade cooling ratio, (3.5% for the first blade row) there is a big difference in the outputs. When the theory that keeps constant the parameter is assumed, it still has to match the reality with a bigger accuracy.

4.6.4.1.4 Creep Life Cost

In this study turbine blade creep life has been considered as the major variable maintenance cost that is related to the operating and ambient conditions.

The complex geometry and anisotropy of blade components and the multi axial stress states they experience, has led to complex relationships between loads, temperature and deformation. It is necessary a creep life consumption model, which optimize the efficiency in the design of turbine blades, with respect to both preventing failure and avoiding over design.

The basic model selected for calculating the creep life consumption, caused by a given pattern of loading called time-temperature parametric method. The most popular is the Larson-Miller method [99]

4.6.4.1.5 The Larson Miller parameter

The Larson-Miller parameter (LMP) is a time temperature parametric method for predicting the lifetime of material as a function of temperature. It is a technique which has shown great accuracy. The method correctly assumes that at constant strain an increase in operating temperature will reduce the time to reach a particular creep state. Problems arise in the algorithm that relates these variables:

Larson miller expression

$$LMP = f(\sigma) = 1.8 \times T(^0K) \times 10^{-3} \times \left(\log t_r + C\right) \qquad [4.10]$$

$$tr = 10^{\left(\frac{1000 \times LMP}{T}\right) - 20} \qquad [4.11]$$

Where *LMP* is a function of the stress level,

T, is the metal temperature in Kelvin,

t_r is the time to failure and

C is a constant, which is often taken as 20 in industrial applications.

This expression can be understand as that any combination of temperature and time which give equivalent values of *LMP*, will also give equivalent values of creep stress. The output of this simple expression will be the time to rupture that will be necessary at temperature and stress conditions that are being studies.

To use the expression, it is necessary to resort to some test data, plotted as a curve that relate the P parameter with the stress level, such curves are know as master curves. This curves are function of the material selected, in this case it was the INCO alloy HX.

$$\text{Blade stress} = \frac{mass * raduis * \left(\frac{2\pi * speed}{60}\right)^2}{cross\,\sec tion.area} \qquad [4.12]$$

4.6.4.1.6 Cumulative Creep Estimation

In practice, turbine blades experience a wide range of stresses and temperatures during start-up and shut-down and different loads. Creep strain accumulates during these stages but assuming a 'worst case' scenario would incur an over-design penalty and lower payloads.

An alternative approach is to adopt the 'Life Fraction Rule', which assumes that rupture will occur when the sum of all the fractions of the rupture life at different stress/temperature combinations equal unity [14].

Fraction of life consumed = $\left(\frac{t_1}{t_{r1}} + \frac{t_2}{t_{r2}} + \frac{t_3}{t_{r3}} + \ldots\ldots\ldots\ldots\right) = \sum \frac{t_i}{tr_i}$ [4.13]

Where t_i is the time spent at a particular stress/ temperature combination where the time to rupture is tr_i .

In this study the cycle of the operating has been assumed to be 12 different segments of time a day in each season each segment represents 2 hours, so it is 91.25 similar days and each 2 hours during a day has its power demand and ambient condition.

Fraction of life consumed =

$$\begin{bmatrix} \left(\frac{1}{tr_{1S}} + \frac{1}{tr_{2S}} + \ldots\ldots + \frac{1}{tr_{12S}}\right) + \left(\frac{1}{tr_{1A}} + \frac{1}{tr_{2A}} + \ldots\ldots + \frac{1}{tr_{12A}}\right) + \\ \left(\frac{1}{tr_{1W}} + \frac{1}{tr_{2W}} + \ldots\ldots + \frac{1}{tr_{12W}}\right) + \left(\frac{1}{tr_{SP}} + \frac{1}{tr_{SP}} + \ldots\ldots + \frac{1}{tr_{12SP}}\right) \end{bmatrix} \times 91.25 \qquad [4.14]$$

$$= \left[\left(\sum_{i=1}^{12} \frac{1}{tr_{iS}} + \sum_{i=1}^{12} \frac{1}{tr_{iA}} + \sum_{i=1}^{12} \frac{1}{tr_{iW}} + \sum_{i=1}^{12} \frac{1}{tr_{iSP}} \right) \times 91.25 \right] \qquad [4.15]$$

Where:

tr_{1S} is the life time at hour number one in a typical summer day

tr_{1A} is the life time at hour number one in a typical Autumn day

tr_{1W} is the life time at hour number one in a typical Winter day

tr_{1SP} is the life time at hour number one in a typical Spring day

Time to failure = $\frac{8760}{Fraction\ of\ Life\ Consumed}$ [4.16]

Apply safety factor of 0.6 gives a life reduction 60%

By using the following equation the cost due to the creep can be estimated according to the cumulative calculation as above.

$$Creep\ Life\ Cost = \left[\left(\frac{Unit\ Expected\ Life\ (hours)}{Time\ to\ Failure\ (hours)} \right) - 1 \right] \times Cost\ of\ replacment \quad [4.17]$$

In this study the material for the turbine blades has been selected to be INCO alloy HX as mentioned above, for more realistic and due to the geometry and blade turbine material some times will not be available (manufacturer confidential data), the other option is to assume a linear correlation between an over fired engine TET and the maintenance costs parameters which appear in the life cycle equation. This assumption has been plotted based on 2 different oilfields in Libya [101].

20 ^{0}K augmentation in TET = 22% extra in Maintenance costs [4.18]

4.7 Emission Cost

4.7.1 CO_2 Emissions Calculation

Assuming complete combustion in the presence of excess air, which is reasonable, CO_2 emissions depend directly on the type, quality and quantity of the fuel used. The CO_2 emitted may be calculated using the following equation:

$$C_x H_y + nO_2 \Rightarrow n_1 CO_2 + n_2 H_2 O \quad [4.19]$$

Where *x, y* are the carbon – hydrogen atomic ratio of the fuel.

The equation states that 1 mole of the fuel will react with *n* moles of O_2 to produce n_1 moles of CO_2 and n_2 moles of H_2O. Performing a molar balance:

$$\begin{aligned} n_1 &= x \\ n_2 &= 0.5y \\ n &= n_1 + 0.5n_2 = x + 0.25y \end{aligned} \quad [4.20]$$

Substituting n, n_1 and n_2 into equation 14:

$$C_xH_y + (x+0.25y)O_2 \Rightarrow xCO_2 + 0.5yH_2O \qquad [4.21]$$

Therefore, 1 mole of fuel will produce x moles of CO_2. But 1 mole of fuel will weight:

x x 12 + y x 1 kg [4.22]

and 1 mole of CO_2 will weight 44.01 kg.

Therefore:

1 kg of fuel = 44.01x / (12x+y) kg of CO_2 [4.23]

Or

1 kg of fuel = 44.01 / (12x+y/x) kg of CO_2 [4.24]

Values of 44.01 and 12 in the previous equations represent the molecular weight and atomic weight of CO_2 and carbon, respectively.

In terms of mass, 1 Kg of methane produces 2.75 Kg of carbon dioxide due to their different molecular weights, (molecular weight of CO_2 = 44.01g/mol and molecular weight of CH_4=16.04 g/mol)

$$m_f = \frac{E}{\eta \times FCV} \qquad [4.25]$$

$$eCO_2 = \frac{44.01}{12} \times C \qquad [4.26]$$

$$mCO_2 = eCO_2 \times m_f \qquad [4.27]$$

The tax liability for operating a gas turbine can be derived from the following equation:

$$Carbon\ Tax = mCO_2 \times Tax\ \ Rate\ \ £/MW/hr \qquad [4.28]$$

$$Carbon\ Tax = (mCO_2 \times Tax\ \ Rate\ \ ) \times 2\ \ £/Seg \qquad [4.29]$$

Where:

Seg – 2 hours (each day includes 12 segment of time each one 2 hours)

m_f - Is the mass fuel consumption

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

$m\ CO_2$ Mass of emitted, CO_2

E Useful energy produced by the system,

$e\ CO_2$ Emissions of CO_2

per unit mass of fuel (e.g. kg CO_2 /kg fuel),

η - Efficiency of the system, based on the lower heating value of fuel,

FCV is the fuel calorific value (lower)

4.8 Financial Considerations

4.8.1 Depreciation Cost

For utility plant, generation, transmission, or distribution, the annual book depreciation rate is a uniform value equal to the divided by the book life. The book life varies for typical plant and is generally 25-35 years. For utility plant, the depreciation rate are may range from 0.040 to 0.028.

Book depreciation, DB = 1/Book life [4.30]

4.8.2 Capital Structure

To derive equation for the fixed charge rate, the utility income statement, income tax statement, and cash report must be examined. Table 4-4 presents a utility income statement; table 4-5 shows a tax statement and table 4-6 a cash report.

Income Statement (year i)	
Total Revenue	TR_i
Less Expenses	
Production Costs	PC_i
Depreciation Book	DB_i
Income Taxes	IT_i - ITC_i
Deferred Income Taxes	$DFIT_i$
Deferred Investment Tax Credit	$ATC_i * NORMITC$
Amortisation of Investment Tax Credit	$AITC_i * NORMITC$
Ad Valorem Taxes and Insurance	AV_i
Operating Incom	$OPIN_i$
Interest on Dept	INT_i
Net Income	NI_i

Table 4-4 Income statement

$$OPIN_i = TR_i - PC_i - DB_i - (IT_i - ITC_i) - DFIT_i - AV_i - ATC_i * NORMITC + AITC_i * NORMITC \quad [4.31]$$

$$NI_i\ TR_i - PC_i - DB_i - (IT_i - ITC_i) - DFIT_i - AV_i - ITC_i * NORMITC + AITC_i * NORMITC - INT_i \quad [4.32]$$

$$DFIT_i = t * (DT_i - DBT_i) * NORMDEPR \quad [4.33]$$

Where:

$NORMITC$ = 1 if the investment tax credit is normalised

0 if the investment tax credit is flow-through

t – Income tax rate

DT_i – Tax depreciation in year i

DBT_i – Book depreciation rate used in tax calculations in year i

$NORMDEPR = 1$	if tax depreciation savings are normalised
0	if tax depreciation savings are flow-through

Income Tax Statement (year i)	
Total Revenue	TR_i
Less Deducted Expenses	
Production Costs	PC_i
Tax Depreciation	DT_i
Ad Valorem Taxes and Insurance	AV_i
Interest on Dept	INT_i
Taxable Income	TI_i

Table 4-5 Income Tax Statement

This implies:

$$TI_i = TR_i - PC_i - DT_i - AV_i - INT_i \quad [4.34]$$

$$\text{Income Tax } (IT_i) = t * TI_i \quad [4.35]$$

Cash Report (year I + 1)	
Cash (start of year)	$CASH_i$
Cash from operation	
Net Income	NI_i
Depreciation Book	DB_i
Deferred Income Taxes	$DFIT_i$
Cash Flow	CF_i
Discount Rate	DR_i
Discounted Cash Flow	DCF_i
Cumulative Discounted Cash Flow	$CDCF$

Table 4-6 Cash report

4.8.3 Cash Flow

Cash flow has been defined by Allen, D [111] as the actual movement of cash in and out of a business. Cash flow in (positive cash flow) is cash received, and cash out (negative cash flow) is cash paid out. The difference between these tow flows is termed the Net Cash Flow. In our concern, cash flow includes accounting parameters such as depreciation.

$$\text{Cash Flow} = NI_i + DB_i + DFIT_i \qquad [4.36]$$

$$\text{Discounted Cash Flow } (DCF_i) = CF_i \cdot DR_i \qquad [4.37]$$

$$\text{Cumulative Discounted Cash Flow (CDCF)} = \Sigma\, DCF_i \qquad [4.38]$$

4.9 Maximum Profit Optimisation Model

A generic procedure is developed to implement the integrated operational modelling environment. Models for both economic and technical factors have been developed, and procedures to model system level metrics such as cost of fuel, emission taxes, maintenance and operations cost, returns paid to investors and revenue

As a result for the above generic procedure, cumulative for all type of costs (cost of fuel, emission taxes, maintenance and operations cost, etc) has been produced, based on power plant performance, price of electricity and price of the fuel.

To evaluate the economic performance of a power plant operating in a deregulated market an objective function, such as net revenue or profit, is defined. With gas-fired power plant selling electricity, spark spread (SS) is sometimes used as the objective function for operational optimisation instead of net revenue or TP. The key element defining the profit is the gross revenue obtained from selling of electricity. This has to be modified by including a number of costs (of fuel, of operations, of maintenance) and other financial aspects. The following equation shows the relationship:

$$TP = TR - LCC \qquad [4.39]$$

For a given period of time (T) the TP is:

$$TP = TR_T - LCC_T \qquad [4.40]$$

Where:

TR = No of units of electricity sold x mean price per unit [4.41]

Here TR is total revenue obtained from the sale of selling electricity.

TP – Total Profit

4.9.1 Total Revenue

TR obtained from the sale of electricity over time period T when both the price and production vary is:

$$TR = \int_T EP(t) \times P(t) dt \qquad [4.42]$$

Where *EP (t)* is the price of electricity at time *t* (£/MWh), and *P(t)* is the electricity being produced by the power plant at time *t,* (MW).

The spread between the market value of the fuel gas and that of the electricity obtained by burning the gas becomes wider as the price of electricity gets higher. As the efficiency of electrical generation system increases, the spread between the market value of the gas and that of power derived by burning the gas also becomes wider [102]

The spark spread is determined using Equation [4.43]

$$SS\,(t) = 10 \text{ x } EP\,(t) - C_{FUEL}\,(t) \text{ x } HR\,(t)\,/1000 \qquad [4.43]$$

Where, as stated above, EP(t) is the instantaneous price of electricity, £/MWh

HR(t) is the instantaneous heat rate of the system, MJ/MWh

C_{FUEL}(t) is the instantaneous price of fuel, £/MJ, and

SS (*t*) is the instantaneous spark spread £/MWh

A cumulative SS for time period T can be calculated along the time line of operation. The cumulative SS is given by Equations 4-44 and 4-45. These give the cumulative SS as the the difference between the gross revenue obtained from the sale of electricity during time T, less the cost of cost of fuel during the same time period.

$$SS\,(\mathrm{T}) = \mathrm{TR}\,(\mathrm{T}) - C_{FUEL}\,(\mathrm{T}) = \int_T SS(t) dt \qquad [4.44]$$

$$= \int_T \left(10 \times EP(t) - C_{FUEL}(t) \times HR(t)/1000\right) dt \qquad [4.45]$$

This chapter has introduced a systematic approach with which to evaluate power plant economics. The components of the total profit equation has been investigated and then the profit equation derived.

4.9.2 Life cycle Cost

Life Cycle Cost = Capital Cost + Fuel Cost + Operating and Maintenance Cost + Emission Tax + Financial Cost (returns paid to investors)

$$\text{LCC (T)} = C_{CAP}\text{(T)} + C_{FUEL}\text{(T)} + C_{OM}\text{(T)} + C_{EMI}\text{(T)} + C_{FIN}\text{(T)} \qquad [4.46]$$

Where:

C_{CAP} (T) - Capital cost at time T

C_{FUEL} (T) - Cost of fuel during time period T

C_{OM} *(T)* - Operations and maintenance cost.

C_{EMI} *(T)* - CO_2 Tax during time period T

C_{FIN} *(T)* - Financial cost (insurance, Tax, and Interest expenses)

4.9.3 Capital cost

This study using real engines already available in the market, hence the cost of the engines will be known, but the other costs need to be add to this cost such as the cost of installation as shown in table 4-1, generally the total of these costs is approximately 80% of the units cost.

$$C_{CAP} = \textit{Unit Cost} + 0.8 \cdot \textit{Unit Cost} \qquad [4.47]$$

In this study, for new power plant selection part the capital cost has been divided to equal instalments over the expected useful life for the project, these instalments will be paid to the lender with an interest rate

$$TCC = C_{CAP} + (\textit{Interest Rate} \cdot C_{CAP}) \qquad [4.48]$$

$$PCC = [C_{CAP} + (\textit{Interest Rate} \cdot C_{CAP})]/t \qquad [4.49]$$

Where:

TCC - Total capital cost

PCC - Period Capital Cost

t – Instalment Time

4.9.4 Fuel Cost

$$C_{FUEL}(T) = F_P \times Fuel\ \ Consumed\ = \left[\frac{LHV \times F_P}{105506}\right] \times Fuel\ \ Consumed \qquad [4.50]$$

$$Fuel\ \ Consumed = Fuel\ \ Flow \times 3600 \times Running\ \ Hours \qquad [4.51]$$

C_{FUEL} (T) = cost of fuel for the period of time (T) (£/MJ)

F_P = *Fuel Price, p/therm*

Fuel consumed for each segment of time during the day is given by Turbomatch and then the optimiser will use it in the objective function as an input data.

4.9.5 Operating and Maintenance Cost

As mentioned earlier the cost of operating and maintenance is divided to fixed cost and variable cost, typically the fixed maintenance and operating cost for gas turbine using natural gas is 2.2 – 2.8 £/MWh for engines with more than 1 MW output power.

$$F_C = EP.\ C_f,\ \ £/MWh \qquad [4.52]$$

F_C - *Fixed Cost*

EP – Output Power, MW

In this study the variable cost mainly comes from the maintenance and repair of the turbine blades due to the creep, and also the variable cost affecting by washing and cleaning of the compressor which can be added to the fixed cost.

$$Creep\ \ Life\ \ Cost = \left[(\frac{Unit\ \ Expected\ \ Life\ \ (hours)}{Time\ \ to\ \ Failure\ \ (hours)}) - 1\right] \times Cost\ \ of\ \ replacment \qquad [4.53]$$

The creep life cost over a specific period of time is equal the creep life cost divided by that period

$$Creep\ life\ cost\ (t) = creep\ life\ cost\ /\ t \qquad [4.54]$$

Cost of turbine blades replacement depends on the type and size of the engine and also can be affected by the maintenance and repair related to this replacement.

As a rules of thump for the cost of the maintenance due to turbine blade creep problem can be about 4% of the total cost for the engine.

$$V_C = Creep\ Life\ Cost \qquad [4.55]$$

V_C – *Variable Cost*

$$C_{OM} = F_C + V_C \qquad [4.56]$$

4.9.6 Emission Tax

According to equation [33], the daily cost due to carbon tax is equal:

$$Carbon\ Tax = \sum_{t=1}^{t=12} (mCO_2 \times Tax\ Rate) \times 2\ \text{£}/Day \qquad [4.57]$$

And the following equation shows the seasonally and yearly carbon tax

$$Carbon\ Tax = \left[\sum_{t=1}^{t=12} (mCO_2 \times Tax\ Rate) \times 2 \right] \times 91.25\ \text{£}/Season \qquad [4.58]$$

$$Yearly\ Carbon\ Tax = \sum_{Sea=1}^{Sea=4} \left[\sum_{t=1}^{t=12} (mCO_2 \times Tax\ Rate) \times 2 \times 91.25 \right] \text{£}/Year \qquad [4.59]$$

4.9.7 Financial Cost

$$C_{FIN} = IT + BD + INT \qquad [4.60]$$

$$IT = tr\ .\ TI \qquad [4.61]$$

$$TI = TR - PC - DT - AV - INT \qquad [4.62]$$

Where:

IT – Income Tax

BD – Book Depreciation

tr – Tax Rate

TI – Taxable Income

TR – Total Revenue

PC – Production Cost

DT – Depreciation Tax

AV - Ad Valorem Taxes and Insurance

INT – Interest Expenses

See tables 4-4, 4-5, and 4-6

4.9.8 Objective Function (Total Profit)

Based on the ambient conditions, operating conditions, and economic aspects as discussed earlier, the profit equation for one operating day can be stated as:

$$\left[\sum_{t=1}^{t=12}\left\{\left(E_{(t)} \times EP_{(t)} - (C_{cap(t)} + C_{Fuel(t)} + C_{OM(t)} + C_{emi(t)} + C_{Fin(t)})\right)\right\}\right] \quad [4.63]$$

For winter season:

$$91.25\left[\sum_{t=1}^{t=12}\left\{\left(Ew_{(t)} \times EPw_{(t)} - (Cw_{cap(t)} + Cw_{Fuel(t)} + Cw_{OM(t)} + Cw_{em(t)} + Cw_{Fin(t)})\right)\right\}\right] \quad [4.64]$$

For spring season:

$$91.25\left[\sum_{t=1}^{t=12}\left\{\left(Es_{(t)} \times EPs_{(t)} - (Cs_{cap(t)} + Cs_{Fuel(t)} + Cs_{OM(t)} + Cs_{em(t)} + Cs_{Fin(t)})\right)\right\}\right] \quad [4.65]$$

For summer season:

$$91.25\left[\sum_{t=1}^{t=12}\left\{\left(Esu_{(t)} \times EPsu_{(t)} - (Csu_{cap(t)} + Csu_{Fuel(t)} + Csu_{OM(t)} + Csu_{em(t)} + Csu_{Fin(t)})\right)\right\}\right] \quad [4.66]$$

For autumn season:

$$91.25\left[\sum_{t=1}^{t=12}\left\{\left(Ea_{(t)} \times EPa_{(t)} - (Ca_{cap(t)} + Ca_{Fuel(t)} + Ca_{OM(t)} + Ca_{em(t)} + Ca_{Fin(t)})\right)\right\}\right] \quad [4.67]$$

Therefore the power plant profit (*TP*) over 1 year is given by:

$$91.25 \left[\sum_{S=1}^{S=4} \ \sum_{t=1}^{t=12} \left\{ \left(E_{(t)} \times EP_{(t)} - \left(C_{cap(t)} + C_{Fuel(t)} + C_{OM(t)} + C_{emi(t)} + C_{Fin(t)} \right) \right) \right\} \right] \quad [4.68]$$

5 GENETIC ALGORITMS

5.1 Background on Genetic Algorithm

GAs are search algorithms inspired by the natural selection and evolution theory proposed by Charles Darwin. The GA is an iterative population-based algorithm where each iteration represents a generation. It transforms a population of individuals (each of which are evaluated and given a fitness value via an objective function) into a new generation, using analogs of naturally occurring genetic operations such as cross-over and mutation. The fitness measure will vary with the problem but normally lies between 0 and 1.

Traditionally a population of individual solutions covering the entire range of possible solutions is generated randomly. All the individuals in the search space are encoded, using a problem specific representation scheme, as a fixed length character string or other mathematical object. The GA usually manipulates individuals as binary-coded strings (this string is likened to a chromosome) and attempts to find the best solution to the problem by genetically breeding the individuals in the population over a number of generations.

A precondition for solving a problem with GAs is that the representation scheme meets the sufficiency requirement that is the representation is capable of expressing a solution to the problem In addition, before initiating a GA to solve a problem using fixed length strings there are four preparatory steps:

- The representation scheme
- The measure of the fitness of individuals
- The parameters and variables for controlling the algorithm
- Designation for terminating a run or arriving at a successful conclusion.

Conventional GAs begin with a determination of string length. An important feature of the GA is the mapping between potential solutions and binary representation of the individuals that occupy the search space of the problem.

.

The fitness measure drives the evolutionary process. It assigns a fitness value to each character string (individual) it encounters in the population, the smaller the proportion of the population selected the faster the algorithm can move from generation to generation, but the more generations that are required before a solution is reached. The number of individuals in a generation can vary from tens to thousands and the number of generations required before a solution is reached can be many thousands. Of course, the fitness measure should be capable of evaluating any of the character strings that it encounters in any generation of the population.

Every run of the GA necessitates that a termination criterion be specified. The best way to exit the GA is after a set number of generations after which the program stops. Also required is a method of designating the result. The present code, as many others, includes a restart feature; see below, so that if the search has not been successful and failed to attain an optimum within the specified maximum number of generations, the process can be restarted without any loss of information.

Operating a GA on fixed-length character strings requires three steps:

- Randomly creation of an initial population of individuals with fixed length character strings
- Iteratively repeat the following instructions (subsets) on the population until the termination criterion has been met:
  a) Using the fitness measure assign a fitness value to each individual in the population.
  b) Select individuals from the population with a probability based on fitness. Create a new population by applying the following three genetic operations to the selected individuals.
     - Reproduction by simply copying an existing individual into the new population.
     - Create a new individual from an existing individual by randomly mutating the string, usually at a single randomly chosen position.

- Create new strings from two or more existing strings by genetically recombining substrings using the crossover operation at randomly chosen crossover points.

➢ Designate the best individual as the result of the genetic algorithm for the run. This individual may or may not represent a solution or an approximate solution to the problem.

Figure 5-1 is a pictorial description of the above steps. The corresponding flow chart is presented in figure 5.2.

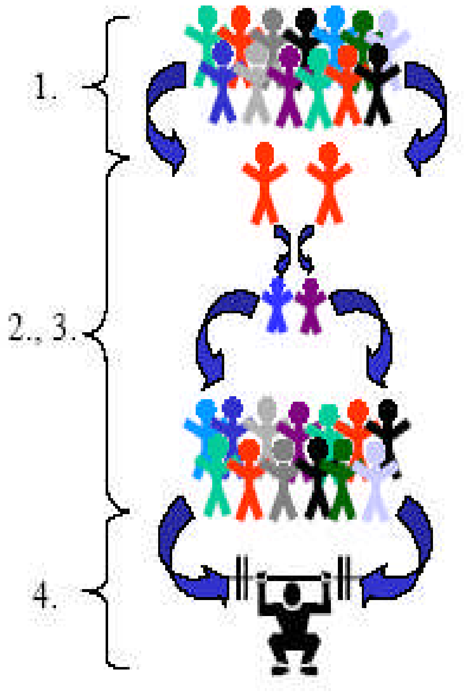


Figure 5-1 Illustration of a simple Genetic Aalgorithm

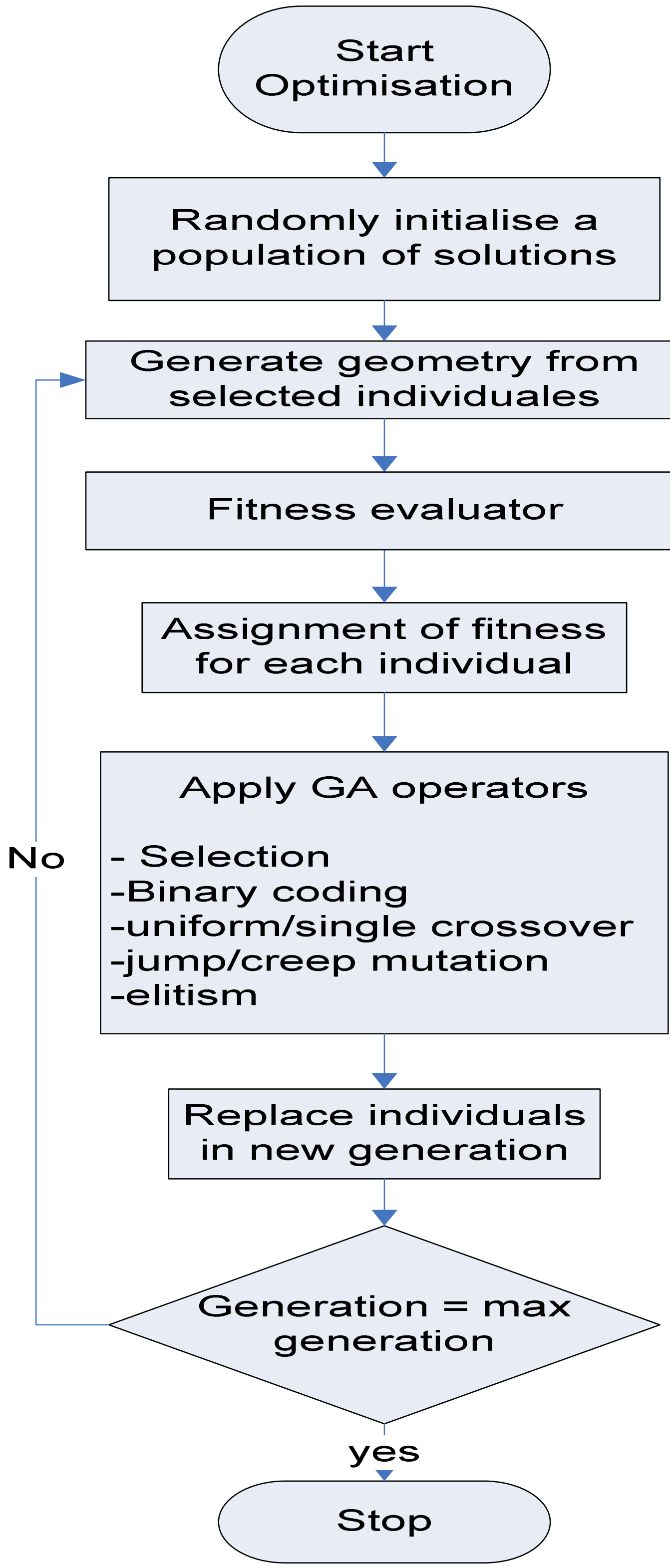


Figure 5-2 Genetic Algorithm flow chart

For a search technique to be "ideal" for multi-dimensional engine modelling it should possess a number of characteristics. In particular, it should be able to cope with:

- Surfaces with only one global maximum or minimum in the presence of many local maxima and minima (multi-modal),
- Surfaces with discontinuities (ill-behaved),
- High-dimensionality and high parameter resolution

To be successful GA search techniques must also be able the following questions:

- Will the initial (random) population distribution affect convergence?
- What is the optimum number of individuals in a population?
- How many generations (iterations) will be required before convergence?

These questions have been investigated by Carroll [103] who showed that GAs are an excellent compromise between globality, flexibility and convergence.

5.1 The Genetic Algorithms Method

This section outlines and illustrates the working of GAs in the solution of optimisation problems including: binary coding, elitism functions, function evaluation, and selection and crossover operators.

5.1.1 Genetic Algorithm Operators

The description of GA operators and their impact on efficiency of optimisation is based on the work of Carroll [104]. Initially, for this project, the best coding, selection and mutation schemes were not known, and a number of alternatives were examined to identify the most promising. This project investigated the use of either a floating point or a binary coding scheme; it also examined different selection schemes and different types of crossover and mutation operators. A brief outline of the genetic operators is given below.

5.1.1.1 Binary Coding

The most popular form of GA coding uses a binary string (chromosome) to represent a set of design parameters [71]. To extend the analogy with nature the features of the string are called genes. All the GA operators described below use the binary coded form of the design parameters, the actual values of the parameters are used only in the fitness function calculation.

The precision with which the design parameters (including the chromosomes) are determined depends on how many bits it requires. The number of bits will determine the range of the parameter, particularly its maximum value. The precision π of a parameter X_i is (Homaifar et al., [105]):

$$\pi = \frac{X_{i.\max} - X_{i.\min}}{2^{\lambda_i} - 1} \quad [5.1]$$

Where λi is the number of bits used for its representation.

If m possible values of the parameter λi are required, its binary representation will need:

$$\lambda_i = \frac{\ln(m)}{\ln(2)} \text{ bits.} \quad [5.2]$$

As an example the string 10011 decodes to the base 10 number

$$1.2^4 + 0.2^3 + 0.2^2 + 1.2^1 + 1.2^0 = 16 + 0 + 0 + 2 + 1 = 19$$

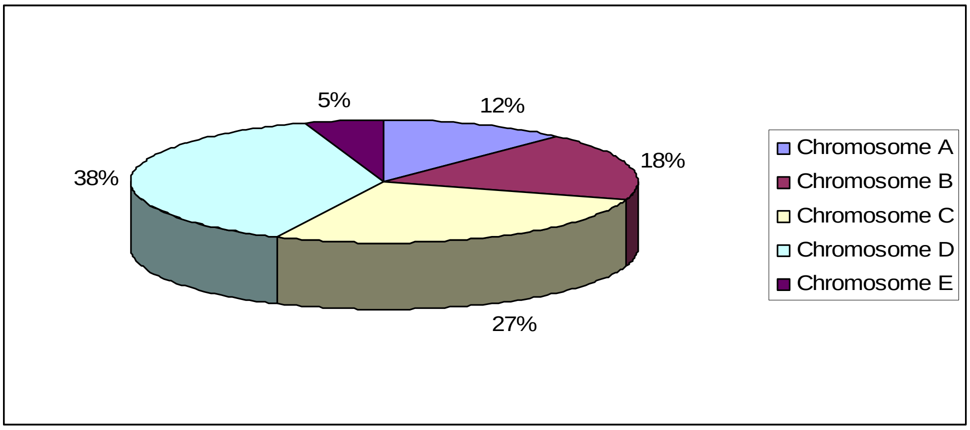
5.1.1.2 Function Evaluation

When the optimal solution to a problem is based on more than one output variable, determination of an appropriate fitness function becomes an even more important step in the optimization process. The fitness function, which is particular form of an objective function, provides a "figure of merit" for each individual chromosome. Those chromosomes designated optimal have a greater probability of taking part in reproduction, see below. An ideal fitness function would correlate closely with the GA's goal, but be computed quickly. Further details of the objective function evaluation process are given later in this chapter.

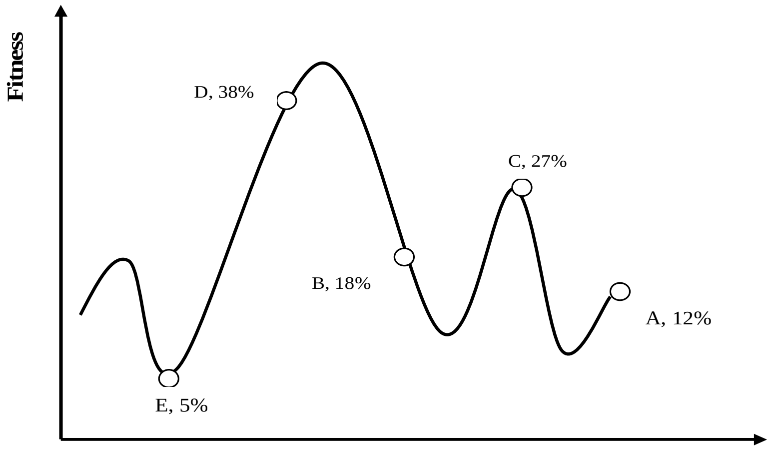
5.1.1.3 Selection Operator (Reproduction)

First the initial population is evaluated and each individual (or a random selection of individuals, depending upon population size) is given a figure of merit based on its fitness as assessed by the objective function. In the simplest caser the GA selects individual from the population to be copied into the next generation, where the probability of selection is based on the fitness of the individual. The selection is carried out with replacement: that is the GA randomly selects two individuals from the current population, assesses their relative fitness and a copy of the fitter individual is retained for the next generation. Both individuals are then replaced into the present population and the process is restarted. This is repeated until the next generation is fully populated.

A simple and common method for selection assigns each chromosome a probability of selection for the next generation that is proportional to the ratio of the individual's figure of merit divided by the total of the figures of merit for the entire population. This selection is easily implemented using an imaginary roulette wheel divided into as many segments as there are individuals in the population [71], [75]. The size of the segment allocated to any individual is proportional to the fitness of the individual, see figures 5-3(A) and 5-3(B). Given a fair wheel, the size of the segment is directly proportional to its likelihood of being selected.



(A): Selection process using roulette wheel for five individual chromosomes with figures of merit; 0.05, 0.12, 0.18, 0.27 and 0.38



(B): Selection process using roulette wheel

Figure 5-3 Roulette wheel selection process

The tournament selection strategy randomly selects pairs of individuals who “fight” to become parents through their relative figures of merit [71], Goldberg and Deb [106] claim that tournament selection is an improvement on roulette wheel selection.

Tournament selection proceeds as follows:

- The initial generation is “mixed” so that the order of individuals is completely random.
- Two individuals are selected at random and their figures of merit assessed by the objective function. The fitnesses are compared. The individual with the higher fitness is chosen as parent A. The two individuals are not placed back into the initial population.
- Two more individuals are selected at random and their figures of merit assessed by the objective function. The individual with the higher fitness is chosen as parent B.
- Parents A and B are used in crossover and mutation operations, to produce the next generation, see below.
- The above process is repeated until the desired number of children is attained.

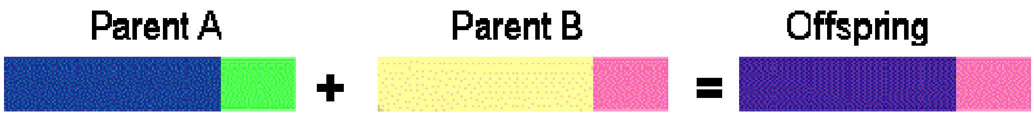
Different methods of selection have slightly different effects on the progress of the GA.

5.2 Crossover Operator

Crossover in GAs is analogous to reproduction (which is biological crossover), and allows a generation of new individuals (new possible solutions in the search space) to be created and tested. The crossover begins with, say parents A and B chosen by tournament selection the basis of their fitness. Crossover will produce two children. Each child will contain genetic material from each parent which may result in fitter individuals. Three possible ways of performing crossover are shown below.

5.2.1 Single point crossover

A single crossover point is set. From parent A, a binary string from the beginning of the chromosome to the crossover point is copied, and the remainder of the new chromosome is copied from parent B, see figure 5-4

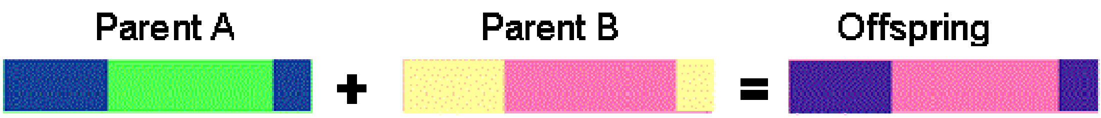


11001011+11011**111** = **11001111**

Figure 5-4 Single point crossover

5.2.2 Two points crossover

Two crossover points are set. From parent A, a binary string from the beginning of the chromosome to the first crossover point is copied. From parent B, the part of its chromosome from the first to the second crossover point is copied. The remainder of the chromosome is copied from parent A.

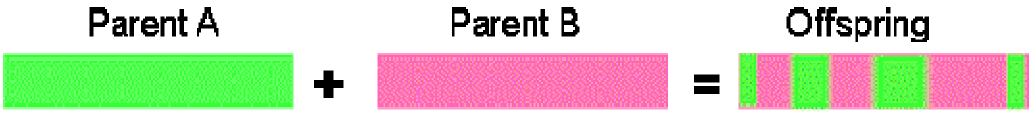


1**10**010**11** + **1**10**111**01 = 11011111

Figure 5-5 Two point crossover

5.2.3 Uniform crossover

Bits are randomly copied from both Parents A and B.

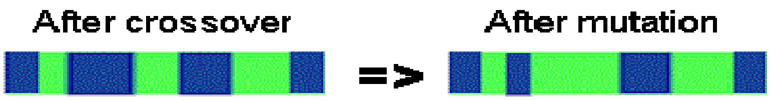


1**10**010**11** + **1**10**111**01 = 11011111

Figure 5-6 uniform point crossover

5.3 Mutation Operator

Mutation is used to maintain diversity in the population. Mutation is the final basic genetic operator for GA. It is necessary because reproduction and crossover on their own can occasionally lose genetic diversity including potentially useful genetic material. This is because they can create a point in the neighbourhood of the current point (local search around the current solution). Mutation achieves its goal by randomly changing parameters in a chromosome (with low probability otherwise it might cause the GA to become too randomly in its search) and can be seen as an insurance policy against premature loss of important notions. In binary coding this means simply changing a 1 into 0 and vice versa, randomly, with a very low probability. In figure 5-7 there is an example of this low probability event, in string 1**1**001001 position 2 is chosen randomly to mutate. The new string is 1**0**001001.



1**1**001001 => 1**0**001001

Figure 5-7 Mutation operator

5.3.1 Creep Mutation

It is possible that the mutation may cause an offspring's chromosomes to be mutated by a single increment. Creep mutation introduces a random parameter value that must remain in the appropriate range, i.e., the parameter could creep one increment up or down from one of the parents' values. In this project the probability of a creep mutation occurring for any one chromosome was set equal to 1/pop [107].

5.4 Elitism Operator

Elitism can speed up GA performance significantly and help to prevent the loss of good solutions once they have been found. When creating a new generation using crossover and mutation, there is a real chance that the fittest chromosome will be lost. Elitism is the name given to the method that preferentially copies the best chromosome (or a number of the fittest chromosomes) to the new population.

5.5 Constraint Handling

Typical GAs are designed to solve unconstrained problems. However, in many fields – particularly engineering - constrained optimisation arises naturally and often . Thus GAs have been adapted to solve optimisation problems that include inequalities and/or equality constraints. This usually includes applying penalties to unfeasible solutions. Constrained optimisation problems are usually written as non-linear programming problem of the following type:

Maximise $\mathrm{f}(\vec{\mathrm{x}})$

Subject to:

$$g_j(\vec{x}) \geq 0, \qquad j = 1, J$$
$$h_k(\vec{x}) \geq 0, \qquad k = 1, K$$
$$x_i^l \leq x_i \leq x_i^u, \qquad i = 1, n$$

Where

- $f(\vec{x})$ is the objective function with n of variables,
- $\vec{x}$, and J, K are the number of inequality and equality constraints, respectively,
- $g_j(\vec{x})$ and $h_k(\vec{x})$ are the j^{th} inequality and k^{th} equality constraints, respectively,
- and the parameter x varies in the range $[x_i^l, x_i^l]$.

When GAs use penalty functions for constraint handling in any problem, the penalty parameters must be correctly set to obtain visible solutions. A typical implementation is:

$$eval\ f(\vec{x})\begin{cases} f(\vec{x}), & \text{feasible region} \\ f(\vec{x}) + penalty(\vec{x}), & \text{unfeasible region} \end{cases}$$

Where penalty $f(\vec{x})$ is zero, if no violation occurs, otherwise it is negative.

5.6 GA Code Employed in this Study

The GA code in the current study is the secure GA171, a Fortran version of the GA driver, developed by Carroll [108]. This program includes tools to enable the GA to carry out optimisation in FORTRAN code using any genetic operators. GA171 initialises a random sample of individuals with different parameters to be optimised using the GA approach. Recently, an option for the use of a micro-GA has been added.

Tournament selection is used with a shuffling technique for selecting random pairs for mating. The routine includes binary coding of individuals, jump and creep mutations, and single-point or uniform crossover, and an option for changing the number of children per pair of parents.

5.7 Test function

A difficulty with multimodal functions is that they have many unwanted local optimal solutions which cause problems for any optimisation algorithm seeking the global optimum solution. This is because with local optima there exist many attractors to which an algorithm can become directed. In this project the simple multimodal function described by Equation [5.3] will be used as an objective function for the test only, to test the efficacy of the GA code and to study the effects of different GA techniques.

$$f(x) = (\sin(5.1 * \pi * x + 0.5)^{nvalley} * \exp(-4 * \log 2) * \frac{(x - 0.0667)^2}{0.64}) \quad [5.3]$$

Figure 5-8 shows a plot of f(x) as a function of x. The range of the test function is [0,1] for the x domain of [0,1]. The test function consists of five unequally spaced peaks at x values of 0.080, 0.247, 0.451, 0.681, and 0.934, and respective heights 1.000, 0.948, 0.770, 0.503 and 0.250. It is deemed illustrative on non-linear optimisation problems found in aerodynamics, and considered to be a reasonably tough problem for the GA. This is an non-dimensional version of the multimodal function with decreasing peaks used by Goldberg and Richardson [109].

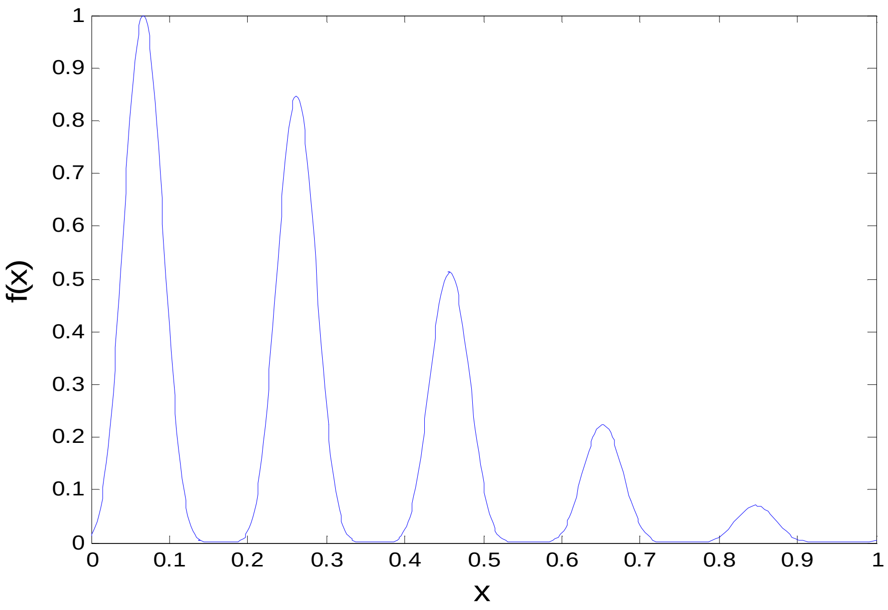


Figure 5-8 The multimodal function

5.8 GA Parameterisation

5.8.1 Effect of Population Size

It is expected that larger populations will find the optimal individual (solution) for a given problem in fewer generations than smaller populations. But larger populations are expected take longer to move from one generation to the next. The parallel problem of the number of generations (time required by the optimisation routine) necessary to locate the global maximum against population size, was evaluated using five populations of 20, 50, 100, 150 and 200 individuals.

Figure 5-9 shows the results and, as expected, the larger the population size (npop), the fewer the number of generations required. More importantly is the question of how many calls to the multimodal function were made to find the optimal solution for the different populations. It was found that npop = 200 required 1600 function evaluations; npop = 100 required 1200 function evaluations; npop=50, required only 1050 function evaluations; npop=20 required only 600 function evaluations. It can be seen that for this particular problem npop=20 case had the best performance in terms of computational cost.

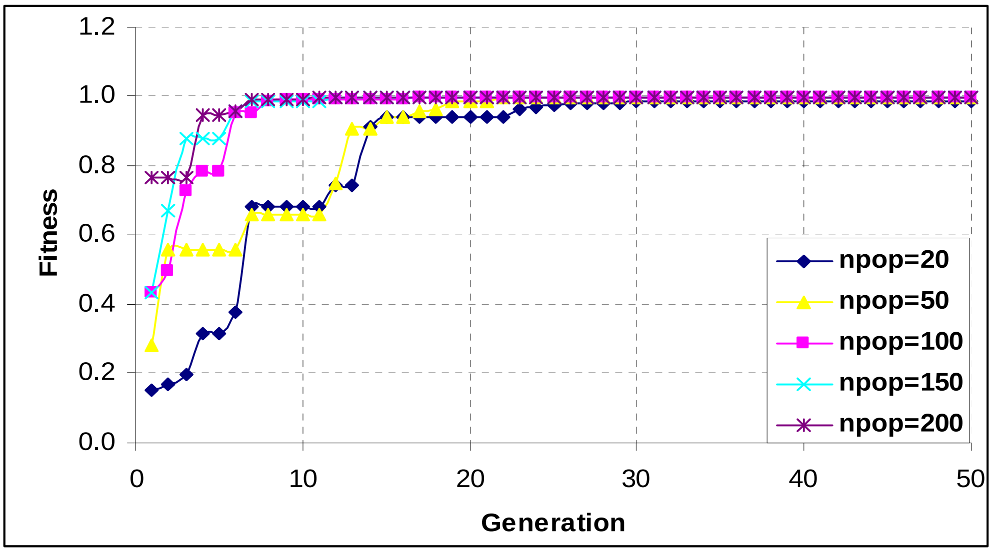


Figure 5-9 GA performance versus population size

For quick function evaluations which take less than 1 second of CPU time it may be acceptable to run larger populations**.**

5.8.2 Effect of Crossover

The multimodal function, f(x) defined in Equation [5.3] was used to investigate The effect of crossover on the efficiency of finding the optimal solution. Single-point crossover tended to preserve fewer individuals than uniform crossover, which made, uniform crossover Carroll's preferred choice [103].

However, here the question is whether there is a significant difference between the two crossover techniques for the present application. Tests were performed using single-point and uniform crossover and the results are shown in figure 5.10. The figure shows GA performance versus crossover and niching scheme. It shows that uniform crossover (iuniform=1, iniche=1) approaches the optimal solution more rapidly than single-point crossover (iuniform=0, iniche=1). Because uniform crossover case reached the optimal solution more rapidly, it was the preferred option in this GA.

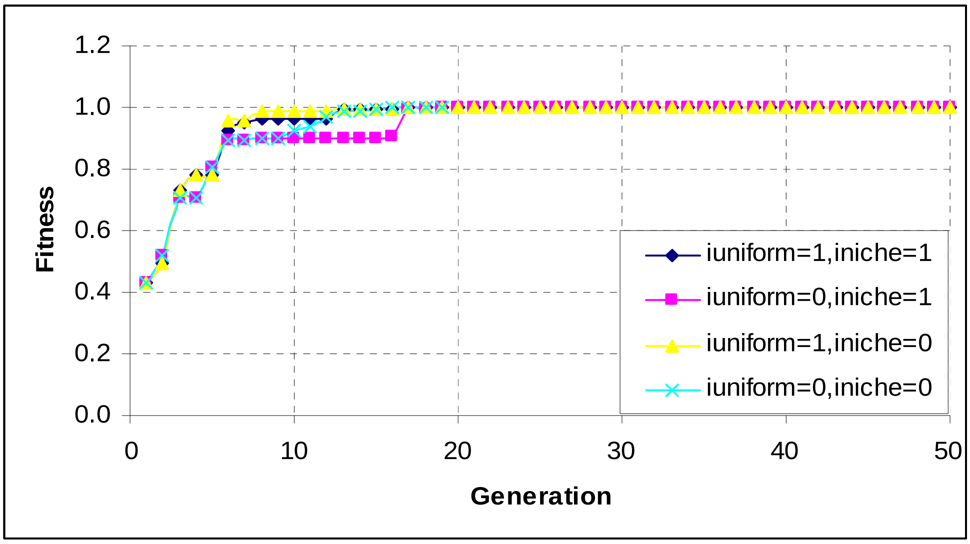


Figure 5-10 GA Performance versus crossover and niching scheme

5.8.3 Effect of Uniform Sharing

Both uniform crossover and sharing (niching) have the important attribute of ensuring GA techniques preserve variety in the genetic pool. Figure 5-10 also shows that when both uniform crossover and niching are turned off, (iuniform=0, iniche=0) the GA performance slows down and without uniform sharing, the optimum solution was not found until generation 29. (Whereas with uniform sharing the optimum solution was found at generation 13). Thus, it appears that uniform crossover and niching supplement each other to a degree, and at least one of these techniques should be included in the GA to improve its performance.

5.8.4 Effect of Creep Mutation

Creep mutation is useful because it can help move the population of individuals gradually towards the optimal solution rather than having to jump towards it. Figure 5.11 shows the "Fitness" of the population as function of the number of iterations (generations) for four sets of conditions. Here we are concerned with two sets of conditions (iuniform=0, iniche=0, icreep=1, ielite=1), the non-uniform, non-niching, creeping case and (iuniform=0, iniche=0, icreep=0, ielite=1) the non-uniform, non-niching, non-creeping case. Note elite =1 for both cases. It can be seen from the figure that with creep mutations removed (icreep = 0) the GA did not find the optimal solution until generation 18, but with the inclusion of the creep mutation enabled (icreep = 1) the GA arrived at the optimum solution sooner, after only 8 generations, and it is concluded that creep mutations benefit the GA.

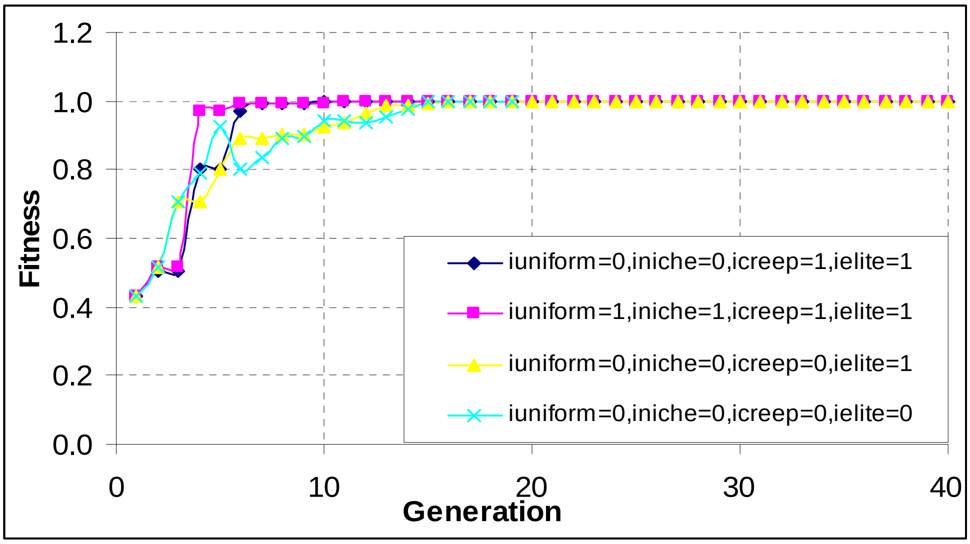


Figure 5-11 GA performance versus creep mutations and elitism

5.9 Summary

This chapter contains a brief description of features of GAs of particular interest for this study. The effects of changes of the various control parameters on the performance of the GA is presented.

This application used tournament selection, uniform crossover and creep mutation with elitism. This combination appeared to work well.

To better understand the GA and to investigate the effects of changing the different solution parameters a number of trials were run. These compared different population sizes, uniform and single point crossover and the exclusion and inclusion of creep mutation. The effect of adding niching to the GA was also tested.

Overall, the GA technique adopted worked well for the given objective function. However, because GAs are inherently parallel, further modification could enhance this technique to be faster and more effective at finding the global optimum.

6 OPTIMISATION SCHEME AND IMPLEMENTATION

An automated optimisation environment has been developed, which simplifies the implementation and testing of optimisation and evaluation schemes, see figure 6-1. The optimisation environment provides a generic interface to external solvers (Turbomatch). In this tool Turbomatch is working as an external solver to provide the genetic algorithm (GA) with the required performance parameters.

The objective function and constraints are evaluated with respect to the ambient and operating conditions and characteristics. The objective of the optimisation process is to maximise the total profit over the expected life for the plant by varying the TET (power setting) or minimising the total cost in the case of repowering an existing power plant.

6.1 Total Profit Objective Function:

In this optimisation process the code will maximise the total profit for the power plant based on operating and ambient conditions and by using the following objective function equation:

Objective Function =

$$(+)\ 91.25\left[\sum_{S=1}^{S=4}\ \sum_{t=1}^{t=12}\left\{\left(E_{(t)}\times EP_{(t)}-\left(C_{cap(t)}+C_{Fuel(t)}+C_{OM\,(t)}+C_{emi(t)}+C_{Fin(t)}\right)\right)\right\}\right] \quad [6.1]$$

In this case the variable parameter will be the turbine entry temperature (TET) for each engine in the library; the parameter will range between maximum and minimum values which are input individually for each engine. In this study the TET represents the power output, and the task is to determine the power output for each engine in the fleet that will maintain the objective function while considering the objective function constraint of the total power demand. Equation 6.1 represents the total profit for one year which includes four seasons and 91.25 typical days in each season.

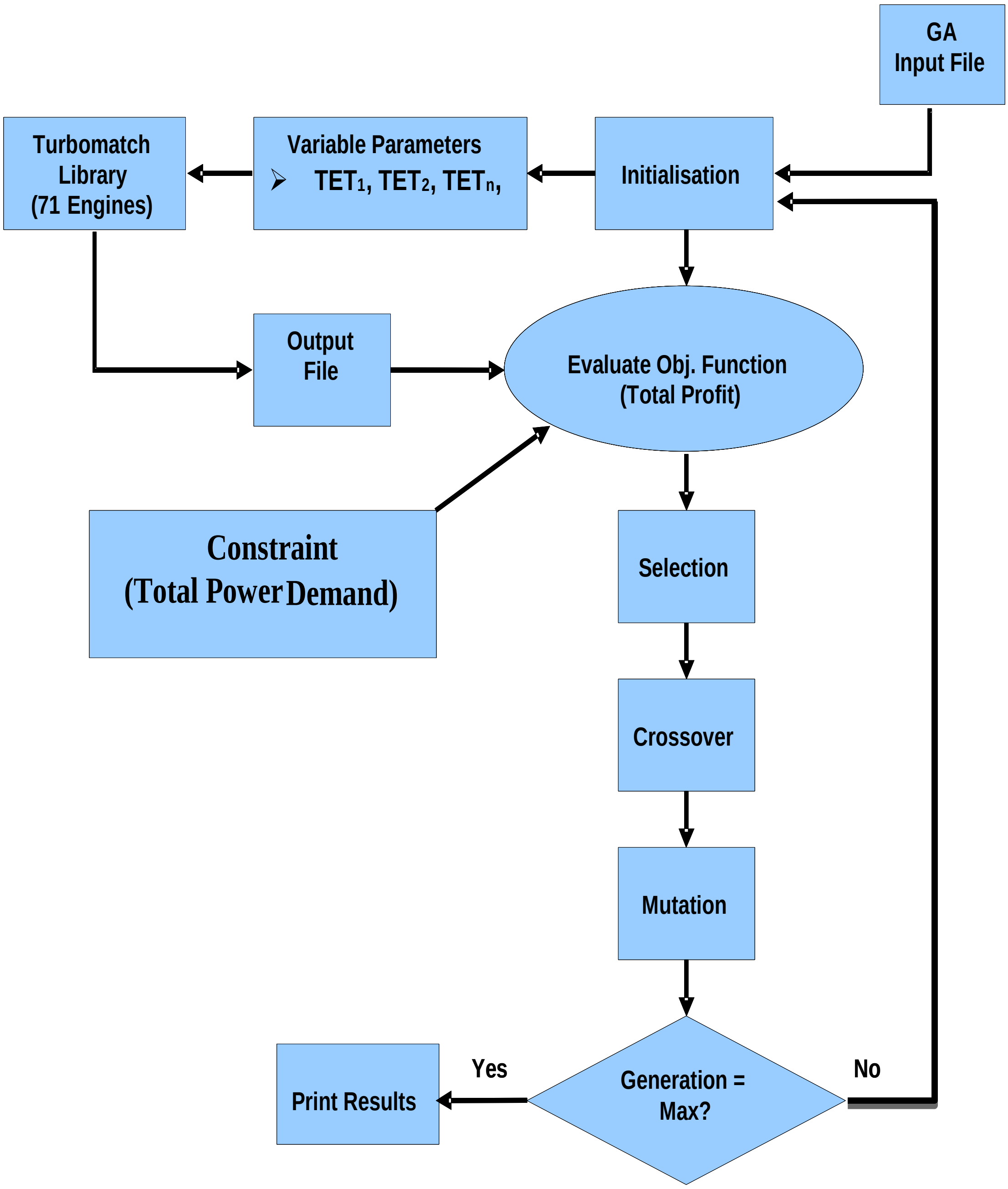


Figure 6-1 Optimisation Logic Design

6.2 Total Cost Objective Function

In case of an existing power plant the challenge will be to minimise the total cost (life cycle cost), and this will lead to maximising the total profit. In this case the gas turbine units are already existing the task is to find the optimum configuration for them that will maintain the minimum total operating and maintenance cost plus any additional related costs such as emission tax.

The variable parameters for this function will be the TET (as with the total profit objective function). The objective function in this case will be the minimum total cost as given in the objective function equation below. This will be for one year.

Objective Function =

$$(-)\ 91.25\left[\sum_{S=1}^{S=4}\ \sum_{t=1}^{t=12}\left\{\left(C_{Fuel\,(t)}+C_{OM\,(t)}+C_{emi\,(t)}\right)\right\}\right] \qquad [6.2]$$

6.3 Case Studies

As mentioned earlier this study will evaluate two objective functions to maximise the total profit (or to minimise the total cost) and to validate the code that has been written. The code applies to two real case studies: The first case study is for an already existing power plant running in an oil field in the Sahara desert, and the second case study assumes the power plant is at the preliminary selection stage but the same ambient and operating conditions apply as for the existing power plant. These case studies are based on the data given in figure 4-3 (ambient temperature and power demand profiles for the summer season), figure 4-5 (the variation of electricity price during typical summer day) and figure 4-11 (monthly natural gas prices). Similar data for the other seasons can be seen in Appendix D.

6.3.1 Repowering an Existing Power Plant

The main aim of this case study is to evaluate the total cost for the given power plant and compare it with the results obtained from using the code, evaluate the benefits of the applied procedure, and investigate the effects of the cost of fuel, maintenance and operating costs and emission tax on profitability.

The power plant that has been selected to be the case study can be considered as a good example for such work because of the surrounding environment and operating conditions. This power plant is owned by the AGOCO oil company which is one of the largest oil and gas companies in Libya. This company has eight oil fields and two refineries located in the south of Libya in the Sahara desert. One of these fields, the Sarir Oil Field, has been selected to be the case study for this work. This field produces about 250,000 bpd of waxy crude oil and the field consists of three main gathering centres, 9 main satellites, a small refinery with capacity of 60,000 bpd, two tank farms and accommodation facilities for about 1200 employees.

The power demand varies over the day and with the season. The Sahara desert is considering a very hot area and the variation of ambient temperature during the same day is very significant. Figures 4-2 and 4-3 show the weather map, ambient temperature and power demand profile during the summer season for this field.

Table 6-5 presents the current operational procedure of the oil field during a typical summer day, the on-site power plants consist of simple cycle engines: three GE LM2500+, four Tornado SGT 200, four Sulzer type 7 and five units Sulzer type 3. The performance simulations of these engines were carried out as same as GE LM2500+, in chapter 3, the Turbomatch input and output data for these engines is shown in Appendix A.

This case study shows the performance for the existing engines and their operating and maintenance cost based on the ambient condition and power demand, Appendix F shows the performance curves for the engines for the two cases of when the engines are clean (no degradation) and the performance based on actual operating conditions in the field on a typical summer day.

In this study the ambient temperature is given for four different days, each day is typical of one season, and each day is divided into twelve segments of two hours each, according to the operational time-table in the field, see Appendix E.

It is noted that every running engines in the fleet is operating at part load, away from its design point due to the high ambient temperature and compressor fouling problems – due mainly to the plant and equipment operating in an oil laden sandy environment.

In particular, for constant power output the effect of compressor fouling requires a progressive increase in fuel flow, a consequent increase in firing temperature and reductions in turbine blade creep life.

The figures in Appendix F show the behaviour of one engine from each type for one season (summer). The performance simulation for the others has been created based on the operating conditions and percentage degradation of each of them.

Changing the operational strategy for the existing units by using GA shows a significant reduction in cost, see tables 6-6 and 6-7. For the summer season the total cost of the current operational strategy is about £11.6 million, while the proposed engine configuration should reduce the total cost to about £9.6 million over the summer season, a saving of about £2 million in total cost for one season over the expected useful life of the plant. Some data has been used based on table 6-1 in this case study.

The objective function solution history is shown in figure 6-2. This plot shows the minimum fitness, and hence the minimum total cost, of the population for each generation. As expected, the larger the population size (npop), the fewer the number of generations are required to find the optimal solution.

Input Data	
Ambient Temp. Profile	See Appendix D
Power Demand Profile	See Appendix D
Engines input data	Turbomatch Library
GA Parameters	GA Input File
Electricity Price (£)	See figure 4-5
Fuel Price	See figure 4-11
Fixed M&O Cost	£3/MW
Engine Capital Cost	Turbomatch Library
Emission Tax	£ 0.079/kg
Depreciacion	0.004
Interest Rate	10%
Tax	4.5%

Table 6-1 Input data for case study 1

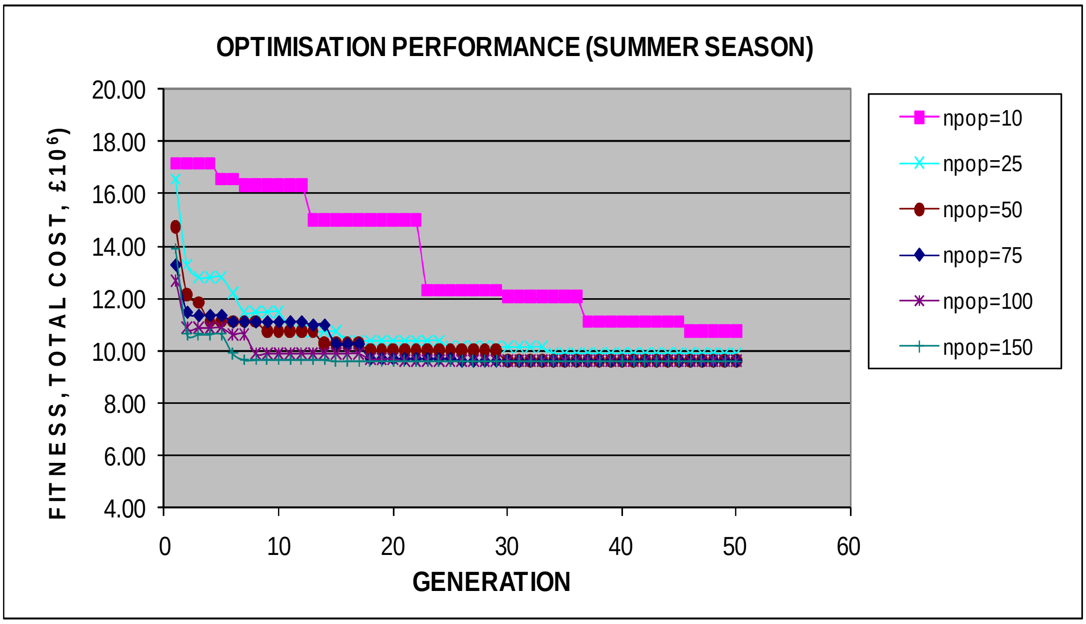


Figure 6-2 GA Total cost convergence history

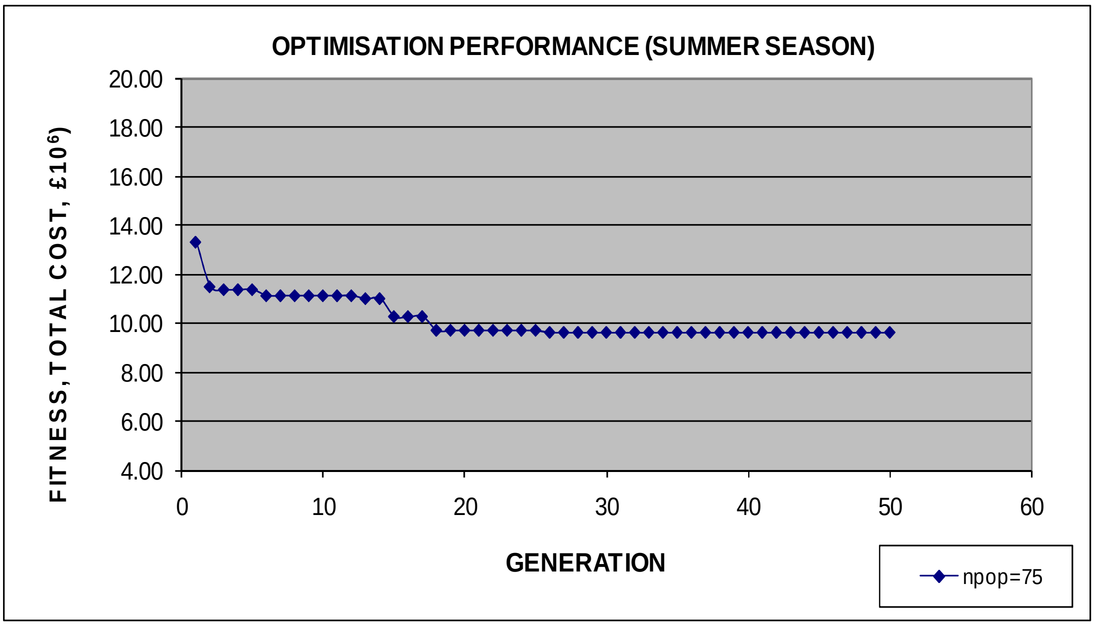


Figure 6-3 GA total cost convergence history (npop=75)

Population size	75
Current generation	50
Maximum function value	9.63×10^{6}
Average function value of generation	10.2×10^{6}
Number of crossovers	2161
Number of creep mutations	26
Elitist reproduction on individuals	19
Number of evaluations	3750

Table 6-2 global statistical report

Unlike what is often observed in GA convergence histories, the plots in figure 6-2 show fairly continuous lines indicative of a smooth design space. This characteristic is due to using an elitism operator whereby the best individuals of a given generation are guaranteed replication in the following generation. In order to ensure that accuracy is maintained and that the optimised selection is unbiased, each numerical solution was required to run until the residual that originates from the non-satisfaction of the discretised continuity and momentum equations, dropped to the same level.

The objective function solution history is shown in Figure 6.3. This plot reports the minimum fitness, and hence the minimum total cost. The final statistical report produced by GA for optimisation run of the total cost is presented in Table 6-2.

The units in this case are running at constant power, and the fuel flow has been increased in order to restore the power loss. The cost of fuel consumption during the summer season increased by approximately £394,000 over the current configuration, see table 6-6. Table 6-3 shows the fuel consumption and the cost brake down for engine GE LM2500+ during the summer and tables 6-6 and 6-7 summarised the various costs for whole the running engines in the summer. The proposed configuration presented by GA shows that the increase in the cost of fuel consumption is a bit lower approximately £315,000 due to the different operational strategy as shown in table 6-7.

Time	T Ambient ^{0}C	Power MW	TET, ^{0}K	Fouled 3% TET,	Fuel Flow Kg/s	Fouled Fuel Flow, 3.3% kg/s,	Fuel Consumed, kg (Clean)	Fuel Consumed, kg (Degraded)	Fuel Cost, £ Clean	Fuel Cost, £ Degraded
0	13	18	1331	1371	0.98683	1.01939539	7,105	7,340	663.0692347	684.9505194
2	13	18	1331	1371	0.98683	1.01939539	7,105	7,340	663.0692347	684.9505194
4	14	19	1348	1388	1.01423	1.04769959	7,302	7,543	681.4797988	703.9686322
6	19	19	1381	1422	1.18961	1.22886713	8,565	8,848	799.3208478	825.6984358
8	24	19	1396	1438	1.20035	1.23996155	8,643	8,928	806.5372514	833.1529807
10	33	19.3	1431	1474	1.33205	1.37600765	9,591	9,907	895.0289047	924.5648586
12	47	21	1513	1558	1.529	1.579457	11,009	11,372	1027.363234	1061.26622
14	47	21	1513	1558	1.529	1.579457	11,009	11,372	1027.363234	1061.26622
16	42	20	1476	1520	1.43251	1.47978283	10,314	10,654	962.5298272	994.2933115
18	34	21	1438	1481	1.2354	1.2761682	8,895	9,188	830.0879914	857.4808951
20	23	20	1414	1456	1.2235	1.2638755	8,809	9,100	822.0921624	849.2212038
22	17	18	1339	1379	0.99782	1.03074806	7,184	7,421	670.4536179	692.5785873
							105,531	109,014	9848.395338	10173.39238

Table 6-3 Fuel cost related to time, ambient temperature and power for GE LM2500+ in summer

The economic analysis of this particular case shows the significant cost penalty due to compressor fouling. In the case where the gas turbine is always operated at constant TET, less fuel is used at the expense of reduced power. Table 6-4 summarises the estimated cost of cleaning. However, research by Lambert et al [112] suggests that the optimum frequency for gas turbine compressor cleaning is usually measured in days rather than weeks. Typically for the small and medium engines used here, a liquid on-line spray cleaning every 3 - 5 days is recommended for optimal washing performance.

If the turbine is not subject to any type of compressor washing, it is much better if it operates under constant power output. This gives a smaller cost penalty than constant TET. This is especially the case if the engine is operating most of the time at part load. In this case, the TET is much less than the design value and has a much smaller effect on turbine blade creep life.

That is clear evidence that the gas turbine engine operations cost is very sensitive to the power setting and the different ambient conditions. Optimising the operation plan can lead to a significant reduction in total cost. Figures 6-4 and 6-5 summarise the benefits on the total cost of applying GA to the operational strategies for both clean and degraded engines.

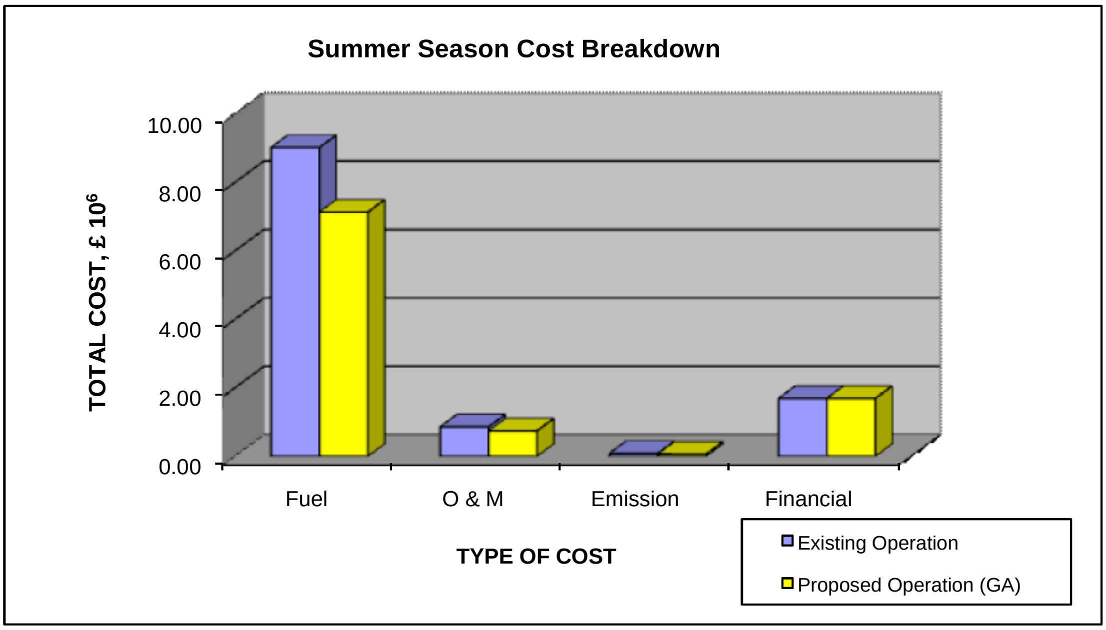


Figure 6-4Total cost breakdown, degraded engines

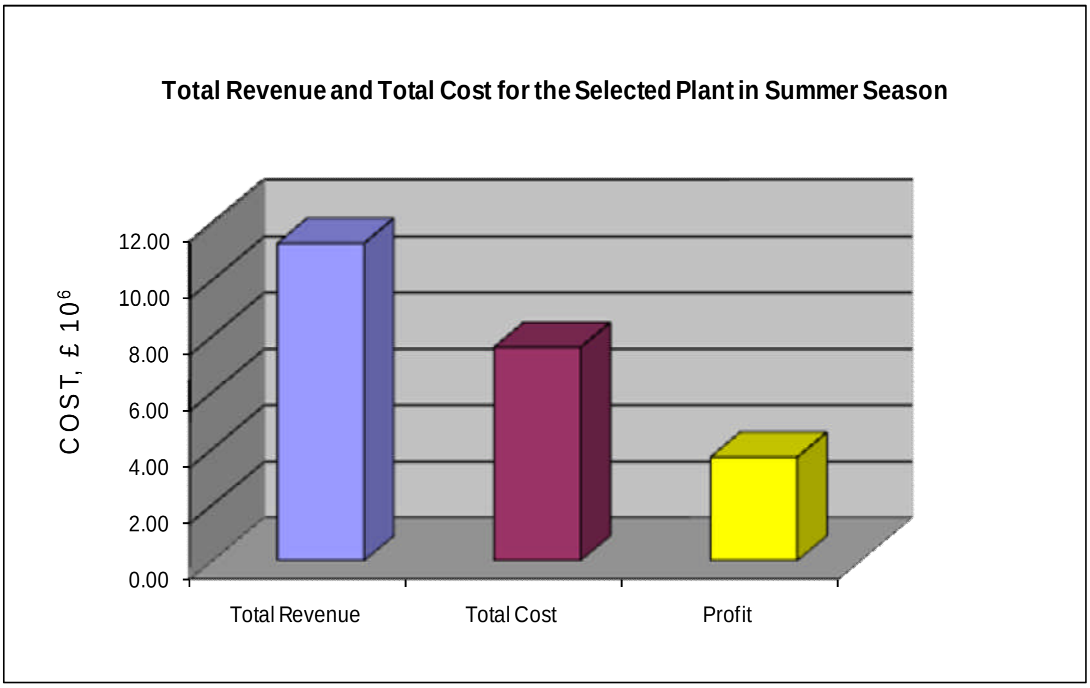


Figure 6-5 Total cost for clean and degraded engines

A	Capital cost of a typical high pressure spraying machine	£60,000	
B	Capital cost per gas turbine engine	£20,000	
C	Savings due to cleaning	£146,000	
D	Capital annual cost per engine	£500	
E	Cleaning fluid cost per clean	£200	
F	Number of washes per season	18	
G	Total seasonal fluid cleaning cost per engine	E x F	£3,600
H	Total seasonal cost of cleaning	(D + G) x K	£65,600
K	Number of engines	16	
L	Total seasonal net saving	C - H	£80,400
M	Total initial outlay (For 1 high pressure machine)	5x A	£300,000
N	Pay-back period (months)	(M/L) x 12	45

Table 6-4 Financial saving obtained by compressor cleaning

TIME INTERVAL, HOURS of DAY	AMBIENT TEMP, ^{0}C	AVERAGE POWER DEMAND, MW	Engines Running (Existing)	Running Engines (Proposed)
0	13	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 10MW
2	13	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = MW
4	14	116	LM2500+ 3 X 19 = 57 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 9.67MW Sulzer 3, 1 x 3.33= 3.33 MW
6	19	125.5	LM2500+ 3 X 19 = 57 MW Tornado, 3x7.5 = 22.5 MW Tornado, 1x7 = 7 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.25= 4.25 MW
8	24	129	LM2500+ 3 X 19 = 57 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x8 =8 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
10	33	131	LM2500+ 3 X 19.3 = 58MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22.5 = 67.5 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.9= 19.5 MW
12	47	136	LM2500+ 3 X 21 = 63 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22 = 66 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.6= 18 MW
14	47	136	LM2500+ 3 X 21 = 63 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22 = 66 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.6= 18 MW
16	42	134	LM2500+ 2 X 20 = 40MW LM2500+ 1 X 21 = 21MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 21.65 = 64.9 MW Tornado, 4 x5.1 = 20.4 MW Sulzer 7, 4 x8.78 =35.12 MW Sulzer 3, 3 x 4.52= 13.56 MW
18	34	135	LM2500+ 2 X 21 = 42MW LM2500+ 1 X 20 = 20MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 21.94 = 65.8 MW Tornado, 4 x5.1 = 20.5 MW Sulzer 7, 4 x8.78 =35.12 MW Sulzer 3, 3 x 4.52= 13.56 MW
20	23	133	LM2500+ 3 X 20 = 60MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 24.7 = 74.1 MW Tornado, 4 x5.87 = 23.48 MW Sulzer 7, 4 x8.86 =35.44 MW Sulzer 3, 5 x 3.2= 16 MW
22	17	123.5	LM2500+ 2 X 18 = 36 MW LM2500+ 1 X 19 = 19 MW Tornado, 3x7.5 = 22.5 MW Tornado, 1x7 = 7 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25 = 75 MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW

Table 6-5 Operational Procedure (Existing & Proposed) for Typical Summer Day

TABLE 6-6 TOTAL COST BRAKE DOWN FOR THE CURRENT OPERATIONAL STRATIGY IN SUMMER

TOTAL COST BREAKDOWN FOR THE APPLIED OPERATIONAL STRATEGY IN SUMMER SEASON

		TIME												FUEL CONSUMED, kg		FUEL COST, £/DAY		£/SUMMER		EXPECTED
		0	2	4	6	8	10	12	14	16	18	20	22	CLEAN ENGINE	DEGRADED ENGINE	CLEAN ENGINE	DEGRADED ENGINE	CLEAN ENGINE	DEGRADED ENGINE	LIFE (YEAR)
LM2500+	UNIT1	18	18	19	20.16	21.5	21	22	22	22	22	22	21	105,531	109,014	9,848.00	10,173.00	898,630.00	928,286.25	6.2
	UNIT2	18	18	19	20.16	21.5	21	22	22	22	22	22	21	105,531	109,014	9,848.00	10,173.00	898,630.00	928,286.25	6.2
	UNIT3	18	18	19	20.16	21.5	21	22	22	22	22	22	21	105,531	109,014	9,848.00	10,173.00	898,630.00	928,286.25	6.2
TORNADO	UNIT1	5.125	5.125	5.125	6.5	5	4.87	5	5	5	5	5	5.37	36,139	38,163	3,373.00	3,561.00	307,786.25	324,941.25	5
	UNIT2	5.125	5.125	5.125	6.5	5	4.87	5	5	5	5	5	5.37	36,139	38,163	3,373.00	3,561.00	307,786.25	324,941.25	5
	UNIT3	5.125	5.125	5.125	6.5	5	4.87	5	5	5	5	5	5.37	36,139	38,163	3,373.00	3,561.00	307,786.25	324,941.25	5
	UNIT4	5.125	5.125	5.125	6.5	5	4.87	5	5	5	5	5	5.37	36,139	38,163	3,373.00	3,561.00	307,786.25	324,941.25	5
SULZER 7	UNIT1	7	7	7	6	7	8	8.125	8.125	8	8	7.75	7.75	82,183	85,717	7,670.00	8,000.00	699,887.50	730,000.00	6.5
	UNIT2	7	7	7	6	7	8	8.125	8.125	8	8	7.75	7.75	82,183	85,717	8,052.00	8,398.00	734,745.00	766,317.50	6.1
	UNIT3	7	7	7	6	7	8	8.125	8.125	8	8	7.75	7.75	82,183	85,717	7,972.00	8,314.00	727,445.00	758,652.50	6.2
	UNIT4	7	7	7	6	7	8	8.125	8.125	8	8	7.75	7.75	82,183	85,717	7,972.00	8,314.00	727,445.00	758,652.50	6.2
SULZER 3	UNIT1	3.5	3.5	3.5	3	3.5	3.4	3.5	3.5	3.2	3.4	3.2	3	48,535	52,175	4,520.00	4,869.00	412,450.00	444,296.25	4.2
	UNIT2	3.5	3.5	3.5	3	3.5	3.4	3.5	3.5	3.2	3.4	3.2	3	48,535	52,175	4,520.00	4,869.00	412,450.00	444,296.25	4.2
	UNIT3	3.5	3.5	3.5	3	3.5	3.4	3.5	3.5	3.2	3.4	3.2	3	48,535	52,175	4,520.00	4,869.00	412,450.00	444,296.25	4.2
	UNIT4	0	0	0	3	3.5	3.4	3.5	3.5	3.2	3.4	3.2	3	33,492	34,513	3,126.00	3,220.00	285,247.50	293,825.00	5.1
	UNIT5	0	0	0	3	3.5	3.4	3.5	3.5	3.2	3.4	3.2	3	33,492	34,513	3,126.00	3,220.00	285,247.50	293,825.00	5.1
														1,002,470	94,514	94,514.00	98,836.00	8,624,402.50	9,018,785.00	

FUEL CONSUMPTION AND TOTAL FUEL COST DURING SUMMER SEASON

CASE ANALYSIS	TOTAL FUEL CONSUMPTION (kg)	TOTAL FUEL COST (£)	O & M COST	EMISSION	FINANCIAL	TOTAL COST
CLEAN ENGINES	1,002,470	8,624,403	980,000	79,195	1,691,250	11,374,848
DEGRADED ENGINES	1,048,113	9,018,785	863,000	82,801	1,691,250	11,655,836
EFFECT OF FOULING	45,643	394,383				

Table 6-6 Total Cost Breakdown for the Current Operational Strategy in Summer Season

GA PROPOSED OPERATIONAL STRATEGY

		TIME												FUEL CONSUMED, kg		FUEL COST, £/DAY		£/SUMMER		EXPECTED)
		0	2	4	6	8	10	12	14	16	18	20	22	CLEAN ENGINE	DEGRADED ENGINE	CLEAN ENGINE	DEGRADED ENGINE	CLEAN ENGINE	DEGRADED ENGINE	LIFE (YEAR)
LM2500+	UNIT1	25.5	25.5	25.5	25.5	25.5	22.5	22	22	21.65	21.94	24.7	25	89,754	92,983	8,375.84	8,677.17	764,296	791,792.09	7.6
	UNIT2	25.5	25.5	25.5	25.5	25.5	22.5	22	22	21.65	21.94	24.7	25	89,754	92,983	8,375.84	8,677.17	764,296	791,792.09	7.6
	UNIT3	25.5	25.5	25.5	25.5	25.5	22.5	22	22	21.65	21.94	24.7	25	89,754	92,983	8,375.84	8,677.17	764,296	791,792.09	7.6
TORNADO	UNIT1	6.62	6.62	6.62	6.5	6.5	5	5	5	5.1	5.1	5.87	6.37	34,555	36,490	3,224.67	3,405.25	294,251	310,728.77	6.4
	UNIT2	6.62	6.62	6.62	6.5	6.5	5	5	5	5.1	5.1	5.87	6.37	34,555	36,490	3,224.67	3,405.25	294,251	310,728.77	6.4
	UNIT3	6.62	6.62	6.62	6.5	6.5	5	5	5	5.1	5.1	5.87	6.37	34,555	36,490	3,224.67	3,405.25	294,251	310,728.77	6.4
	UNIT4	6.62	6.62	6.62	6.5	6.5	5	5	5	5.1	5.1	5.87	6.37	34,555	36,490	3,224.67	3,405.25	294,251	310,728.77	6.4
SULZER 7	UNIT1	10	10	10	9.37	9.5	8	8	8	8.78	8.78	8.86	9.5	84,216	87,840	7,859.04	8,197.23	717,137	747,997.13	8.6
	UNIT2	0	0	0	9.37	9.5	8	8	8	8.78	8.78	8.86	9.5	78,682	82,065	7,342.60	7,658.31	670,013	698,820.40	7.2
	UNIT3	0	0	0	0	0	8	8	8	8.78	8.78	8.86	0	57,947	59,225	5,407.61	5,526.88	493,445	504,327.53	11
	UNIT4	0	0	0	0	0	0	8	8	8.78	8.78	8.86	0	45,888	47,723	4,282.27	4,453.51	390,757	406,382.82	11.5
SULZER 3	UNIT1	0	0	3.33	4.25	4.25	3.9	3.6	3.6	4.52	4.52	3.2	4	38,482	41,368	3,591.14	3,860.46	327,692	352,267.14	5.4
	UNIT2	0	0	0	0	4.25	3.9	3.6	3.6	4.52	4.52	3.2	0	25,087	26,969	2,341.12	2,516.75	213,627	229,653.17	6.1
	UNIT3	0	0	0	0	0	3.9	3.6	3.6	4.52	4.52	3.2	0	22,314	23,988	2,082.34	2,238.56	190,014	204,268.61	6.5
	UNIT4	0	0	0	0	0	3.9	3.6	3.6	0	0	3.2	0	20,178	21,691	1,883.01	2,024.20	171,825	184,708.63	6.9
	UNIT5	0	0	0	0	0	3.9	3.6	3.6	0	0	3.2	0	20,178	21,691	1,883.01	2,024.20	171,825	184,708.63	6.9
														800,454	837,469	74,698.37	78,152.61	6,816,226	7,131,425.40	

FUEL CONSUMPTION AND TOTAL FUEL COST DURING SUMMER SEASON						
CASE ANALYSIS	TOTAL FUEL CONSUMPTION (kg)	TOTAL FUEL COST (£)	O& M COST	EMISSION	FINANCIAL	TOTAL COST
CLEAN ENGINES	800,454.00	6,816,226	931,000	63,236	1,691,250	9,501,712
DEGRADED ENGINES	837,469.00	7,131,425	744,000	66,160	1,691,250	9,632,835
EFFECT OF FOULING	37,015.00	315,199				
RECOVERABLE ENGINES, 60%	22,209.00	189,120				

BENEFIT FROM THE CODE FOR ONE SEASON FOR CLEAN ENGINES,	£	1,873,136
BENEFIT FROM THE CODE FOR ONE SEASON FOR DEGRADED ENGINES,	£	2,023,000

TABLE 6-7 TOTAL COST BRAKE DOWN FOR GA PROPOSED OPERATIONAL STRATIGY IN SUMMER

Table 6-7 Total Cost Breakdown for GA Proposed the Current Operational Strategy in Summer Season

6.3.2 Selecting New Power Generation

This case study using the same ambient and operating data used in the previous case study to select a new set of engines and to set their optimum power setting in order to maximise the total profit over their expected useful life. The selection process will use Turbomatch engines library as an engine data base resource.

As stated in Equation 6-1 the total profit objective function includes the total revenue and the total capital cost in addition to the life cycle costs. The optimisation process follows the same procedure as the previous case, but using the equation of total profit objective function.

Based on the ambient and operating input data, the objective function shows that the maximum fitness (total profit) is approximately £3.72 million, see in figure 6-6, and this can be achieved by selecting 6 identical units of GE LM2500+ and operate them according to the profile as illustrated in table 6-10. The operational summer performance for these engines can be seen in Appendix G.

GA convergence history for the summer season total profit is shown in figure 6-6; see previous study for explanation of the effects of the GA parameters. The objective function solution history is shown in Figure 6-7. Also maximum profit for one year is shown in figures 6-8 and 6-9. These plots report the maximum fitness, and hence the maximum total profit. The final statistical report produced by GA for optimisation run of the total cost in summer season is presented in Table 6-8 and for one year in table 6-9. The operation plan has been set based on the minimum life cycle cost considering the different type of costs mentioned earlier. The revenue and cost brake down are shown in table 6-11 and figures 6-10 and 6-11.

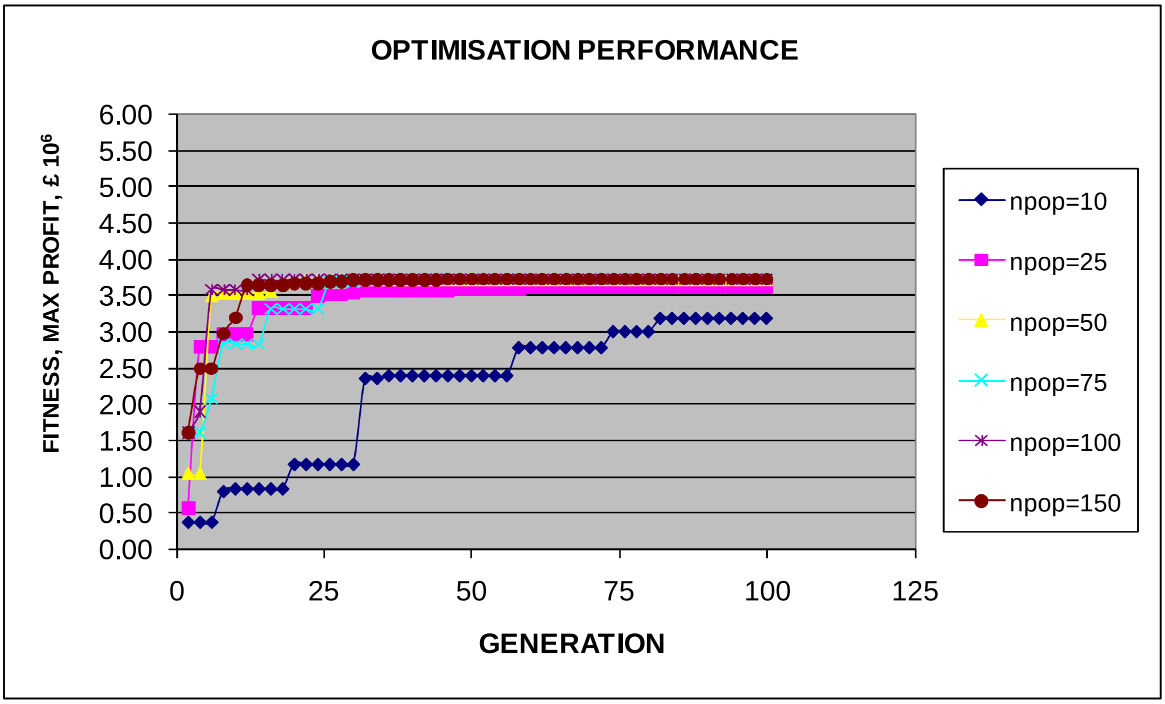


Figure 6-6 Genetic Algorithm convergence history for summer season

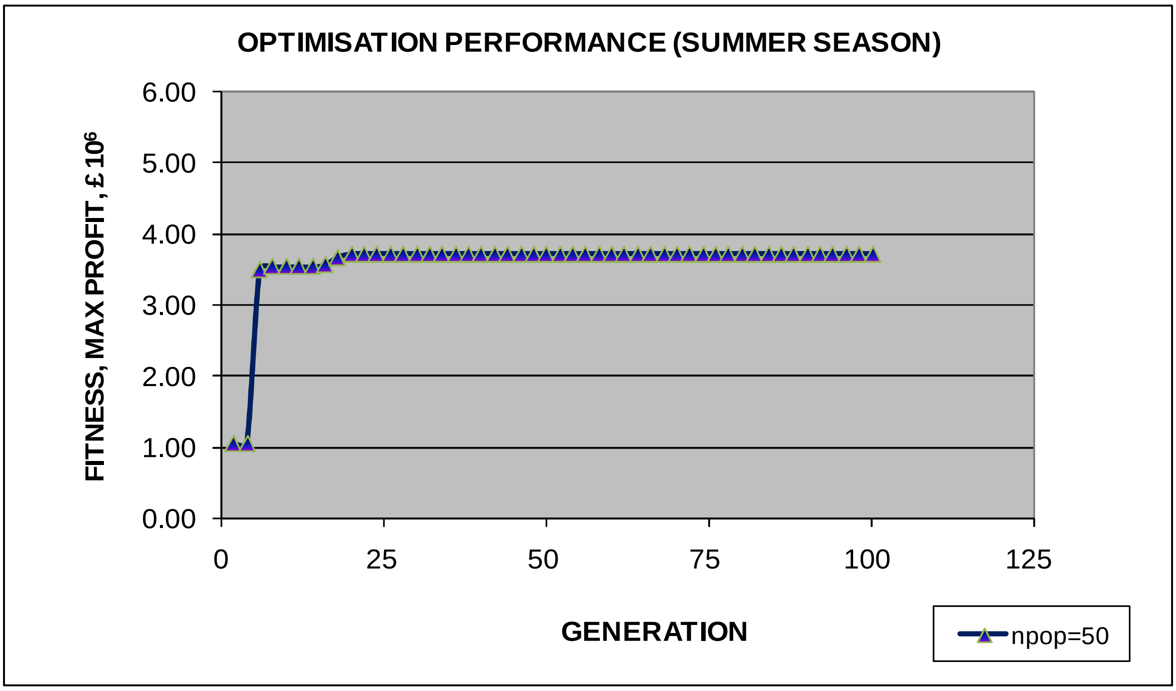


Figure 6-7 GA Total cost convergence history, summer profit (npop=50)

Population size	50
Current generation	100
Maximum function value	3.73×10^6
Average function value of generation	3.50×10^6
Number of crossovers	2845
Number of creep mutations	41
Elitist reproduction on individuals	66
Number of evaluations	5000

Table 6-8 Global statistical report (summer profit)

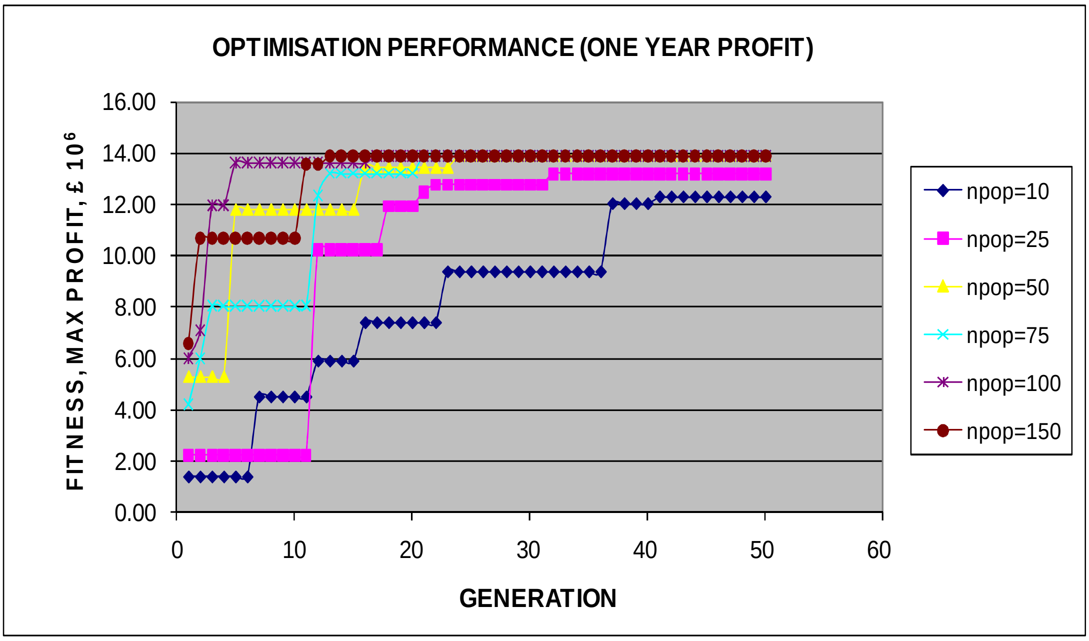


Figure 6-8 Genetic Algorithm convergence history for one year profit

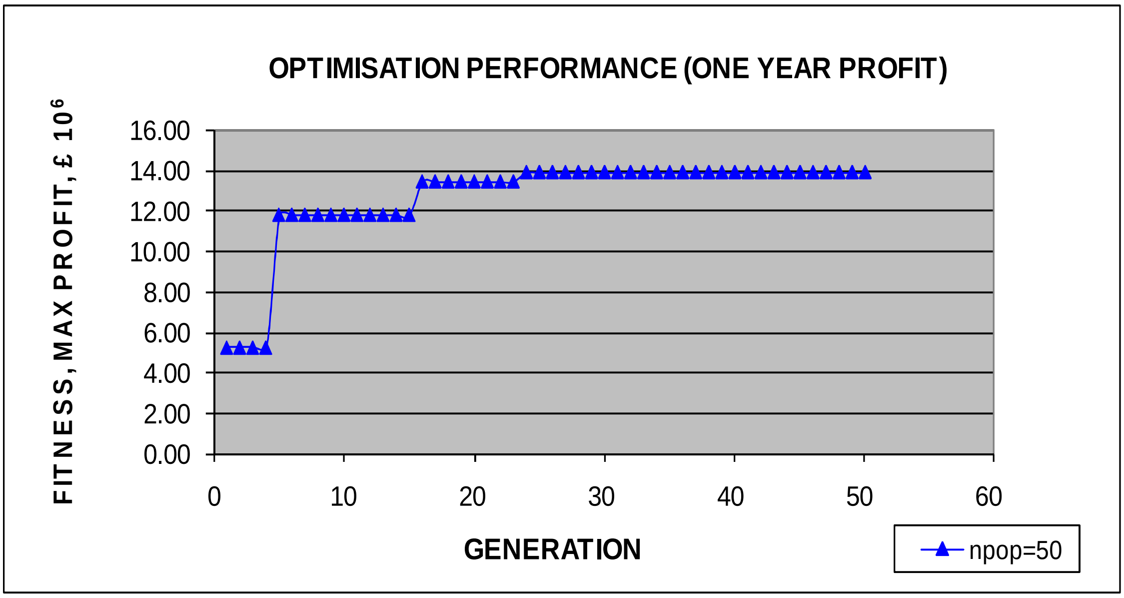


Figure 6-9 GA Total cost convergence history, one Year Profit (npop=50)

Population size	50
Current generation	100
Maximum function value	$13.9x10^6$
Average function value of generation	$12.7x10^6$
Number of crossovers	3425
Number of creep mutations	53
Elitist reproduction on individuals	72
Number of evaluations	5000

Table 6-9 Global statistical report (one year profit)

TIME INTERVAL, HOURS OF DAY	AMBIENT TEMP, ^{0}C	AVERAGE POWER DEMAND, MW	SELECTED ENGINES	POWER SETTING
0	13	113	GE LM2500+	5 X 22.6MW
2	13	113	GE LM2500+	5 X 22.6MW
4	14	116	GE LM2500+	5 X 23.2 MW
6	19	125.5	GE LM2500+	5 X 25.1 MW
8	24	129	GE LM2500+	6 X 21.5 MW
10	33	131	GE LM2500+	6 X 21.8 MW
12	47	136	GE LM2500+	6 X 22.6 MW
14	47	136	GE LM2500+	6 X 22.6 MW
16	42	134	GE LM2500+	6 X 22.3 MW
18	34	135	GE LM2500+	6 X 22.5 MW
20	23	133	GE LM2500+	6 X 22.2 MW
22	17	123.5	GE LM2500+	6 X 20.5 MW

Table 6-10 Selected Engines and their Operation Plan Based on the Maximum Profit (Summer Season)

	OPERATIONAL STRATEGY AND TOTAL COST BREAKDOWN FOR THE NEW SELECTED ENGINES																		
		TIME														FUEL CONSUMED, kg	FUEL COST, £/DAY	£/SUMMER	EXPECTED
			0	2	4	6	8	10	12	14	16	18	20	22		Ful Consumed, kg	CLEAN ENGINE	CLEAN ENGINE	LIFE (YEAR)
LM2500+		UNIT1	0	0	0	0	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		75,398	7,036.00	642,035.00	7.8
		UNIT2	22.6	22.6	23.2	25.1	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		116,097	10,834.00	988,602.50	7.8
		UNIT3	22.6	22.6	23.2	25.1	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		116,097	10,834.00	988,602.50	7.8
		UNIT4	22.6	22.6	23.2	25.1	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		116,097	10,834.00	988,602.50	7.8
		UNIT5	22.6	22.6	23.2	25.1	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		116,097	10,834.00	988,602.50	7.8
		UNIT6	22.6	22.6	23.2	25.1	21.8	21	22.6	22.6	22.3	22.5	22.2	20.5		116,097	10,834.00	988,602.50	7.8
																655,883	61,206.00	5,585,047.50	

ELECTRICITY PRODUCED, MWh	ELECTRICITY PRICE, £/MW	REVENUE £	CAPITAL COST £	TOTAL FUEL CONSUMPTION (kg)	TOTAL FUEL COST, £	O & M COST £	EMISSION TAX £	FINANCIAL £	TOTAL COST £	TOTAL PROFIT/SUMMER £
245,638	50	12,281,900	990,000	655,883	5,585,048	886,950	51,815	89,100	8,400,762	3,725,893

Table 6-11 Operational Strategy and Cost Brake Down for the New Selected Engines

Table 6-11 Operational Strategy and Cost Breakdown for the New Selected Engines

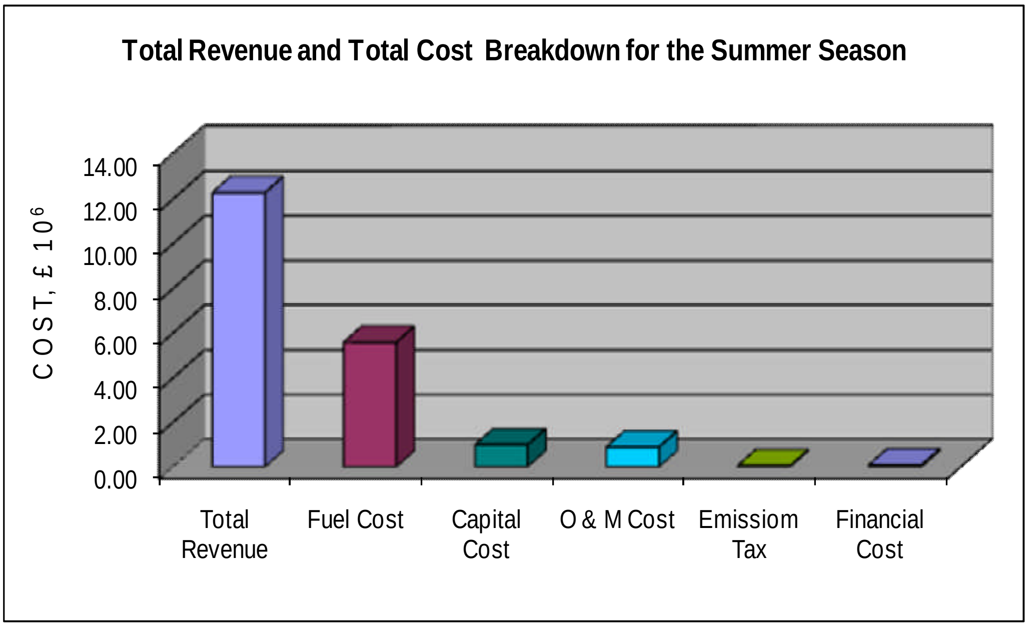


Figure 6-10 Total Cost Breakdown for the Selected Plant in Summer Season

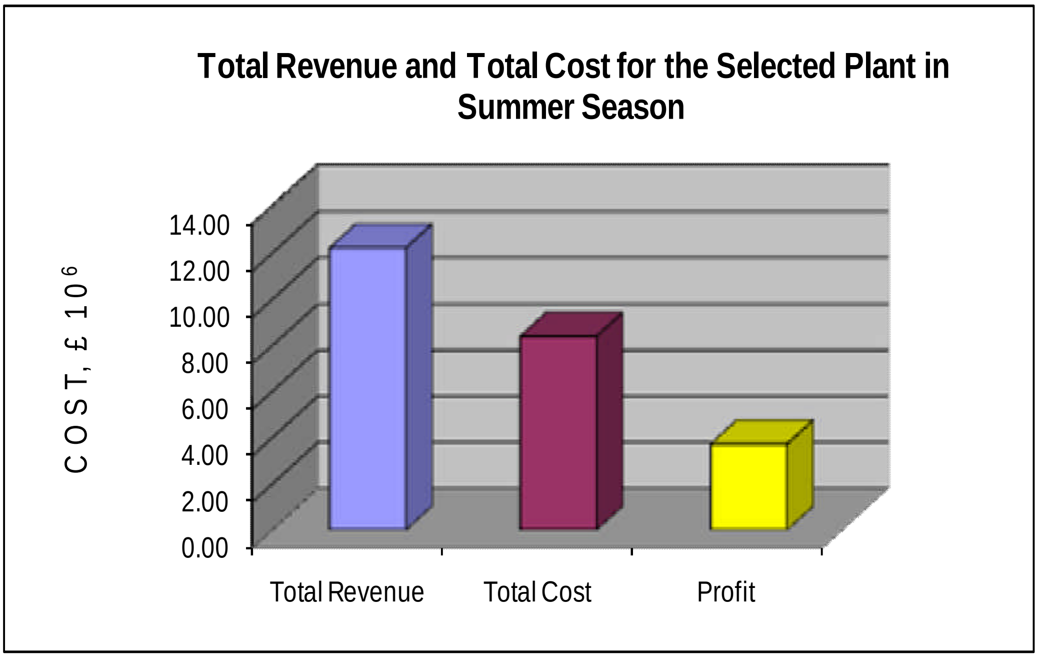


Figure 6-11 Total Revenue and Total Cost for the Selected Plant in Summer Season

7 CONCLUSION AND RECOMMENDATIONS FOR FUTURE RESEARCH

7.1 Conclusion

This research has presented the detailed development of a computational method for maximising the total profit and minimising the total cost in operating gas turbine engine combinations. This includes changes in both ambient and operating conditions over the expected life of the power plant. A specific code has been developed and is described which links a GA optimisation code with Turbomatch; the two codes have been linked to run as a single linked package. With Turbomatch running as a solver the GA works to optimise the problem.

This task was accomplished by creating Turbomatch library that includes more than 83 different engines and here many subroutines which have been developed inside the code: including a variable maintenance cost, emission tax and financial cost subroutines.

The motivation for the present study has been the growth in power demand and the dramatically increase in oil and gas prices. Hence; the need for powerful optimisation tool that will enable the user to select and operate the most cost effective power generation set to purchase. Also this will contribute to cut down the time consuming of optimisation process.

Over the different phases of this study many aspects have been investigated. This has included a literature review which has shown how relevant previews and included some introductory remarks about gas turbine design and off design performance and the principle of fouling degradation followed by economic and life cycle review and the relevant optimisation method review.

The second phase of this study has investigated the behaviour of the industrial gas turbine at different ambient and operating conditions to determine the relevant parameters that would maximise the objective function (total profit) and also set the best values for these parameters.

The third phase of this study has been a techno-economic analysis which has analysed the different function costs included in the revenue and the total cost. Total revenue is broken down to the electricity produced and the price of the electricity, considering the relevant changes in price of electricity.

The second part of the techno-economic analysis considers the total cost (life cycle). This includes the capital cost for the engines, fuel cost, emission tax, maintenance and operating costs, (divided into fixed and variable costs) depreciation and tax and interest rates which are included as financial costs.

The variable maintenance cost investigation shows that the creep life for the turbine blades has the major influence on the maintenance cost from both replacement cost and down-time.

The fourth phase of this study prepares the GA for implementation and develops the objective function based on the relevant parameters from the engine performance simulation and the total profit from the techno-economic analysis.

A brief description of genetic algorithm and its working mechanisms is presented in this part of work. In addition, the code is validated as user secures GA171. It also provides a study of the parameterisation of the GA variables to efficiently search a highly multimodal parameter space for a global maximum.

Following GA parameterisation, the link between GA and Turbomatch is developed. Different subroutines are used to calculate the different cost functions added to the code with the flexibility to add any other cost subroutines without significant change in the code.

The advantage of applying genetic algorithm methods, when compared with gradient based optimisation methods, is related to their robustness, their capability for parallel process, their efficiency when the number of variable parameters is high and when the objective function has many local minima and the fact that they do not require modification for a specific problem.

The method has been applied to optimise the power plant configuration with the aim of achieving maximum total profit with constraints on the total power demand.

The code has been applied to two different case studies

The first was to minimise the total cost for an existing power plant by repowering the existing engines. The results show a significant reduction in the total cost (approximately £8 million annually) when compared with the total cost for its current operational strategy. The major life cycle cost comes from savings in fuel which accounts for about 75 % of the total cost.

The investigation of the power plant studied has shown that all the 16 engines ran at part load away from the design point all the time, even when the ambient temperature was relatively low. It has also been noted that compressor fouling has a significant effect on the fuel flow rate and the firing temperature. The firing temperature has been increased by about 45^0K in the GE LM2500+ units and the fuel flow increased by 3.3% due to degradation caused mainly by compressor fouling. For the smaller units in the fleet the percentage of degradation is higher as smaller gas turbine engines are more sensitive to compressor fouling.

The research has also shown that a simple way of calculating the benefits of compressor cleaning of about £760,000 a year can be achieved from reduction in fuel costs. Simultaneously, the turbine blade life can be improved

This study has been applied to very hot and dusty environment with a very large difference in ambient temperature during the same day. At the same time, the power demand profile showed a significant variation over the day. These changes mean, of course that the optimum engine combination will vary with time of day.

The second case study relates to the selection of new engines and the determination of their daily and seasonally operational profiles to maximise total profit over their expected useful lives. In this study different cost functions were involved in the GA objective function. These included the total revenue, capital cost and the total cost (life cycle cost). The GA results determine maximum total profit and engine configuration to maintain maximum profit.

7.2 General Observations

In general this research shows that GA is a powerful optimisation tool that can be applied in gas turbine power plant usage optimisation. The overall results have shown a significant improvement in the total profit when using the GA optimiser. This performance investigation has also shown that the most important parameter effecting operating and maintenance costs are the turbine entry temperature for each engine. This is therefore, the variable parameter used for each engine.

The optimisation code developed in this research has been validated through the two case studies as mentioned above.

The first has shown that about 17% reduction in the total cost can be achieved by optimising the engines combined power profiles during the day while considering the total power demand constraint. About 92% of this saving comes from the fuel consumption. This has shown how gas turbine power plant profitability is very sensitive to both operating and ambient conditions.

In the second case study the code proposes a set of engines from the Turbomatch library based on the maximum total profit as stated in the objective function. In addition, that code is able to show the cost brake down for the selected engines.

The user of the code can modify or add any cost function without modifying the main core of the GA program.

Finally this research has developed an optimisation method based on genetic algorithms which can maximise the profit for a typical industrial gas turbine portfolio at various power demands while also considering the local changing ambient conditions.

7.3 Recommendations

From the above it is clear that this present work is a sound basis for a future more detailed study. Accordingly, the following future work is recommended:

- An investigation of the heat production of the power plant as this can be another type of energy in addition to electrical or mechanical,
- To further investigate the cost of shutdown and restart,
- An investigation of NO_X and CO emissions produced in addition to CO_2.
- An Investigation different type of fuel and the economic benefit of using cheaper heavy oil fuel,
- To carry out more detailed calculation for the creep life, and investigate the use of different turbine blade materials,

- To consider reliability and availability costs,

  Undertake an economic assessment of compressor washing and filtration systems, and

- To adapt the interface code to provide a friendlier user interface.

REFFERENCES

[1] H Selandurayand and M Hariffin “Power Plant Optimization in a Regulated Environment Electricity Supply Industry: A Least Cost Generation Approach, 2nd IEEE International Conference on Power and Energy (PECon 08), December 1-3, 2008, Johor Baharu, Malaysia.

[2] P.P.Walsh, P.Fletcher “Gas turbine performance”, Blackwell Science-ASME, 1998 Ltd, Oxford, UK.

[3] Brooks, F. J. (2000), GE Gas Turbine Performance Characteristics, GE Report.

[4] 30] Frank, J. B., “Gas turbine performance characteristics”, GER-3567H, GE Power Systems, Schenectady, New York

[5] Erdam H A & Sevilgen H S, Case study: effect off ambient temperature on the electricity production and fuel consumption of simple cycle gas turbine, , applied thermal engineering 26 (2006) 230-236)

[6] Gorji M & Fouladi F, Ambient temperature effects on gas turbine power plant: a case study, Department of mechanical engineering, faculty of engineering, university of Mazandaran Babol, Iran

[7] Zwebek, A. and Pilidis, P. (2003), Degradation effects on combined cycle power plant performance – Part II: steam turbine cycle component degradation effects, Journal of Engineering for Gas Turbines and Power, Vol. 125, pp. 658 663.

[8] Weisman, J. and Eckart, L. E. (1985), Modern Power Plant Engineering, Prentice-Hall, USA.

[9] Zwebek, A. and Pilidis, P. (2004), Degradation Effects on combined cycle power plant performance-Part III: gas and steam turbine component degradation effects, Journal of Engineering for Gas Turbines and Power, Vol. 126, pp. 306-315.

[10] Mund, F. and Pilidis, P. (2004), A review of gas turbine online washing systems, Turbo Expo 2004

[11] Mund, F. (2006), Coordinated Application of CFD and Gas Turbine Performance Methods (PhD Thesis), Cranfield University, UK.

[12] Giampolo, T. (1997), The Gas Turbine Handbook, The Fairmont Press, USA.

[13] Zwebek, A. (2002), Combined Cycle Performance Deterioration Analysis (PhD Thesis), Cranfield University, UK

[14] Boyce, M. and Gonzalez, F. (2005), A study of on-line and off-line turbine washing to optimise the operation of a gas turbine, ASME Turbo Expo 2005.

[15] Williams, L. J. (1981), The optimisation of time between overhauls for gas turbine compressor units (4th Symposium of Gas Turbine Operations and Maintenance edition), In N.R.C of Canada, Canada.

[16] Diakunchak, I.S., “Performance degradation in industrial gas turbines”, ASME Paper No. 91-GT-228

[17] Kurz R. and Brun K. (2001), Degradation in gas turbine systems, Journal of Engineering for Gas Turbines and Power, Vol. 123, No. 1, pp. 70-77.

[18] Syverud, E. and Bakken, L. (2005), Online water wash tests of GE J85-13, ASME Turbo Expo 2005.

[19] Tabakoff, W. (1986), Compressor Erosion and Performance Deterioration, AIAA/ASME 4th Joint Fluid Mechanics, Plasma Dynamics, and Laser Conference May 12-14, Atlanta.

[20] Singh, D. (1996), Simulation of Performance Deterioration in Eroded Compressor, American Society of Mechanical Engineering.

[21] Morillo De Hart, L. (1994), Performance Diagnostic of RR Avon Engines Using Gas Path Analysis, (MSc Thesis), Cranfield University, UK.

[22] Caguiat, D., Zipkin, D. M., and Patterson, J. (2002), Compressor fouling testing on Rolls Royce/Allison 501-K17 and general electric LM2500 gas turbine engines, American Society of Mechanical Engineers Turbo Expo 2002, Vol. 2, No. B, pp. 933-942.

[23] Zwebek, A. (1993), One giga watt single shaft Industrial Gas Turbine Design (MSc Thesis), Cranfield University, UK.

[24] Lakshminarasimha, A., Boyce, M., and Meher-Homji, C. (1994), Modelling and analysis of gas turbine performance deterioration, Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol. 116, No. 1, pp. 46-52.

[25] Howell, A. R. and Calvert, W. J. A. (1978), New Stage Stacking Technique for Axial Flow Compressor Performance Prediction, 100:698-703.

[26] Aker, G. and Saravanamuttoo, H. (1989), Predicting gas turbine performance degradation due to compressor fouling using computer simulation techniques, Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol. 111, No. 2, pp. 343-350.

[27] Gulen, S., Griffin, P., and Paolucci, S. (2002), Real-time on-line performance diagnostics of heavy-duty industrial gas turbines, Journal of Engineering for Gas Turbines and Power, Vol. 124, No. 4, pp. 910-921.

[28] T W Song, J L Sohn, T S Kim, J H Kim, and S T Ro (2004), An analytical approach to predict particle deposit by fouling in the axial compressor of the industrial gas turbine DOI: 10.1243/095765005X7547.

[29] Diakunchak, I. S. (1991), Performance deterioration in industrial gas turbines (International Gas Turbine and Aero-engine Congress and Exposition jun 3-6 edition), New York, USA.

[30] Ahmed Arhoma Benjalool (2006), Evaluation of Performance Deterioration on Gas Turbines due to Compressor Fouling, Cranfield University, UK., MSc Thesis.

[31] Haq, I. and Saravanamuttoo, H. (1991), Detection of axial compressor fouling in high ambient temperature conditions, American Society of Mechanical Engineers , pp. 9.

[32] Zaba, T. L. P. (1984), Experience in the Operation of Air Filters in Gas Turbine Installations (29th International Gas Turbine Conference Exhibition edition), ASME.

[33] McDermott, P. (1991), Gas turbine compressor fouling. The case for on-line cleaning, Turbomachinery International, Vol. 32, No. 1, pp. 34-38.

[34] Upton A.W.J. (1974), Axial Compressor and Turbine Blade Fouling Some Causes, Effects and Cleaning Methods (Gas Turbine Operations and Maintenance Symposium edition), National Research Council Canada, Canada.

[35] Mustafa, Z. (2006), Analysis of Droplets in Compressor Gas Turbines (PhD Thesis), Cranfield University, UK.

[36] Hoeft, R. F. (1993), Heavy Duty Gas Turbine Operating and Maintenance Considerations, GE I&PS.

[37] Osborne, R. (1977), Design of an Industrial Gas Turbine Intake, American Air Filter Co.

[38] Seddigh, F. and Saravanamuttoo, H. (1990), Proposed method for assessing the susceptibility of axial compressors to fouling, International Gas Turbine and Aeroengine Congress and Exposition, Jun 11-14 1990, Brussels, Belg, pp. 8.

[39] A.P.Tarabrin and et. al., 1996, "An analysis of Compressors Fouling and a Cleaning Method of their Blades", ASME Paper No. 96-GT-363.

[40] Meher-Homji and Cyrus B (1990), Gas turbine axial compressor fouling, a unified treatment of its effects, detection, and control, 1990 ASME Cogen-Turbo:4th International Symposium on Gas Turbines in Cogeneration, Repowering, and Peak-Load Power Generation, Aug 27-29 1990, New Orleans, LA, USA, Vol. 5, pp. 179-190.

[41] Mattingly, J.D, 1996, "Elements of Gas Turbine Propulsion ", McGraw-Hill, Inc

[42] Fuks, N.A., 1955, "The Mechanism of Aerosols", Moscow, USSR Academy of Science.

[43] Olhovsky, G.G., 1985, "Power Gas Turbine Units", Moscow, Energoatomizdat, 303p.

[44] Tabakoff, W., Lakshminarasimha, A. N., and Pasin, M. (1990), Simulation of Compressor Performance Deterioration Due to Erosion, Journal of Engineering for Power, Vol. 112, No.1, ASME.

[45] Sire, J and Stalder, J.-P.. (2001), Salt percolation through gas turbine air filtration systems and its contribution to total contaminant level, Proceedings of the International Joint Power Generation Conference, Vol. 2, pp. 445-446.

[46] Kolkman, H. (1993), Performance of gas turbine compressor cleaners, : Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol. 115, No. 3, pp. 674-677.

[47] Syverud, E. and Bakken, L. (2005), Online water wash tests of GE J85-13, ASME Turbo Expo 2005

[48] Caguiat, D., Zipkin, D. M., and Patterson, J. (2002), Compressor fouling testing on Rolls Royce/Allison 501-K17 and general electric LM2500 gas turbine engines, American Society of Mechanical Engineers Turbo Expo 2002, Vol. 2, No. B, pp. 933-942.

[49] Brumbaugh, D. (2002), Inlet air filtration adapts to evolving gas turbine technology, Power Engineering, Vol. 106, No. 10, pp. 51-54.

[50] Giampolo, T. (1997), The Gas Turbine Handbook, The Fairmont Press, USA.

[51] Mathioudakis, K. and Tsalavoutas T. (2002), Uncertainly Reduction in Gas turbine Performance Diagnosis by Accounting for Humidity effects, Journal of Engineering for Gas Turbines and Power, Vol. 124, pp. 801808.

[52] Vigueras Zuniga, M. O. (2003b), Technoeconomic Analysis of Compressor Washing (MSc Thesis), Cranfield University, UK.

[53] Langford, J. L. F. (1977), Contamination removal method, Patent 4,065,322, USA.

[54] Yee, R. and Myers, L. (2003), Enhanced tf40b gas turbine engine design changes to improve resistance to the Landing Craft Air Cushion (LCAC) operational environment, Naval Surface Warfare Center Carderock Division, C. S. C. A. M. C. USA, Vol. 3 at Atlanta Georgia USA; American Society of Mechanical Engineers, USA, pp. 495-499.

[55] Thames, J., Stegmaier, J., and Ford, J. J. J. (1989), On-line compressor washing practices and benefits, American Society of Mechanical Engineers, Jun 4-8 1989, Toronto, Ontario, Canada , pp. 6.

[56] Fielder, J. (2003), Evaluation of zero compressor wash routine in RN service, 2003 ASME Turbo Expo, Vol. 3, pp. 543-547.

[57] Bagshaw K.W. (1974), Maintaining Cleanliness in Axial Compressor (National Research Council edition), Canada.

[58] Ramsden, K. (2002), Axial Compressor Design and Performance Notes, Cranfield University, UK

[59] Scott, J. N. (1979), Axial compressor monitoring by measuring depression (In N.R.C. of Canada edition), 3rd Symposium of Gas Turbine Operations and Maintenance, Canada.

[60] Saravanamuttoo, I. and Lakshminarasimha, A. (1985), Preliminary assessment of compressor fouling, Turbomachinery International, Vol. 26, No. 7, pp. 14-18.

[61] A. W. A. Arebi (2005), The benefits of compressor cleaning on power output for oil and gas field application, (MSc Thesis), Cranfield University, UK

[62] Cyrus B. Meher-Homji, Andrew Bromley (2004), gas turbine axial compressor fouling and washing, Proceedings of the Thirty Third Turbomachinery Symposium 2004

[63] Howell, A. R. and Calvert, W. J. A. (1978), New Stage Stacking Technique for Axial Flow Compressor Performance Prediction, 100:698-703.

[64] Rodney R. Gay et al, (2000) Economic Evaluation of Plant Upgrades Using Plant Optimisation Software, GE Enter Software, LLC Davis, CA

[65] W. D. Marsh, (1980), Economic of electric utility power generation, Oxford engineering science series, Oxford University.

[66] M Junaibi, (2007), "Techno-Economic study of a CHP Plant for Thermal EOR Operations MSc Thesis, Cranfield University. 2007.

[67] Harry G. Stoll (1989), Least cost electric utility planning, John Wiley & Sons, USA

[68] Energy solution center, Citing Internet resources, http//www.energysolution center.org, (accessed 2004).

[69] G. Major, Learning from experiences with Small-scale Cogeneration, Centre for the analysis and dissemination of demonstrated energy technologies, Caddet analysis support unit, 1995.

[70] Educogen, Guide to Cogeneration, SAVE Program contract N° XVII/4.1031/P/99-159, 2001.

[71] Goldberg, D. E, (1989), Genetic Algorithms in Search, Optimisation and machine Learning, Addison Wesley, Reading, MA, 59-88

[72] Davis L, (1991) Handbook of Genetic Algorithms, Van Nostrand Reinhold, New Yourk

[73] Beasley D, Bull D. R and Martin R. R., (1993) An overview of genetic algorithms, part 1 fundamentals, University Computing, vol. 15 No2., pp 58-69

[74] Beasley D, Bull D. R and Martin R. R., (1993) An overview of genetic algorithms, part 2 Research Topics, University Computing, vol. 15 No4., pp 170-181

[75] Deb, Kalyanmoy, (1999), Multi Objective Genetic Algorithms: problem difficulties and construction to test problems, evolutionary contribution, Vol. 7 No 3 pp 205-230

[76] Ven Veldhuizen, David and Lamont, Gary, (1999), evolutionary algorithms: Analyzing the State of the Art, Evolutionary Computation, Vol. 8 No 2, pp 125-147

[77] Jiménez, Jose, Cuesta, Pedro and Abderramn Jesus, (2000), Mixed stratigy in genetic algorithms, European Congress on Computational Methods in Applied Since and Engineering, ECCOMAS, Barcelona, Spain

[78] Holland, M.J., Thake, T. F, "Rotor Blade Cooling in High Pressure Turbines", J.

[79] Wahba, W., "Design Optimisation of Centrifugal Pump Impellers Using a Parallel Genetic Algorithm". PhD Thesis, Cranfield University. 2001.

[80] Dorigo M and Maniezzo V., (1993), "Parallel Genetic Algorithms: Introduction and Overview of Current Research", IOS Press, Amsterdam, pp 5-42

[81] Kikuo Fujita et al, (1996), "Genetic Algorithms Based Optimal Planning Method of Energy Plant Configurations" The 1996 ASME Design Engineering Tehnical Conferences September 18-22, 1996, Irvine, California

[82] Richard Knight et al, (2006), "GTPOM: Thermo-Economic Optimization of Whole Gas Turbine Plant, Journal of Engineering for Gas Turbines and Power", Vol. 128 / 535]

[83] Corneliu Popescu, Daniela E. Popescu, International Geothermal Conference, Reykjavík, Sept. 2003]

[84] Turbomatch Scheme for Aero/ Industrial Gas Turbine Engine Design Point/Off Design Performance Calculation Cranfield University: Cranfield University; 2004.

[85 Web GE Energy [Web Page].Accessed 2004. Available at: http://www.gepower.com.

[86] Vassilios Pachidis. (2009), Gas Turbine Performance Simulation (unpublished Course Manual), Cranfield University.

[87] S. A. Ashair (1992) "An Industrial gas Turbine Design with LM2500 Cycle Using API 616 Standard Specification, Cranfield University

[88] MSc Lecture Notes, SOE/1077, Cranfield University,2000

[89] Temraz, H.K., Salama, M.M.A., and Chikhani, A.Y. (1997), "Review of electric load forecasting methods, Electrical and Computer Engineering," 1997. IEEE 1997 Canadian Conference on , Volume: 1 , 25-28

[90] Breipohl, A.M., "Electricity price forecasting models," Power Engineering Society Winter Meeting, 2002. IEEE , Volume: 2 , 27-31 Jan. 2002

[91] AGOCO Oil Company, (2003), " Sarir Oil Field Data Sheet Report, Ambient and Operational, 07-03/123T, July 2003, (Unpublished Report)

[92] NOC, (2005), "Electricity Market for the Libyan Oil Companies" Annual Economic Report, NOC/LL37, (Unpublished Report)

[93] Gas Turbine World, (2004-05), "Gas Turbine W Handbook", Pequot Publication, Volume 24, 2006.

[94] A. Polizakis, (2006), "Techno Economic Evaluation of Trigeneraton Plant: Gas Turbine Performance, Absorption Cooling and District Heating, Cranfield University

[95] NOC, (2005), "Hydrocarbon and Natural Gas News), National Oil Corporation Libya

[96] Meherwan P. Boyce, (2002), "Gas Turbine Engineering Handbook" Gulf Professional Publishing, USA

[97] Fatigue and Fracture MSc Lecture Notes, Cranfield University, 1999/2000

[98] Holland, M.J., Thake, T. F, "Rotor Blade Cooling in High Pressure Turbines", J.

[99] Rao. J.S, (2000) "Turbine blade life estimation", Alpha science international

[100] G.A.Webster, (1994) "High temperature component life assessment", Chapman & Hall.

[101] AGOCO, (2002), “Maintenance Cost Profile Over 10 years, Sarir and Nafoora Oilfields” Rep 02/810

[102] F.Roger, N.George, Life cycle costing, BSP Personal Books, 1989

[103] David L. Carroll, A new GA Technique in optimisation of permanent implants, 1998.

[104] David L. Carroll, Chemical Laser Modeling with Genetic Algorithms, AIAA Journal, Vol. 34, No. 2, February 1996

[105] Homaifar A., Lai S. H. V. and Qi X., (1994), “Constrained Optimisation Via Genetic Algorithms” Simulation 62 (4) 242-254

[106] Goldberg D. E. and Deb K., (1992), “Genetic Algorithms, Nois, and the Sizing of Populations”, Complex System, 6, pp. 333-362.

[107] De Jong K. A., (1975), “An Analysis of the Behaviour of a Class of Genetic Adaptive Systems”, PhD thesis, University of Michigan

[108] David L. Carroll http://cuaerospace.com/carroll/ga.html

[109] Goldberg D. E. and Richardson J., (1987), “Genetic algorithms with sharing for multimodal function optimisation”, Genetic Algorithms and their Applications: Proceedings of the Second International Conference on Genetic Algorithms. London: Lawrence Erlbaum, 1987. P41-9

[110] Kolkman, H. (1993), Performance of gas turbine compressor cleaners, : Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol. 115, No. 3, pp. 674-677.

[111] Allen, D Economic evaluation of projects 3^{ed} ed institute of chemical engineering 1991

[112] Paul Lambert and it al, 2004, “Developments in on-line gas Turbine compressor

APPENDICES

APPENDIX A

A.1 INPUT FILE TURBOMATCH DESIGN POINT PERFORMANCE CALCULATIONS, GE LM2500+

```
INDUSTRIAL GAS TURBINE SIMULATION
GENERAL ELECTRIC  LM2500 PLUS
////
DP SI NG CT FP
-1
-1
INTAKE          S1-2    D1-4        R100
COMPRE          S2-3    D5-10       R101    V5  V6
PREMAS          S3,12,4  D11-14
PREMAS          S4,13,5  D15-18
BURNER          S5-6    D19-21      R102
MIXEES          S6,13,7
TURBIN          S7-8    D22-29,101          V23
MIXEES          S8,12,9
TURBIN          S9-10   D30-38              V30 V31
NOZCON          S10-11,1 D39        R107
PERFOR          S1,0,0  D30,40-42,107,100,102,0,0,0,0,0,0
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 23.1          ! PRESSURE RATIO
8 0.873         ! ISENTROPIC EFFICIENCY
9 0.0           ! ERROR SELECTION
10 4.0          ! MAP NUMBER
!PREMAS
11 0.025        ! BLEED AIR
12 0.00         ! FLOW LOSS
13 1.0          ! PRESSURE RECOVERY
14 0.0          ! PRESSURE DROP
!PREMAS
15 0.075        ! BLEED AIR
16 0.0          ! FLOW LOSS
17 1.0          ! PRESSURE RECOVERY
18 0.0          ! PRESSURE DROP
!BURNER
19 0.075        ! FRACTIONAL PRESSURE LOSS DP/P
20 1.0          ! COMBUSTION EFFICIENCY
21 -1.0         ! FUEL FLOW
!HP TURBINE
22 0.0          ! AUXILIARY WORK
23 -1.0         ! NDMF
24 -1.0         ! NDSPEED CN
25 0.896        ! ISENTROPIC EFFICIENCY
26 -1.0         ! PCN
27 1.0          ! COMPRESSOR NUMBER
28 4.0          ! TURBINE MAP NUMBER
29 -1.0         ! POWER LOW INDEX
!POWER TURBINE
```

```
30 27600000.00          ! AUXILIARY WORK
31 -1.0                 ! NDMF
32 -1.0                 ! NDSPEED CN
33 0.854                ! ISENTROPIC EFFICIENCY
34 -1.0                 ! PCN
35 0.0                  ! COMPRESSOR NUMBER
36 4.0                  ! MAP NUMBER
37 -1.                  ! POWER LOW INDEX
38 -1.                  ! COMWORK
!NOZCON
39 -1.                  ! THROAT AREA
!PERFOR
40 1.00                 ! PROPELLER EFFICIENCY
41 0.0                  ! SCALING INDEX
42 0.0                  ! REQUIRED THRUST
-1
1 2 80.5                ! INLET MASS FLOW
6 6 1505.0              ! COMBUSTION OUTLET TEMPERATURE
-1
-3
```

OUTPUT FILE TURBOMATCH DESIGN POINT PERFORMANCE CALCULATIONS, GE LM2500+

The Units for this Run are as follows:-

Temperature = K   Pressure = Atmospheres   Length = metres

Area = sq metres Mass Flow = kg/sec   Velocity = metres/sec

Force = Newtons   s.f.c.(Thrust) = mg/N sec   s.f.c.(Power) = mug/J

Sp. Thrust =   N/kg/sec   Power =   Watts

***** DESIGN POINT ENGINE CALCULATIONS *****

***** AMBIENT AND INLET PARAMETERS *****

Alt. =   0.0   I.S.A. Dev. =   0.000   Mach No. = 0.00
Etar = 0.9951   Momentum Drag =   0.00

***** COMPRESSOR 1 PARAMETERS *****

PRSF = 0.21667E+02   ETASF = 0.10633E+01   WASF = 0.45436E+00
Z = 0.85000   PR = 23.100   ETA = 0.88250
PCN =   1.0000   CN = 1.00000   COMWK = 0.38125E+08

***** COMBUSTION CHAMBER PARAMETERS *****

ETASF = 0.10000E+01
ETA = 1.00000   DLP = 1.7240   WFB =   1.6286

***** TURBINE 1 PARAMETERS *****

CNSF = 0.78538E+02   ETASF = 0.10391E+01   TFSF = 0.28844E+01
DHSF = 0.19144E+05
TF = 414.346   ETA = 0.88500   CN = 2.060
AUXWK = 0.00000E+00

***** TURBINE 2 PARAMETERS *****

CNSF = -0.67001E-02   ETASF = 0.10391E+01   TFSF = 0.71215E+00
DHSF = 0.18575E+05
TF = 414.346   ETA = 0.88500   CN = 2.060
AUXWK = 0.27600E+08

Additional Free Turbine Parameters:-
Speed = *****%   Power = 0.27600E+08

***** CONVERGENT NOZZLE 1 PARAMETERS *****

NCOSF = 0.10000E+01
Area = 1.5755   Exit Velocity = 112.54 Gross Thrust =   8979.77
Nozzle Coeff. = 0.97152E+00

Scale Factor on above Mass Flows, Areas, Thrusts & Powers =   1.0000

Station	F.A.R.	Mass Flow	Pstatic	Ptotal	Tstatic	Ttotal	Vel	Area
1	0.00000	80.500	1.00000	1.00000	288.15	288.15	0.0	*****
2	0.00000	80.500	******	0.99510	*****	288.15	*****	*****
3	0.00000	80.500	******	22.98682	*****	745.07	*****	****
4	0.00000	78.488	******	22.98682	*****	745.07	*****	*****
5	0.00000	72.601	******	22.98682	*****	745.07	*****	*****
6	0.02243	74.230	******	21.26281	*****	1505.00	*****	*****
7	0.02075	80.116	******	21.26281	*****	1453.53	*****	*****
8	0.02075	80.116	******	4.59106	*****	1065.23	*****	*****
9	0.02023	82.129	******	4.59106	*****	1057.85	*****	*****

10	0.02023	82.129	******	1.02953	*****	767.87	*****	*****
11	0.02023	82.129	1.00000	1.02953	762.23	767.87	112.5	1.5755
12	0.00000	2.013	*****	22.98682	*****	745.07	*****	*****
13	0.00000	5.887	******	22.98682	*****	745.07	*****	*****

Shaft Power = 27600000.00
Net Thrust = 8979.77
Equiv. Power = 28178988.00
Fuel Flow = 1.6286
S.F.C. = 59.0078
E.S.F.C. = 57.7953
Sp. Sh. Power = 342857.16
Sp. Eq. Power = 350049.56
Sh. Th. Effy. = 0.3930

INPUT FILE TURBOMATCH OFF DESIGN PERFORMANCE CALCULATIONS

```
INDUSTRIAL GAS TURBINE SIMULATION
GENERAL ELECTRIC  LM2500 PLUS
MODELLED BY HUSSEIN BEN-HARIZ, SME, DEC 2007////
OD SI NG CT FP
-1
-1
INTAKE          S1-2     D1-4          R100
COMPRE          S2-3     D5-10         R101     V5  V6
PREMAS          S3,12,4  D11-14
PREMAS          S4,13,5  D15-18
BURNER          S5-6     D19-21        R102
MIXEES          S6,13,7
TURBIN          S7-8     D22-29,101             V23
MIXEES          S8,12,9
TURBIN          S9-10    D30-38                V30 V31
NOZCON          S10-11,1 D39           R107
PERFOR          S1,0,0   D30,40-42,107,100,102,0,0,0,0,0,0
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 23.1          ! PRESSURE RATIO
8 0.8825        ! ISENTROPIC EFFICIENCY
9 0.0           ! ERROR SELECTION
10 4.0          ! MAP NUMBER
!PREMAS
11 0.025        ! BLEED AIR
12 0.00         ! FLOW LOSS
13 1.0          ! PRESSURE RECOVERY
14 0.0          ! PRESSURE DROP
!PREMAS
15 0.075         ! BLEED AIR
16 0.0          ! FLOW LOSS
17 1.0          ! PRESSURE RECOVERY
18 0.0          ! PRESSURE DROP
!BURNER
19 0.075        ! FRACTIONAL PRESSURE LOSS DP/P
20 1.0          ! COMBUSTION EFFICIENCY
21 -1.0         ! FUEL FLOW
!HP TURBINE
22 0.0          ! AUXILIARY WORK
23 -1.0         ! NDMF
24 -1.0         ! NDSPEED CN
```

```
25 0.885            ! ISENTROPIC EFFICIENCY
26 -1.0             ! PCN
27 1.0              ! COMPRESSOR NUMBER
28 4.0              ! TURBINE MAP NUMBER
29 -1.0             ! POWER LOW INDEX
!POWER TURBINE
30 27600000.00  ! AUXILIARY WORK
31 -1.0             ! NDMF
32 -1.0             ! NDSPEED CN
33 0.885            ! ISENTROPIC EFFICIENCY
34 -1.0             ! PCN
35 0.0              ! COMPRESSOR NUMBER
36 4.0              ! MAP NUMBER
37 -1.              ! POWER LOW INDEX
38 -1.              ! COMWORK
!NOZCON
39 -1.              ! THROAT AREA
!PERFOR
40 1.00             ! PROPELLER EFFICIENCY
41 0.0              ! SCALING INDEX
42 0.0              ! REQUIRED THRUST
-1
1 2  80.5           ! INLET MASS FLOW
6  6 1505.0         ! COMBUSTION OUTLET TEMPERATURE
-1
-1
6  6  1250.0        ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0        ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0        ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0        ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0        ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0        ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0        ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650          ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700          ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    -40.0          ! --New OD Calculation; DT=-40.0
-1
6  6  1250.0        ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0        ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0        ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0        ! OD Calculation; DT=0;  TET = 1400.0K
```

```
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    -30.0            ! --New OD Calculation; DT=-30.0
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0         ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    -20.0            ! --New OD Calculation; DT=-20.0
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0         ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
```

```
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    -10.0           ! --New OD Calculation; DT=-10.0
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0         ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0          ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2     10.0           ! --New OD Calculation; DT=10
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0         ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
```

```
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    20.0             ! --New OD Calculation; DT=20
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0         ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0          ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650            ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700           ! OD Calculation; DT=0;  TET = 1700.0K
-1
2    30.0             ! --New OD Calculation; DT=30
-1
6  6  1250.0         ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0         ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0         ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0          ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0         ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0         ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0         ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650           ! OD Calculation; DT=0;  TET = 1650.0K
```

```
-1
-1
6  6  1700            ! OD Calculation; DT=0;  TET = 1700.0K
-1
2     40.0             ! --New OD Calculation; DT=40
-1
6  6  1250.0          ! OD Calculation; DT=0;  TET = 1250.0K
-1
-1
6  6  1300.0          ! OD Calculation; DT=0;  TET = 1300.0K
-1
-1
6  6  1350.0          ! OD Calculation; DT=0;  TET = 1350.0K
-1
-1
6  6  1400.0           ! OD Calculation; DT=0;  TET = 1400.0K
-1
-1
6  6  1450.0          ! OD Calculation; DT=0;  TET = 1450.0K
-1
-1
6  6  1500.0          ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
6  6  1550.0          ! OD Calculation; DT=0;  TET = 1550.0K
-1
-1
6  6  1600.0          ! OD Calculation; DT=0;  TET = 1600.0K
-1
-1
6  6  1650            ! OD Calculation; DT=0;  TET = 1650.0K
-1
-1
6  6  1700            ! OD Calculation; DT=0;  TET = 1700.0K
-1
-3
```

A.2 INPUT FILE TURBOMATCH DESIGN POINT PERFORMANCE CALCULATIONS SGT200 TORNADO

```
! Simulation of SGT200TORNADO-SIEMANS Gas turbine and Power Turbine
DP SI NG VA FP
 -1
 -1
INTAKE S1, 2            D1-4         R100
COMPRE S2, 3            D5-11        R102    V5    V6
PREMAS S3, 4, 22        D12-15
DUCTER S4, 5            D16-19       R103    W6, 5
BURNER S5, 6            D20-22       R104
MIXEES S6, 22, 7
TURBIN S7, 8            D23-30, 102, 31      V24
DUCTER S8, 9            D32-35       R107
TURBIN S9, 10           D36-45           V37   V36
DUCTER S10, 11          D46-49       R110
NOZCON S11, 12, 1       D50          R111
PERFOR  S1,0,0          D36,51-53,111,100,104,0,0,0,0,0,0,0,0,0
CODEND
DATA ITEMS////
! Intake
1 0.0                   ! INTAKE DATA: ALTITUDE
2 0.0                   ! DEV FROM STANDART TEMP
3 0.0                   ! MA-NUMBER
4 0.99                  ! PRESSURE RECOVERY
! Compressor
5 -1.0                  ! COMP: Z
6 1.0                   ! RELATIVE ROTATIONAL SPEED
7 12.4                  ! PRESSURE RATIO
8 0.84                  ! ISENTROPIC EFFICIENCY
9 0.0                   ! ERROR SWITCH
10 3.0                  ! MAP-NUMBER
11 0.0                  ! STATOR ANGLE RELATIVE TO DP
! Premas
12 0.98                 ! Premas: Cooling bypass: LAMBDA W
13 0.0                  ! DELTA W
14 1.0                  ! LAMBDA P
15 0.0                  ! DELTP
! Ducter
16 0.0                  ! Divergent Diffuser: Switch
17 0.002                ! TOTAL PRESSURE LOSS: DELTA (P)/Pin
18 0.0                  ! COMBUSTION EFFICIENCY
19 0.0                  ! LIMITING VALUE FOR FUEL FLOW: NO = 100000
! Burner
20 0.065                ! BURNER: PRESSURE LOSS
21 0.995                ! COMB. EFF.
22 -1.0                 ! FUEL FLOW
! Compressor Turbine
23 0.0                  ! TURBINE DATA: AUXWORK
24 0.8                  ! DESIGN NON DIM FLOW / MAX
25 0.6                  ! DESIGN NON DIM SPEED
26 0.88                 ! ISENTROPIC EFF
27 -1.0                 ! ROT SPEED OF PT
28 1.0                  ! NUMBER OF COMPRESSOR DRIVEN
29 5.0                  ! MAP NUMBER
30 -1.0                 ! POWER LAW INDEX
31 0.0                  ! NGV ANGLE RELATIVE TO DP
! Ducter
32 0.0                  ! DIVERGENT-CONVERGENT DUCT AT TURBINE AND POWERTURBIN:
33 0.0                  ! Total pressure loss: DELTA (P)/Pin
34 0.0                  ! COMBUSTION EFFICIENCY
35 0.0                  ! LIMITING VALUE FOR FUEL FLOW :NO = 100000
! Power Turbine
36 7680000.0            ! POWER TURBINE DATA: AUXWORK
37 -1.0                 ! DESIGN NON DIM FLOW / MAX
38 -1.0                 ! DESIGN NON DIM SPEED
```

```
39 0.88                  ! ISENTROPIC EFF
40 1.0                   ! ROT SPEED OF PT
41 0.0                   ! NUMBER OF COMP DRIVEN
42 1.0                   ! MAP NUMBER
43 -1.0                  ! POWER LAW INDEX
44 -1.0                  ! Compressor Work: Power turbine = -1
45 0.0                   ! NGV
! Ducter
46 0.0                   ! GAS COLLECTOR DUCT AFTER POWER TURBINE: Switch
47 0.01                  ! Total pressure loss: DELTA (P)/Pin
48 0.0                   ! Combustion efficiency
49 0.0                   ! Limiting value for Fuel Flow: no = 100000
! Nozzle
50 -1.0                  ! Convergent nozzle
! Performance
51 1.0                   ! POWER TURBINE:  PROPELLER EFF
52 0.0                   ! SCALING SWITCH
53 0.0                   ! REQUIRED THRUST at Design point
54 0.0                   ! Nozzle Gross Thrust
-1
1 2   29.3               ! Airflow
6 6   1314.0             ! TET
-1
-3
```

A.3 INPUT FILE TURBOMATCH DESIGN POINT PERFORMANCE CALCULATIONS SULZER 7

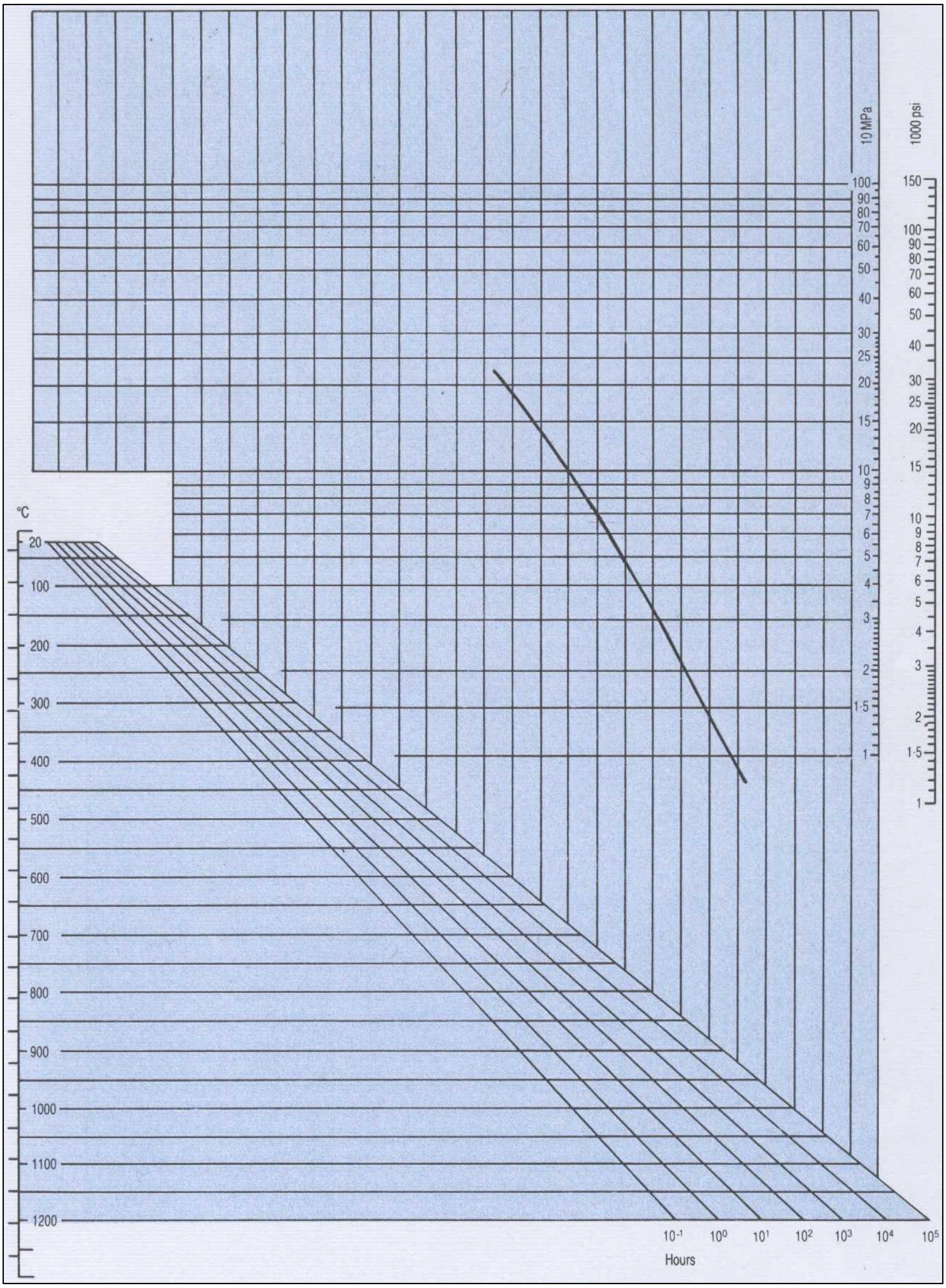
```
DP SI NG CT FP
-1
-1
INTAKE S1-2       D1-4              R300
COMPRE S2-3       D5-11             R301      V5   V6
PREMAS S3,4,9     D12-15
BURNER S4-5       D16-18            R303
MIXEES S5,9,6
TURBIN S6-7       D19-26,301,27               V19  V20
NOZCON S7-8,1     D28               R305
PERFOR S1,0,0     D19,29-31,305,300,303,0,0,0,0,0,0,0,0
CODEND

BRICK DATA ITEMS////
! INTAKE
1 0.0         ! ALTITUDE
2 0.0         ! ISA DEVIATION:Tamb=288.15 K, Pamb=1.01325 bar
3 0.0         ! MACH NUMBER
4 0.9951      ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0        ! SURGE MARGIN
6 0.6         ! DESIGN SPEED
7 7.6         ! DESIGN PRESSURE RATIO
8 0.885       ! ISENTRIOPIC EFFICIENCY
9 1.0         ! ERROR SELECTION
10 3.0        ! COMPRESSOR MAP NUMBER
11 0.0        ! RELATIV TO DP VARIABLE STATOR ANGLE
! SPLITTER
12 0.96       ! LAMBDA (W)
13 0.0        ! DELTA (W)
14 1.0        ! LAMBDA (P)
15 0.0        ! DELTA (P)
! BURNER
16 0.07       ! PRESSURE LOSS 0%
17 0.998      ! EFFICIENCY
18 -1.0       ! FUEL FLOW (-1 = TET SPECIFIED. SEE SV DATA)
! POWER TURBINE
19 10600000.0  ! AUXILLARY POWER REQUIRED
20 0.8        ! NON DIMENTIONAL MASS FLOW
21 0.6        ! NON DIMENTIONAL SPEED
22 0.90       ! ISENTROPIC EFFICIENCY
23 -1.0       ! RELATIV ROTATIONAL SPEED
24 1.0        ! COMPRESSOR NUMBER
25 3.0        ! TURBINE MAP NUMBER
26 1000.0     ! POWER INDEX N
27 0.0        ! ANGLE
! NOZCON
28 -1.0       ! AREA FIXED
! PERFORMANCE
29 1.0        ! PROPELLER EFFICIENCY
30 0.0        ! SCALLING INDEX
31 0.0        ! REQUIRED DP NET THRUST OR POWER OUTPUT FOR PT

-1
1 2 64.3     ! MASS FLOW
5 6 1198.0  ! TURBINE ENTRY TEMPERATURE
-1
```

APPENDIX B

Creep rupture properties of INCO alloy HX sheet



APPENDIX C

Engines prices (2005), [93]

Gas Turbine	ISO Base Load	Heat Rate Btu/kW-hr	LHV Efficiency	Budget Price	$ per kW
501-KB7	5275 kW	11,200 Btu	30.5%	$1,750,000	$332
M7A-01	5840 kW	11,230 Btu	30.4%	$2,310,000	$396
PGT5B	5900 kW	10,700 Btu	31.9%	$2,050,000	$347
GTES-6	6000 kW	12,640 Btu	27.0%	$1,795,000	$299
501-KH5 (steam injection)	6420 kW	8560 Btu	39.9%	$2,300,000	$358
601-KB9	6450 kW	10,615 Btu	32.1%	$2,450,000	$380
GT6001	6700 kW	10,840 Btu	31.5%	$2,700,000	$403
UGT-6000	6700 kW	11,270 Btu	30.3%	$2,100,000	$313
Tornado.	6750 kW	10,820 Btu	31.5%	$2,750,000	$407
M7A-02	6960 kW	11,050 Btu	30.9%	$2,700,000	$388
Taurus 70	7520 kW	10,100 Btu	33.8%	$2,670,000	$355
Tempest	7710 kW	11,265 Btu	30.3%	$2,995,000	$388
601-KB11	7920 kW	10,350 Btu	33.0%	$3,200,000	$404
UGT-6000+	8300 kW	10,650 Btu	32.0%	$2,350,000	$283
THM1304-11	10,080 kW	11,760 Btu	29.0%	$3,950,000	$392
G3142J	10,450 kW	13,320 Btu	25.6%	$3,750,000	$359
Mars 100	10,690 kW	10,520 Btu	32.4%	$4,000,000	$374
UGT-10000	10,800 kW	9750 Btu	35.0%	$3,350,000	$310
PGT10B	11,700 kW	10,660 Btu	32.0%	$4,700,000	$402
GTES-12	12,000 kW	9890 Btu	27.0%	$3,588,000	$299
Cyclone DLE	12,875 kW	10,070 Btu	33.9%	$4,980,000	$387
Titan 130	13,500 kW	10,250 Btu	33.3%	$4,500,000	$335
SB60-1	13,570 kW	11,490 Btu	29.7%	$5,930,000	$437
PGT16	13,750 kW	9670 Btu	35.3%	$6,750,000	$491

Gas Turbine	ISO Base Load	Heat Rate Btu/kW-hr	LHV Efficiency	Budget Price	$ per kW
LM2500PH .... (steam injection)	.28,280 kW	8325 Btu	41.0%	$11,500,000	$407
LM2500+PK	.28,600 kW	8860 Btu	38.5%	$12,500,000	$437
RB211-6562	.28,775 kW	9225 Btu	37.0%	$9,300,000	$323
GT10C	.29,060 kW	9480 Btu	36.0%	$10,900,000	$375
MF-221	.30,000 kW	10,670 Btu	32.0%	$10,000,000	$333
RB211-6761.	.31,750 kW	8735 Btu	39.1%	$10,300,000	$324
IM5000	.33,550 kW	9210 Btu	37.1%	$12,900,000	$384
PG6561B	.39,620 kW	10,710 Btu	31.9%	$13,000,000	$328
TG20 B7/8	.39,970 kW	11,200 Btu	30.5%	$12,500,000	$313
UGT-25000 STIG. ..... (steam injection)	.40,100 kW	7990 Btu	42.7%	$8,200,000	$204
PG6581B	.42,100 kW	10,640 Btu	32.1%	$14,500,000	$344
LM6000PC	.42,620 kW	8250 Btu	41.4%	$14,100,000	$331
GTX100.	.43,000 kW	9215 Btu	37.0%	$13,800,000	$321
LM6000PD	.43,400 kW	8200 Btu	41.6%	$15,000,000	$346
LM6000PD(DLE)	.43,400 kW	8200 Btu	41.6%	$15,800,000	$364
LM6000PC Sprint	.47,300 kW	8200 Btu	41.6%	$15,700,000	$332
W251B11/12	.49,500 kW	10,450 Btu	32.6%	$13,900,000	$280
IM5000-STIG .... (steam injection)	.51,160 kW	7780 Btu	43.9%	$14,900,000	$291
Trent DLE	.51,190 kW	8210 Btu	41.6%	$15,500,000	$303
FT8 Twin	.51,300 kW	8885 Btu	38.4%	$16,500,000	$322
GT8C	.52,800 kW	9920 Btu	34.4%	$15,300,000	$290
GT8C2	.57,200 kW	9830 Btu	34.7%	$15,900,000	$278
V64.3	.63,000 kW	9690 Btu	35.2%	$17,500,000	$278
V64.3A.	.67,000 kW	9830 Btu	34.7%	$20,400,000	$304

Gas Turbine	ISO Base Load	Heat Rate Btu/kW-hr	LHV Efficiency	Budget Price	$ per kW
LM1600PA	13,750 kW	9865 Btu	34.6%	$8,000,000	$582
LM1600DLE	13,750 kW	9865 Btu	34.6%	$8,500,000	$618
H-15	13,800 kW	11,010 Btu	31.0%	$6,300,000	$456
MF111B.	14,570 kW	11,020 Btu	31.0%	$6,200,000	$425
Avon	14,580 kW	12,100 Btu	28.2%	$5,200,000	$357
GTES-16	16,000 kW	9225 Btu	37.0%	$4,784,000	$299
UGT-10000 STIG. (steam injection)	16,000 kW	7950 Btu	43.0%	$4,500,000	$281
UGT-16000	16,300 kW	11,230 Btu	30.4%	$3,950,000	$242
LM1600-PB STIG. (steam injection)	16,900 kW	8605 Btu	39.7%	$8,280,000	$490
GT35	17,000 kW	10,600 Btu	32.2%	$6,500,000	$382
GT15000	17,500 kW	9750 Btu	35.0%	$6,275,000	$359
UGT-15000	17,500 kW	9970 Btu	34.2%	$4,950,000	$283
LM2000	18,000 kW	9615 Btu	35.5%	$7,950,000	$440
UGT-15000+	20,000 kW	9970 Btu	34.2%	$5,500,000	$275
PGT25	22,450 kW	9395 Btu	36.3%	$9,900,000	$441
LM2500PE	22,800 kW	9280 Btu	36.8%	$11,000,000	$482
GT10B	24,770 kW	9985 Btu	34.2%	$8,942,000	$361
UGT-15000 STIG. (steam injection)	25,000 kW	8100 Btu	42.1%	$6,200,000	$248
RB211-6556	25,360 kW	9745 Btu	35.0%	$8,750,000	$346
FT8	25,470 kW	8950 Btu	38.1%	$9,730,000	$382
UGT-25000	26,200 kW	9550 Btu	35.7%	$6,800,000	$260
PG5371PA.	26,300 kW	11,990 Btu	28.5%	$7,680,000	$292
H-25	26,900 kW	10,280 Btu	33.2%	$8,300,000	$309
GT25000	27,500 kW	9710 Btu	35.1%	$9,270,000	$337

Ambient Temperature and Power Demand Profiles for different Seasons [91]

Time, Hour	T Amb, C (Summer)	Power Demand, MW (Summer)	T Amb, C (Autumn)	Power Demand, MW (Autumn)	T Amb, C (Winter)	Total Power Demand, (Winter)	T Amb, C (Spring)	Power Demand, MW (Spring)
0	13	113	10	117	5	127	7	125
2	13	113	10	117	5	127	7	125
4	14	116	12	115	6	127	9	124
6	19	125.5	14	115	9	119	13	121
8	24	129	19	125	16	124	19	125
10	33	131	27	130	23	128	26	129
12	47	136	38	133	34	130	36	132
14	47	136	38	133	34	130	36	132
16	42	134	34	131	30	129	33	131
18	34	135	28	132	25	131	27	130
20	23	133	23	130	16	130	20	129
22	17	123.5	15	121	8	128	14	120

APPENDIX D

APPENDIX E

Operational Procedure for Typical Summer Day (Existing & Proposed)

TIME, HOUR	AMBIENT TEMP, ^{0}C	POWER DEMAND, MW	Running Engines (Existing)	Proposed Operation
0	13	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 10MW
2	13	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = MW
4	14	116	LM2500+ 3 X 19 = 57 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 3.66= 11 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 9.67MW Sulzer 3, 1 x 3.33= 3.33 MW
6	19	125.5	LM2500+ 3 X 19 = 57 MW Tornado, 3x7.5 = 22.5 MW Tornado, 1x7 = 7 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.25= 4.25 MW
8	24	129	LM2500+ 3 X 19 = 57 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x8 =8 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
10	33	131	LM2500+ 3 X 19.3 = 58MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22.5 = 67.5 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.9= 19.5 MW
12	47	136	LM2500+ 3 X 21 = 63 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22 = 66 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.6= 18 MW
14	47	136	LM2500+ 3 X 21 = 63 MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 22 = 66 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.6= 18 MW
16	42	134	LM2500+ 2 X 20 = 40MW LM2500+ 1 X 21 = 21MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 21.65 = 64.9 MW Tornado, 4 x5.1 = 20.4 MW Sulzer 7, 4 x8.78 =35.12 MW Sulzer 3, 3 x 4.52= 13.56 MW
18	34	135	LM2500+ 2 X 21 = 42MW LM2500+ 1 X 20 = 20MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 21.94 = 65.8 MW Tornado, 4 x5.1 = 20.5 MW Sulzer 7, 4 x8.78 =35.12 MW Sulzer 3, 3 x 4.52= 13.56 MW
20	23	133	LM2500+ 3 X 20 = 60MW Tornado, 4 x7.5 = 30 MW Sulzer 7, 2 x9.5 =19 MW Sulzer 7, 1 x9=9 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 24.7 = 74.1 MW Tornado, 4 x5.87 = 23.48 MW Sulzer 7, 4 x8.86 =35.44 MW Sulzer 3, 5 x 3.2= 16 MW
22	17	123.5	LM2500+ 2 X 18 = 36 MW LM2500+ 1 X 19 = 19 MW Tornado, 3x7.5 = 22.5 MW Tornado, 1x7 = 7 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 3 x 5= 15 MW	LM2500+ 3 X 25 = 75 MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW

Operational Procedure for Typical Autumn Day (Existing & Proposed)

TIME, HOUR	AMBIENT TEMP, ^{0}C	POWER DEMAND, MW	Running Engines (Existing)	Proposed Operation
0	7	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 10MW
2	7	113	LM2500+ 3 X 18 = 54 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = MW
4	9	119	LM2500+ 3 X 20 = 60 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 9.67MW Sulzer 3, 2 x 3..16= 6.33 MW
6	12	123	LM2500+ 3 X 21 = 63 MW Tornado, 4x5.25 = 21 MW Sulzer 7, 3 x8 =24 MW Sulzer 3, 5 x 3= 15 MW	LM2500+ 3 X 25 = 75 MW Tornado, 4 x6.25 = 25 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW
8	17	125	LM2500+ 3 X 20 = 60 MW Tornado, 4x6.5 = 26 MW Sulzer 7, 4 x6 =24 MW Sulzer 3, 5 x 3= 15 MW	LM2500+ 3 X 25.3 = 76 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.25= 4.25 MW
10	26	127.5	LM2500+ 3 X 20.3 = 62 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7=28 MW Sulzer 3, 5 x 3.5= 17.5 MW	LM2500+ 3 X 25.3 = 76 MW Tornado, 4 x6.25 = 25 MW Sulzer 7, 2x9.5 = 18 MW Sulzer 3, 2 x 4.25= 8.5 MW
12	34	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 22.3 = 67 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.8= 19 MW
14	34	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 22.3 = 67 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.8= 19 MW
16	30	128	LM2500+ 3 X 25.2 = 75.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 23.5 = 70.5 MW Tornado, 4 x4.5 = 18 MW Sulzer 7, 3x7 = 21 MW Sulzer 3, 5 x 3.7= 18.5 MW
18	23	129	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
20	18	124	LM2500+ 3 X 25.2 = 75.5MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW	LM2500+ 3 X 25.66 = 77 MW Tornado, 4 x6 = 24 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.= 4 MW
22	11	119	LM2500+ 3 X 20 = 60 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x7.12 = 28.5 MW Sulzer 7, 1x10 = 10MW Sulzer 3, 1 x 4=4 MW

Operational Procedure for Typical Winter Day (Existing & Proposed)

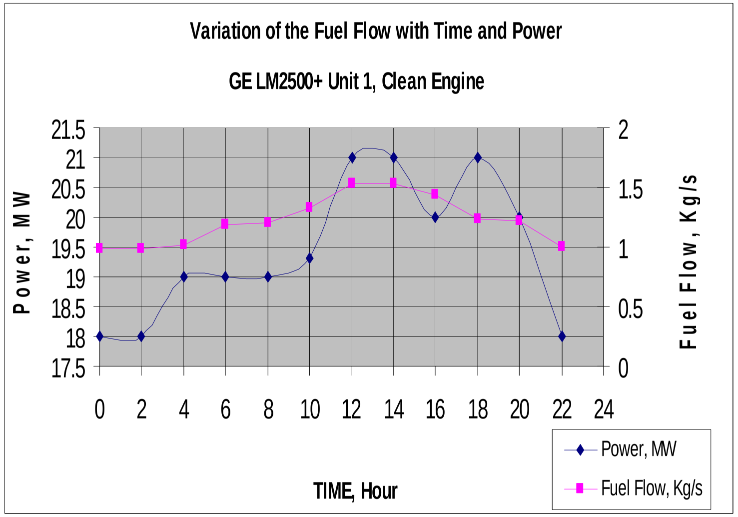
TIME, HOUR	AMBIENT TEMP, ^0C	POWER DEMAND, MW	Running Engines	Proposed Operation
0	5	127	LM2500+ 3 X 20.3 = 62 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7=28 MW Sulzer 3, 5 x 3.4= 17 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 3x8 = 24 MW
2	5	127	LM2500+ 3 X 20.3 = 62 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7=28 MW Sulzer 3, 5 x 3.4= 17 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 3x8 = 24 MW
4	6	127	LM2500+ 3 X 20.3 = 62 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7=28 MW Sulzer 3, 5 x 3.4= 17 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 3x8 = 24 MW
6	9	119	LM2500+ 3 X 20 = 60 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.62 = 26.5 MW Sulzer 7, 1x10 = 9.67MW Sulzer 3, 2 x 3..16= 6.33 MW
8	16	124	LM2500+ 3 X 25.2 = 75.5MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW	LM2500+ 3 X 25 = 75 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.25= 4.25 MW
10	23	128	LM2500+ 3 X 25.2 = 75.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.75 = 27 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
12	34	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 22.3 = 67 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.8= 19 MW
14	34	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 22.3 = 67 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.8= 19 MW
16	30	129	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
18	25	131	LM2500+ 3 X 21 = 63 MW Tornado, 4 x4.87 = 19 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.4= 17MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 3 x 3.5= 10.5 MW
20	16	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.33 = 76 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 19 MW Sulzer 3, 2 x 4.5= 9 MW
22	8	128	LM2500+ 3 X 25.2 = 75.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9 = 18 MW Sulzer 3, 2 x 4= 8 MW

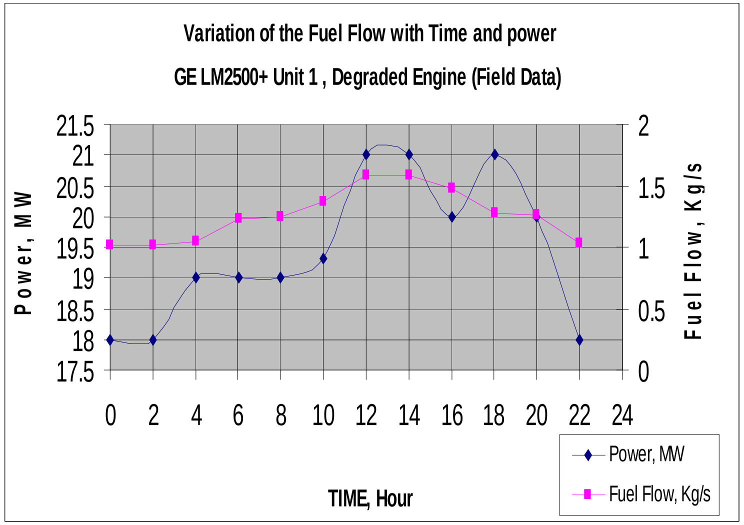
Operational Procedure for Typical Spring Day (Existing & Proposed)

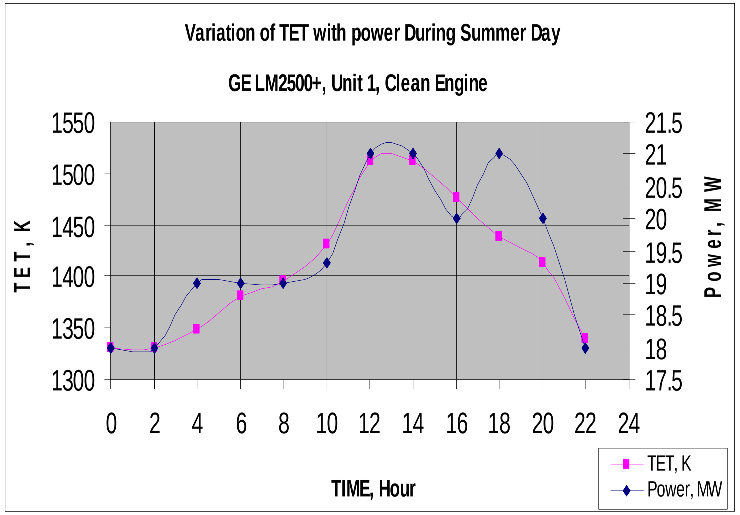
TIME, HOUR	AMBIENT TEMP, ^{0}C	POWER DEMAND, MW	Running Engines	Proposed Operation
0	7	125	LM2500+ 3 X 20 = 60 MW Tornado, 4x6.5 = 26 MW Sulzer 7, 4 x6 =24 MW Sulzer 3, 5 x 3= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.25 = 25 MW Sulzer 7, 3x7.83 = 23.5 MW
2	7	125	LM2500+ 3 X 20 = 60 MW Tornado, 4x6.5 = 26 MW Sulzer 7, 4 x6 =24 MW Sulzer 3, 5 x 3= 15 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.25 = 25 MW Sulzer 7, 3x7.83 = 23.5 MW
4	9	124	LM2500+ 3 X 25.2 = 75.5MW Tornado, 4 x6.37 = 25.5 MW Sulzer 7, 2x9.5 = 19MW Sulzer 3, 1 x 4= 4 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6 = 24 MW Sulzer 7, 3x7.83 = 23.5 MW
6	12	121	LM2500+ 3 X 20.6 = 62 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25.33 = 76 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 19MW
8	19	125	LM2500+ 3 X 20 = 60 MW Tornado, 4x6.5 = 26 MW Sulzer 7, 4 x6 =24 MW Sulzer 3, 5 x 3= 15 MW	LM2500+ 3 X 25.3 = 76 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.37 = 19MW Sulzer 3, 1 x 4.25= 4.25 MW
10	26	129	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
12	36	132	LM2500+ 3 X 21.6 = 65 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7.75 =31 MW Sulzer 3, 5 x 3.2= 16 MW	LM2500+ 3 X 21.6 = 65 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7.75 =31 MW Sulzer 3, 5 x 3.2= 16 MW
14	36	132	LM2500+ 3 X 21.6 = 65 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7.75 =31 MW Sulzer 3, 5 x 3.2= 16 MW	LM2500+ 3 X 21.6 = 65 MW Tornado, 4 x5 = 20 MW Sulzer 7, 4 x7.75 =31 MW Sulzer 3, 5 x 3.2= 16 MW
16	33	131	LM2500+ 3 X 21 = 63 MW Tornado, 4 x4.87 = 19 MW Sulzer 7, 4 x8 =32 MW Sulzer 3, 5 x 3.4= 17MW	LM2500+ 3 X 22.5 = 67.5 MW Tornado, 4 x5 = 20 MW Sulzer 7, 3x8 = 24MW Sulzer 3, 5 x 3.9= 19.5 MW
18	27	130	LM2500+ 3 X 25.8 = 77.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.625 = 26.5 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.5= 9 MW
20	20	129	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW	LM2500+ 3 X 25.5 = 76.5 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 18MW Sulzer 3, 2 x 4.25= 8.5 MW
22	14	120	LM2500+ 3 X 20.3 = 61 MW Tornado, 4 x5.125 = 20.5 MW Sulzer 7, 4 x7 =28 MW Sulzer 3, 3 x 3.5= 10.5 MW	LM2500+ 3 X 25 = 75 MW Tornado, 4 x6.5 = 26 MW Sulzer 7, 2x9.5 = 19MW

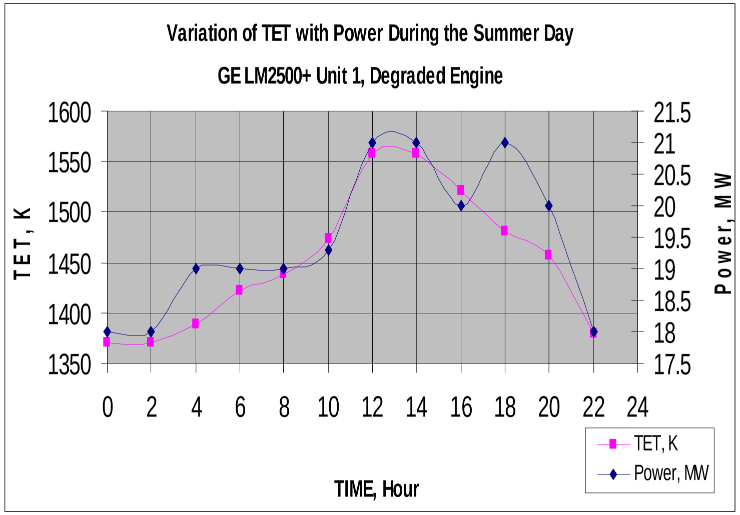
Engines Performance for the Current and Proposed Operational Strategy

A - GE LM2500+ Unit 1, Operating Condition



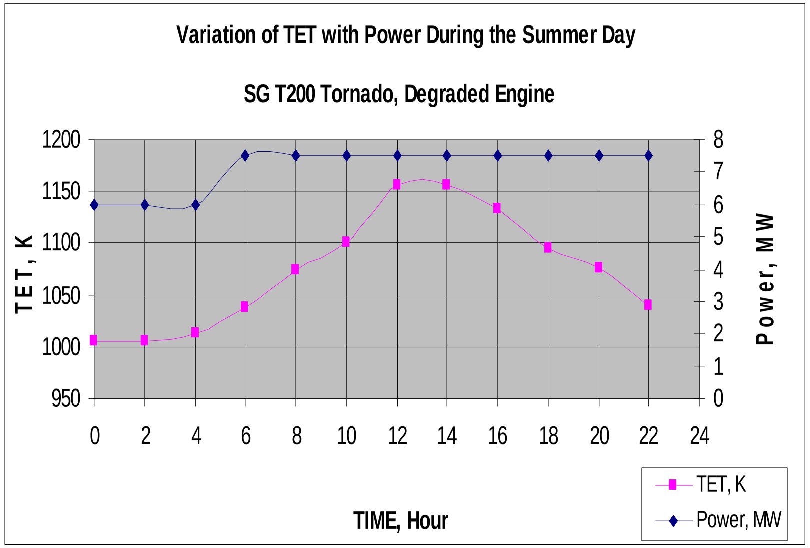


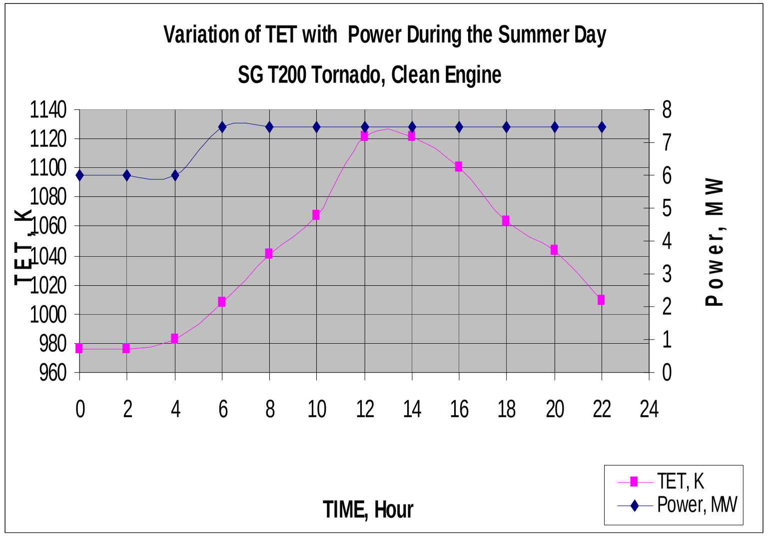


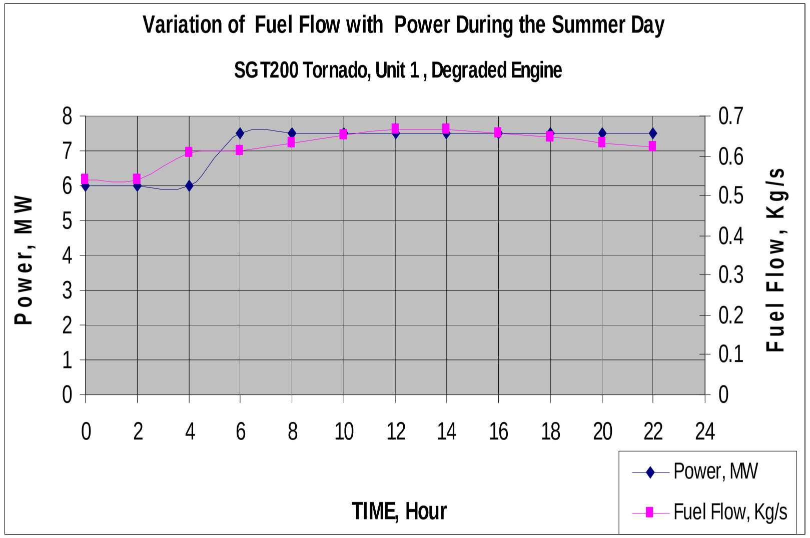


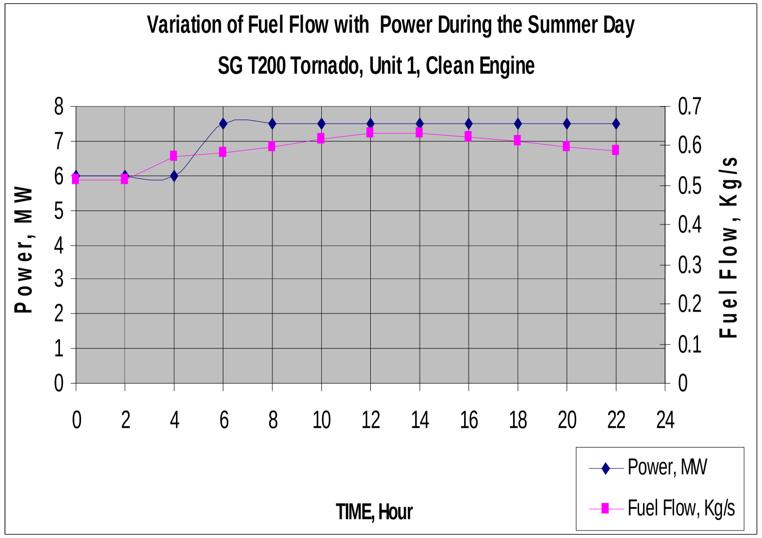
APPENDIX F

B - SG T200 Tornado Operating Condition

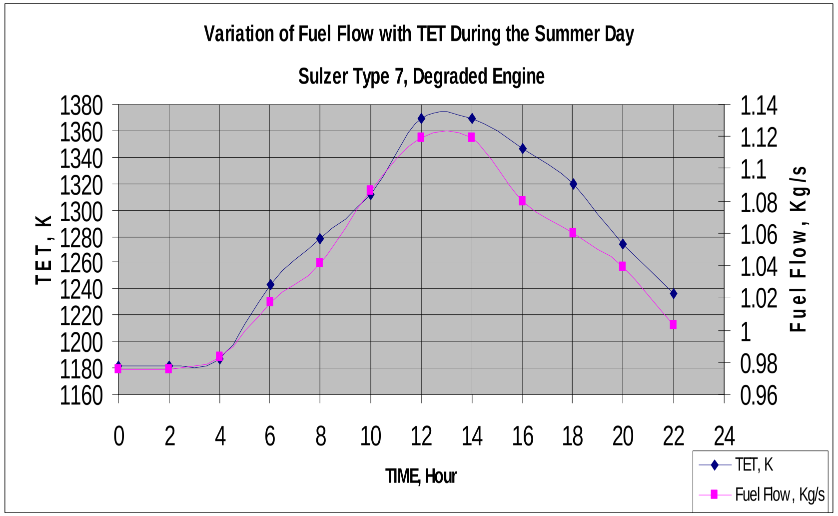


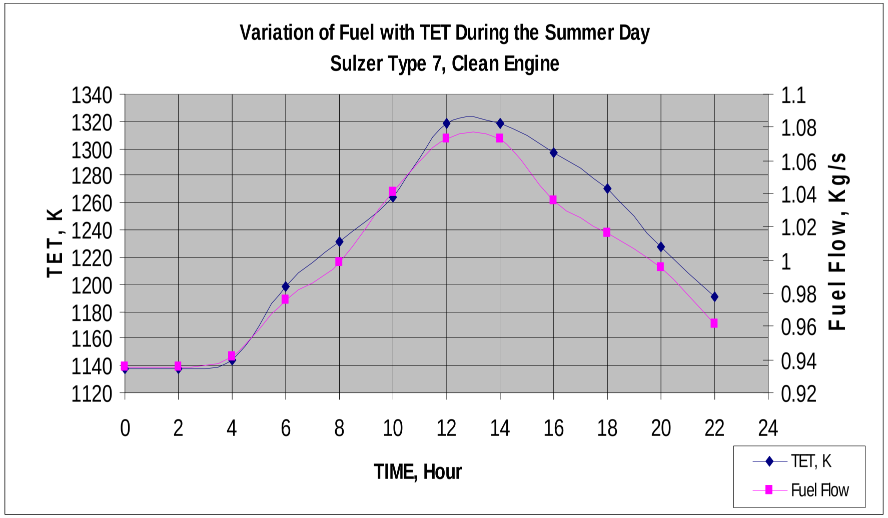


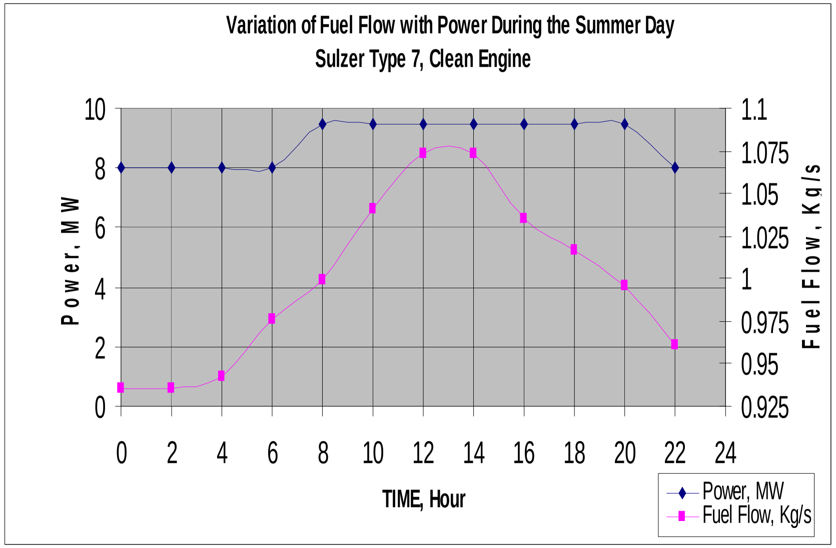


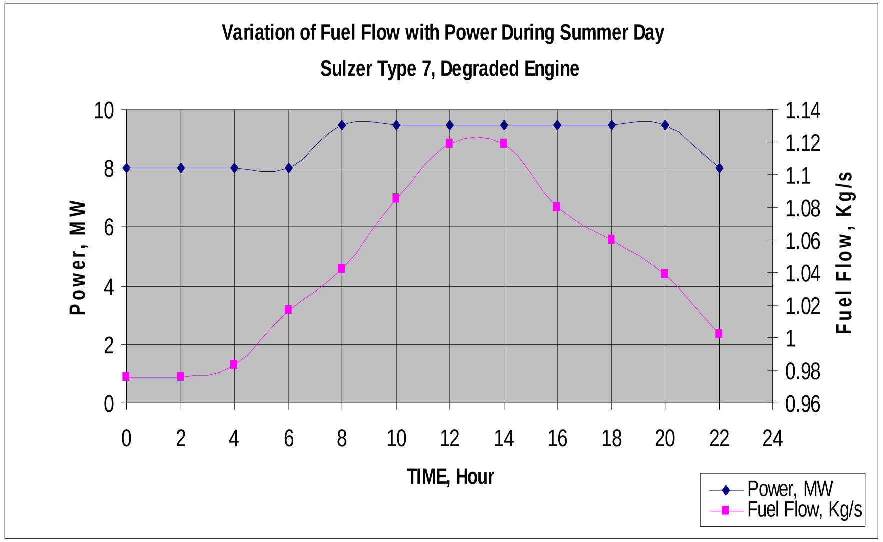


C- Sulzer Type 7, Unit 1, Operating Condition

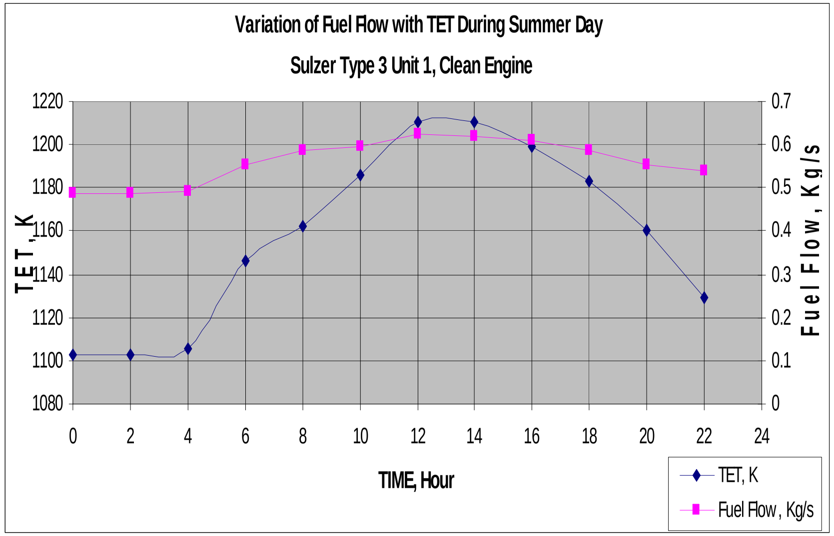


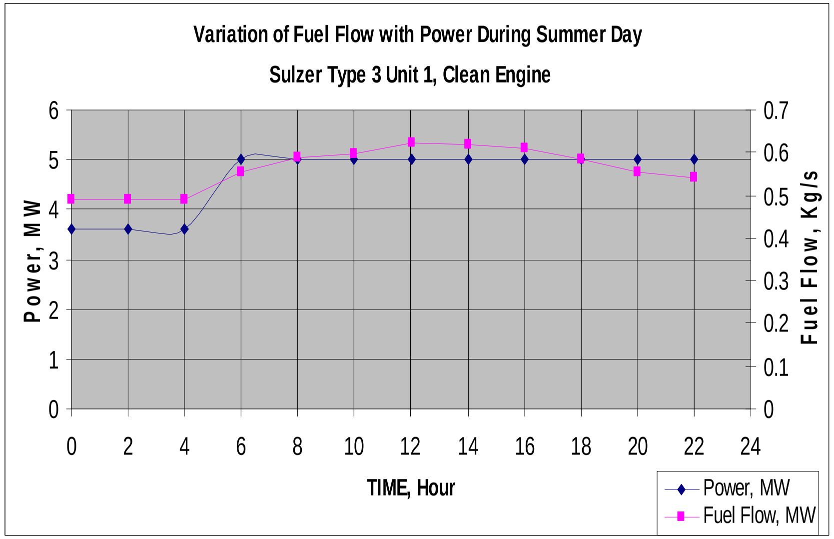


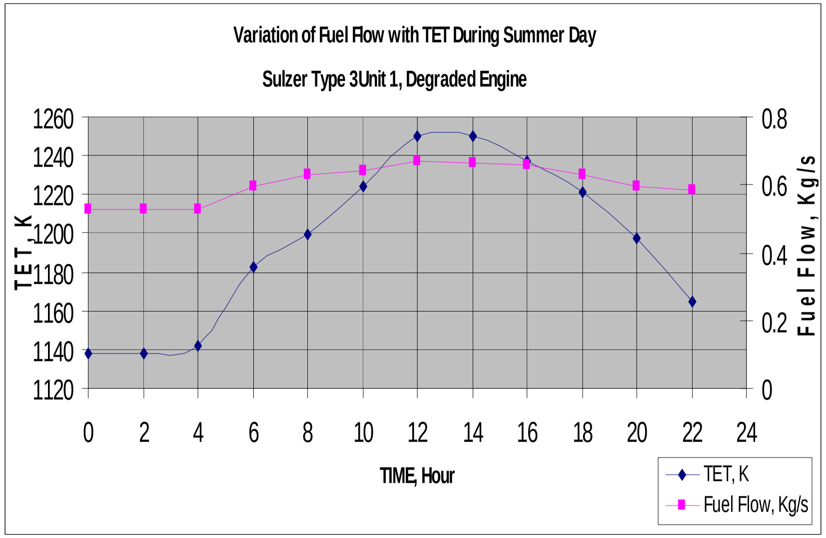


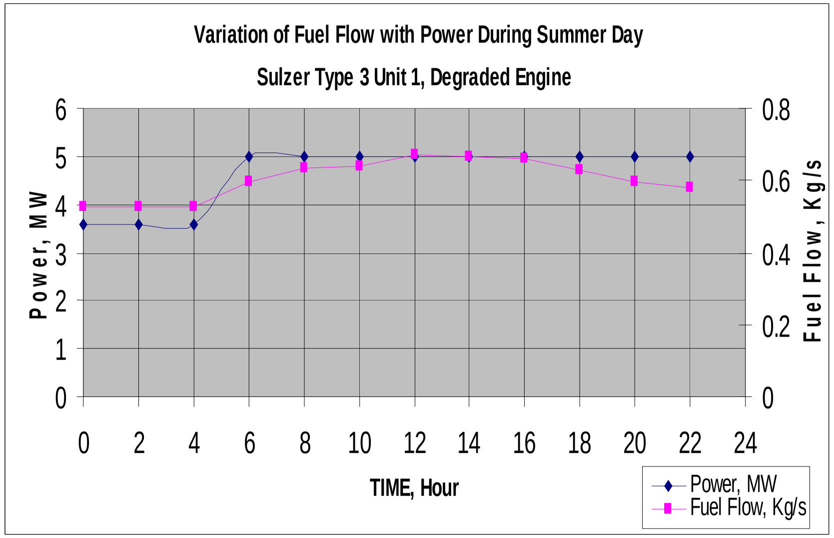


D- Sulzer Type 3, Unit 1, Operating Condition

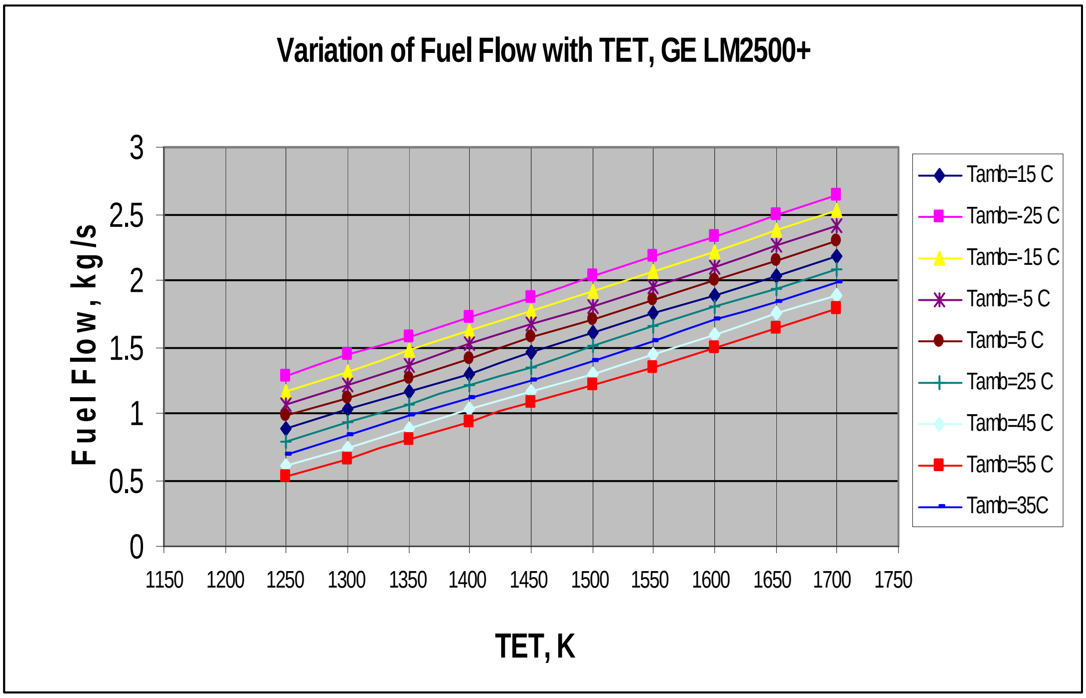




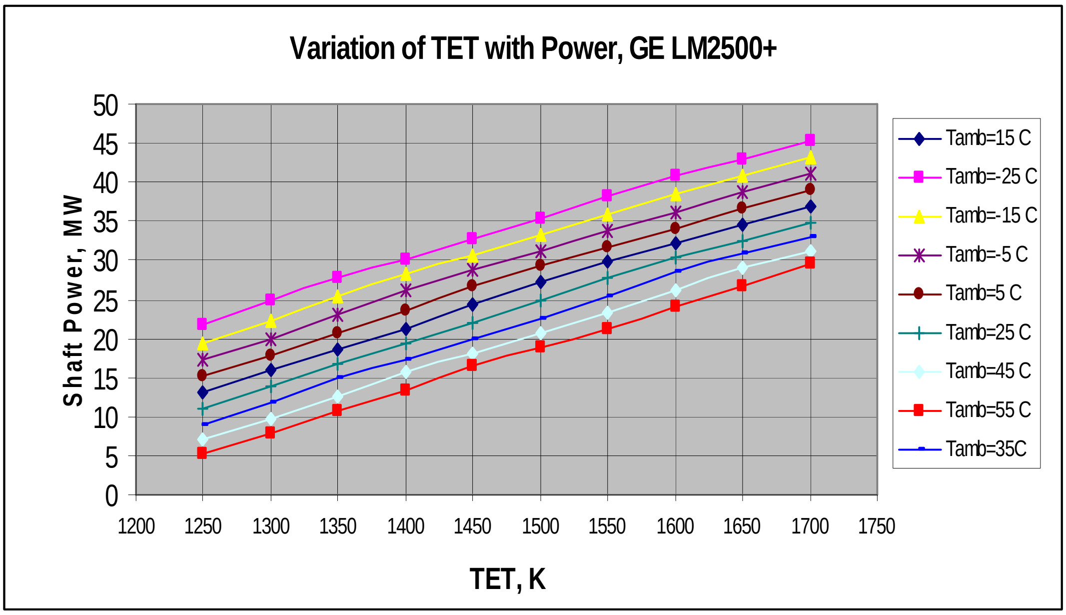




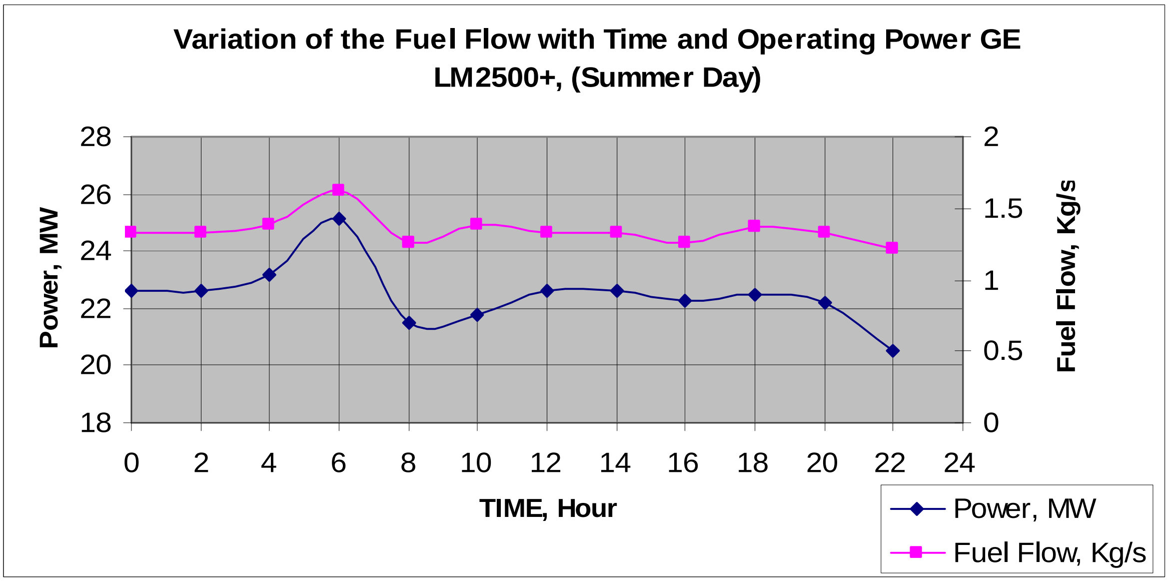
APPENDIX G

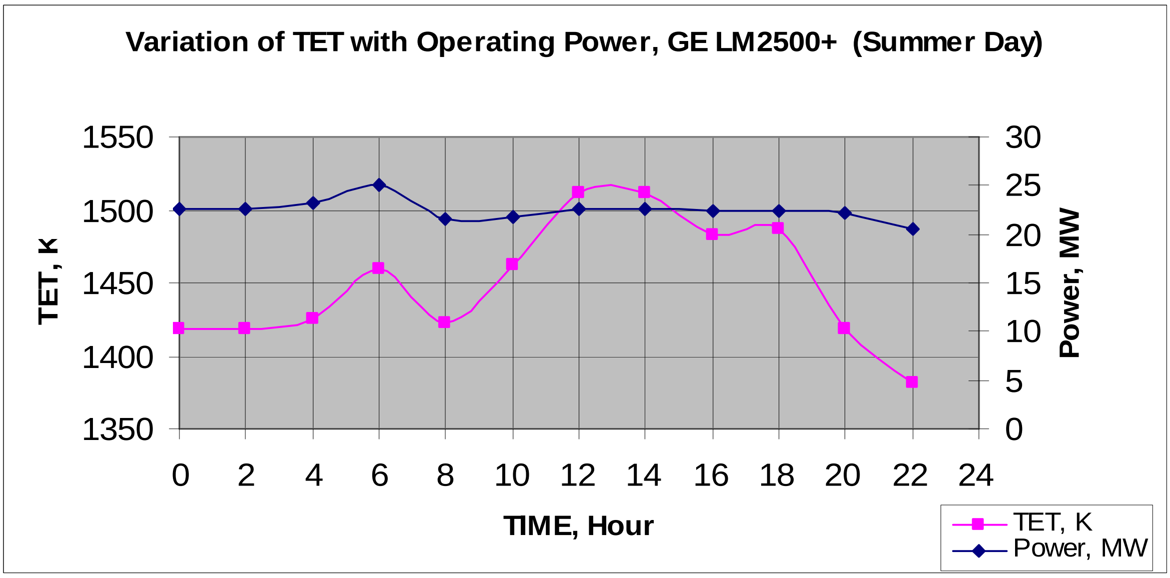


Variation of Fuel Flow with TET, GE LM2500+
Fuel Flow, kg/s
3
2.5
2
1.5
1
0.5
0
1150
1200
1250
1300
1350
1400
1450
1500
1550
1600
1650
1700
1750
TET, K
Tamb=15 C
Tamb=-25 C
Tamb=-15 C
Tamb=-5 C
Tamb=5 C
Tamb=25 C
Tamb=45 C
Tamb=55 C
Tamb=35C



Variation of TET with Power, GE LM2500+
Shaft Power, MW
50
45
40
35
30
25
20
15
10
5
0
1200
1250
1300
1350
1400
1450
1500
1550
1600
1650
1700
1750
TET, K
Tamb=15 C
Tamb=-25 C
Tamb=-15 C
Tamb=-5 C
Tamb=5 C
Tamb=25 C
Tamb=45 C
Tamb=55 C
Tamb=35C


Variation of the Fuel Flow with Time and Operating Power GE LM2500+, (Summer Day)
Power, MW
Fuel Flow, Kg/s
TIME, Hour
Power, MW
Fuel Flow, Kg/s
28
26
24
22
20
18
2
1.5
1
0.5
0
0
2
4
6
8
10
12
14
16
18
20
22
24


Variation of TET with Operating Power, GE LM2500+ (Summer Day)
TET, K
Power, MW
TIME, Hour
TET, K
Power, MW
1550
1500
1450
1400
1350
30
25
20
15
10
5
0
0
2
4
6
8
10
12
14
16
18
20
22
24

APPENDIX H

Code Link, Brief Description

This code link a genetic algorithm with the TURBOMATCH code. The genetic algorithm is the main driver and it is implemented in file ga170.f90.

It current requires a user-dependent objective function which is called "func(j, val)" and which was defined in the gnetic algorithm file ga170.f90

The objective function, has been developed, this will allow easy maintenance of the code, and allow the user to modify his objective function without modifying the main core of the GA program

The turbomatch code contains 10 files (below) + an extra link file "**link_ga_tm.f90**"

Add_handler.f90,

briska.f90,

briskb.f90,

ctmaps.f90,

inpass.f90,

master.f90

outer.f90 (contains most of key variables or data required by the GA algorithm)

tm_help.f90

varcom.f90

vartur.f90

The turbomatch code was originally a standalone application with main program file; this main file has been replaced by the file link_ga_tm.f90 which contains one subroutine called **"LINK_TURBOMATCH"** with the some input (enginenumber, tet) and some outputs depending on the user (currently there are 4 outputs:

SP, FF, ESFC, THEFF, but we can add more if require by the **gafortran** program, most of the key output variables can be found in file outer.f90)

To completely define the objective function used by the ga170 program, we generate an extra file **engine_cost.f90**, which contains the functional definitions of all the cost functions as following:

Revenue

Capital Cost

Fuel Cost

O&M Cost

Emission Tax

Financial Cost

If a new cost function is required, it should be defined in this file.

All the 16 files mentioned above have been grouped in the same workspace (directory last version), to make the link effortless, and easier to maintain. So in theory, if we consider that the turbomatch code & ga code always works, the user should only modify two files (**objective_function.f90**, and **engine_cost.f90**), and possibly the file"**link_ga_tm.f90"** only to export new output variables or data to the objective function.