بسم الله الرحمن الرحيم

Effects of Air Temperature on the Efficiency of Gas Turbines in Garri Power Plant

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Dedication

For all people I care for and I love

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List of symbol

P_{out}	Power out put
Q_{in}	Input Heat
Q_{out}	Output Heat
CV_{LDO}	Calorific Value of Light Diesel Oil
CV_{LPG}	Calorific Value of Liquefied petroleum
η_{th}	Thermal efficiency
m_{LDO}	Mass of light Diesel Oil
m_{LPG}	Mass of Liquid Petroleum Gas

List of Abbreviations

CHP	Combined Heat and Power
LHV	Low heat Value
PPM	Part per million
O&M	Operation and Maintenance
HRSG	Heat Recovery Steam Generator
GT	Gas Turbine
PSIG	Pound per Square Inch Gage
SCR	Selective Catalytic Reduction
HPE	Harbin Power Engineering
NEC	National Electrical Corporation
LDO	Light Diesel Oil
LDC	Local Dispatching Center
SCADA	Supervisory Control And Data Acquisition
CCR	Central Control Room
DCS	Distributed Control System
PLC	Programmable Logic Controller
OS	Operation Station
HVAC	Heating ventilation air conditioning
UPS	Uninterruptible Power Supplies
CRT	Control
FSR	Fuel Stroke Reference
PTW	Permit to Work
LPG	Liquefied Petroleum Gas
Psig	Per square inch gauge

KRC	Khartoum Refinery Company
C&I	Combustion and Inspection

Abstract

The objective of the present work is to investigate the affect of the a temperature on the performance of gas turbines in Garri Power Plant.

Several site visits were conducted to collect relevant data followed by calcu during period of twenty one months for ***Garri Power Plant*** so as to stu effects of ambient conditions on performance. Also the research was exter study the different types of fuels used during the same period in order to s the efficiencies were affected.

In this study, remedies to improve the efficiencies are suggested to overcom impacts caused by variable ambient conditions.

Various aspects of the above proposed improvements were explored an effects on ***Garri Power Plant*** efficiency, economics, reliability, maintai and lifetime were highlighted.

Other recent improvements of efficiency such as reducing inlet air tempera spraying water at air intake and using LPG as a fuel are mentioned as recommendations.

ملخص الدراسة

هذه الدراسة هو دراسة تأثير درجة حرارة الهواء الخارجية على كفاءة وآداء التربينات الغازية فى
ى.

حث علي عدد من الزيارات الميدانية لمحطة قري لجمع البيانات المتعلقة بحساب القدرة والكفاءة
للمعلومات الخاصة بالمحطة من حيث التشغيل والصيانة. تبع ذلك اجراء حسابات الكفاءة للمحطة.
البحث علي اثر درجة حرارة الهواء الداخل ونوع الوقود المستخدم علي الاداء.

بعض الحلول لمعالجة تدني الكفاءة بسب درجة حرارة الهواء.

حث عدة جوانب لدراسة المتغيرات المذكورة اعلاه علي محطة قري من حيث الكفاءة، الاقتصادية،
الصيانة، والعمر التشغيلي.

فاءة اوصي البحث بتقليل درجة حرارة الهواء الداخل عن طريق رشه بالماء كما اوصي باستخدام
ول المسال كوقود رئيسي.

Chapter 1

Introduction

1.1 Historical background

Engineering advancements pioneered the gas turbine in the early 1900s, and the turbines began to be used for stationary electric power generation in the late 1930s. Turbines revolutionized airplane propulsion in the 1940s and are currently the economic and environmentally preferred choice for new power generation plants in all over the world[1].

Gas turbines can be used in a variety of configurations :(1) simple cycle operation which is a single gas turbine power only, (2) combined heat and power (CHP) operation which is a simple cycle gas turbine with a heat recovery exchanger which recovers the heat in the turbine exhaust and converts it to useful thermal energy usually in the form of steam or hot water, and (3) combined cycle operation in which high pressure steam is generated from recovered exhaust heat and used to create additional power using a steam turbine. Some combined cycles extract steam at an intermediate pressure for use in industrial processes and are combined cycle CHP systems[1].

Gas turbines are available in sizes ranging from 500 kilowatts (kW) to 350 megawatt (MW)[11]. The most efficient commercial technology for central station power-only generation is the gas turbine-steam turbine combined-cycle plant, with efficiencies approaching 60 % Low Heat Value(LHV). One simple-cycle gas turbines for power-only generation are available with efficiencies approaching 40% (LHV). Gas turbines have long been used by utilities for peaking capacity. However, with changes in the power industry and advancements in the technology, the gas turbine is now being increasingly used for base-load power .

Gas turbines produce high-quality exhaust heat that can be used in the CHP configurations to reach overall system efficiencies (electricity and useful thermal energy) of 70 to 80 %. By the early 1980s, the efficiency and reliability of small gas turbines (1 to 40 MW) had progressed sufficiently to be an attractive choice for industrial and large institutional users for CHP applications.

Gas turbines are one of the cleanest means of generating electricity, with emissions of oxides of Nitrogen (NOx) from some large turbines in the single-digits part per million (ppm)range, either with catalytic exhaust cleanup or lean pre-mixed combustion. Because of their relatively high efficiency and reliance on natural gas as the primary fuel, gas turbines emit substantially less carbon dioxide (CO_2)per kilowatt-hour (kWh)generated than any other fossil technology in general commercial use[1].

A combustion gas turbine, like any other internal combustion engine, is a machine which converts the thermal energy of burning fuel into useful power which, in turn is converted into mechanical energy[2].

1.2 Objective

The main objective of this work is to study the factors contributing to gas turbine inefficiency in Garri Power Plant.

1.3 Methodology

Calculations for the output of the plant followed by analysis work for the whole factors affecting the efficiency of the Garri gas turbines are required.

Chapter 2

Gas Turbines in General

2.1 Application

The oil and gas industry commonly use gas turbines to drive pumps and compressors, process industries use them to drive compressors and other large mechanical equipment, and many industrial and institutional facilities use turbines to generate electricity for use on-site. When used to generate power on-site, gas turbines are often used in combined heat and power mode where energy in the turbine exhaust provides thermal energy to the facility.

There were an estimated 40,000 MW of gas turbines based CHP capacity operating in the United States in 2000 located at over 575 industrial and institutional facilities. Much of this capacity is concentrated in large combined-cycle CHP systems that maximize power production for sale to the grid. However, a significant number of single-cycle gas turbines based CHP systems are in operation at a variety of applications as shown in (Figure 3.1). Simple-cycle CHP applications are most prevalent in smaller installations, typically less than 40 MW[1].

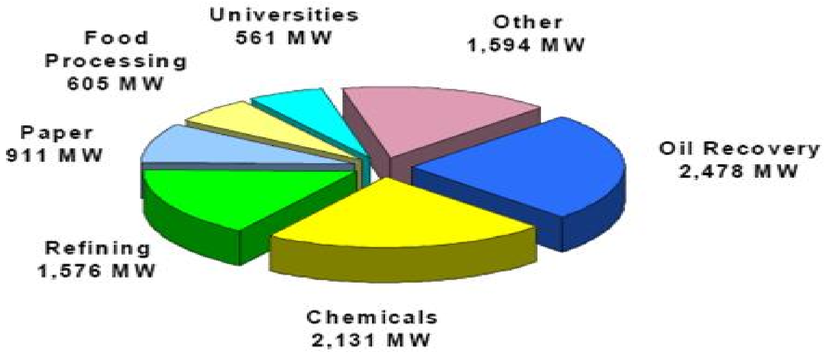


Figure 2.1 Existing Simple Cycle Gas Turbine CHP-9,854 MW at 359 sites (USA)

Source :PA Consulting; Energy Nexus Group

Gas turbines are ideally suited for CHP applications because their high-temperature exhaust can be used to generate process steam at conditions as high as 1,200 pounds per square inch gauge (psig) and 900 degree Fahrenheit (°F) or used directly in industrial process for heating or drying. A typical CHP application for gas turbines is a chemical plant with 25 MW simple cycle gas turbine supplying base-load power to the plant with an unfired heat recovery steam generator (HRSG) on the exhaust. Approximately 29 MW thermal (MWth) of steam is produced for process use within the plant.

A typical commercial/institutional CHP application for gas turbines is a college or university campus with a 5 MW simple-cycle gas turbine. Approximately, 8 MWth of 150 to 400 psig steam (or hot water) is produced in an unfired heat recovery steam generator and sent into a central thermal loop for campus space heating during winter months or to single-effect absorption chillers to provide cooling during summer.

While the recovery of thermal energy provides compelling economies for gas turbine CHP, smaller turbines supply prime power in certain applications. Large industrial facilities install simple-cycle gas turbines without heat recovery to provide peaking power in capacity constrained areas, and utilities often place gas turbines in the 5 to 40 MW size range at substations to provide incremental capacity and grid support. A number of turbine manufacturers and packagers offer mobile turbine generator units in this size range that can be used in one location during a period of peak demand and then trucked to another location for the following season[1].

2.2 Technology Description

2.2.1 Basic Process and Components

Gas turbines systems operate on the thermodynamic cycle know as the Brayton cycle .In a Brayton cycle, atmospheric air is compressed, heated and then expanded, with the excess of power produced by the expander (also called the turbine) over that consumed by the compressor used for power generation. The power produced by an expander turbine and consumed by a compressor is proportional to the absolute temperature of the gas passing through the device. Consequently it is advantageous to operate the expansion turbine at the highest practical temperature consistent with the economic materials and internal blade cooling technology and to operate the compressor with inlet air flow at as low a temperature as possible. As the technology advances permiting higher turbine inlet temperature, the optimum pressure ratios also increases.

Higher temperature and pressure ratios result in higher efficiency and specific power .Thus the general trend in gas turbine advancement has been towards a combination of higher temperatures and pressures. While such advancements increase the manufacturing cost of the machine, the higher value, in terms of greater power output and higher efficiency, provides net economic benefits. The industrial gas turbine is a balance between performance and cost that results in the most economic machine for both the user and manufacturer.

2.2.2 Modes of Operation

There are several variations of the Brayton cycle in use today. Fuel consumption may be decreased by preheating the compressed air with heat from the turbine exhaust using a recuperator or regenerator; the compressor work may be reduced and net power increased by using intercooling or precooling; and the exhaust may be used to raise steam in a boiler and to generate additional power in a combined cycle. (Figure 2.2) shows the primary components of a simple cycle gas turbine.

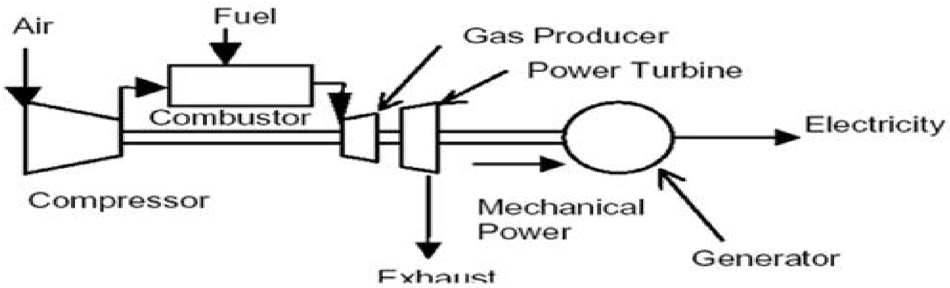


Figure2.2 Components of a Simple-Cycle Gas Turbine

Gas turbine exhaust is quite hot, up to 425°c to 480°c for smaller industrial turbines and up to 590°c for some new, large central station utility machines and aeroderivative turbines. Such high exhaust temperatures permit direct use of the exhaust. With the addition of a heat recovery steam generator, the Exhaust heat can produce steam or hot water. A portion or all of the steam generated by the HRSG may be used to generate additional electricity through a steam turbine in a combined cycle configuration.

A gas turbine based system is operating in combined heat and power mode when the waste heat generated by the turbine is applied in an end-use. For example, a simple-cycle gas turbine using the exhaust in a direct heating process is a CHP system, while a system that features all of the turbine exhaust feeding a HRSG and all of the steam output going to produce electricity in a combined-cycle steam turbine is not.

2.3 Types of Gas Turbines

Aeroderivative gas turbines for stationery power are adapted from their jet and turboshaft engine counterparts. While these turbines are lightweight and thermally efficient, they are usually more expensive than products designed and built exclusively for stationery applications. The largest aeroderivative generation turbines available are 40 to 50 MW in capacity. Many aeroderivative gas turbines for stationery use operate with compression ratios in the range of 30:1, requiring a high-pressure external fuel gas compressor. With advanced system developments, larger aeroderivative turbines (>40 MW) are approaching 45 % simple-cycle efficiencies (LHV) [1].

Industrial or frame gas turbines are exclusively for stationery power generation and are available in the 1 to 600 MW capacity range[13].

They are generally less expensive, more rugged, can operate longer between overhauls, and are more suited for continuous base-load operation with longer inspection and maintenance intervals than aeroderivative turbines. However, they are less efficient and much heavier. Industrial gas turbines generally have more modest compression ratios (up to 1:16) and often do not require an external fuel gas compressor. Larger industrial gas turbines (>100 MW) are approaching simple-cycle efficiencies of approximately 40 %(LHV) and combined-cycle efficiencies of 60 %(LHV).

Industry uses gas turbines between 500 kW to 40 MW for on-site power generation and as mechanical drivers. Small gas turbines also drive compressors on long distance natural gas pipelines. In the petroleum industry turbines drive gas compressors to maintain well pressures and enable refineries and petrochemical plants to operate at elevated pressures. In the steel industry, turbines dive air compressors used for blast furnaces. In process industries such as chemicals, refining and paper, and in large commercial institutional applications, turbines are used in combined heat and power mode generating both electricity and steam for use on-site[1].

2.4 Design Characteristics

Thermal output:	Gas turbines produce a high quality (high temperature) thermal output suitable for most combined heat and power applications .High-pressure steam can be generated or the exhaust can be used directly for process drying and heating.

Fuel flexibility: Gas turbines operate on natural gas, synthetic gas, landfill gas, and fuel oils. Plants typically operate on gaseous fuel with a stored liquid fuel for backup to obtain the less expensive interruptible rate for natural gas.

Reliability and Life: modern gas turbines have proven to be reliable power generators given proper maintenance. Time to overhaul is typically 25,000 to 50,000 hours.

Size range: Gas turbines are available in sizes from 500 kW to 600 MW.

Emissions: Many gas turbines burning gaseous fuels (mainly natural gas) feature lean premixed burners (also called dry low-NOx combustors) that produce NOx below 25 ppm, with laboratory data down to 9 ppm, and simultaneous low CO emissions in the 10 to 50 ppm range. 4 Selective catalytic reduction (SCR) or catalytic combustion further reduce NOx emissions. Many gas turbines sited in locales with stringent emissions regulations sue SCR after-treatment to achieve single-digit (below 9 ppm) NOx emissions.

Part-load operation: Because gas turbines reduce output by reducing combustion temperature, efficiency at part load can be used substantially below that of full-power efficiency[1].

2.5 Performance Characteristics

Electrical Efficiency

The thermal efficiency of the Brayton cycle is a function of pressure ratio, ambient air temperature, turbine inlet air temperature, the efficiency of the compressors and turbine elements, turbine blade cooling requirements, and any performance enhancements (i.e .recuperation, inter cooling, inlet air cooling, reheat, steam injection, or combined cycle). All of these parameters, along with gas turbine internal mechanical design features, have been improving with time. Therefore newer machines are usually more efficient than older ones of the same size and general type . The performance of a gas turbine is also appreciably influenced by the purpose for which it is intended. Emergency power units generally have lower efficiency and lower capital cost, while turbines intended for prime power, compressor stations and similar applications with high annual capacity factors have higher efficiency and higher capital costs. Emergency power units are permitted for a maximum number of hours per year and allowed to have considerably higher emissions than turbines permitted for continuous duty [1].

2.6 Advantages of Gas Turbine engines

1. Very high power-to-weight ratio, compared to reciprocating engines.
2. Smaller than most reciprocating engines of the same power rating.
3. Moves in one direction only, with far less vibration than a reciprocating engine.
4. Fewer moving parts than reciprocating engines.
5. Low operating pressures.
6. High operation speeds.
7. Low lubricating oil cost and consumption[12].
8. Gas turbine units have the advantage that the number of machines on line can be increased or decreased in a very short time.
9. The gas turbine is the most readily available power plant to meet the variation of load profile.
10. The installation costs for gas turbines per Megawatt are relatively low and the turbines can be operational in a very short time.
11. Gas turbines require less highly specialized operation and maintenance staff[6].

2.7 Disadvantages of Gas Turbine engines

1. Cost is much greater than for a similar-sized reciprocating engine (very high-performance, strong, heat-resistant materials needed).
2. Use more fuel when idling compared to reciprocating engines.
3. Slow response to changes in power settings[12].

Chapter 3

Literature Review

3.1 Literature review

Jahnig, Charles E [3] performed a study on gas turbine power system with fuel injection and combustion catalyst . The study showed the present invention involves a method and system for producing power in gas turbines wherein fuel is combusted directly in the gas turbine under substantially isothermal conditions. More particularly, the present invention achieves significantly higher fuel efficiency and other benefits in a gas turbine power system by firing fuel at multiple points or zones as the hot gas passes through an expansion turbine, so as to offset substantially all of the drop in temperature that would otherwise be associated with the expansion of the gas. Maximum pressure in the system will be increased compared to conventional systems, whereby more power will be provided from a given flow rate of gas, and without a large increase in the turbine diameter. In simplest terms, the new system consists of a "combustor-turbine "in which the gas expands over a pressure ratio of perhaps 2/1 to 20/1, while gas temperature is maintained roughly constant by burning fuel during the process of expansion, followed by a turbo-expansion step without addition or removal of heat to provide a cooled outlet gas . The latter gas then goes to a heat exchanger and it is at low enough temperature to permit using an exchanger of reasonable size and cost, using practical materials that are already available and have been used in such service .

The heat exchanger serves to cool the gas, while preheating an air or gas stream that has been pressurized. After preheating, the gas is next passed through a compression turbine where its pressure is raised to the required level . The gas temperature will also increase since no major amount of heat is removed during this compression step. The gas can then be further heated if desired, in a combustion zone before entering the aforementioned combustor-turbine .

Compared to other systems handling the same gas flow rate, the new system of the present invention is characterized by much higher efficiency, higher maximum operating pressure, higher fuel consumption and power output, without the need to increase the maximum operating temperature[3].

A primary object of the present invention is thus to provide a gas turbine system that can generate power at much higher efficiency without requiring new and difficult developments in technology and the need for higher maximum temperatures in the turbine. A further object is to increase the net power output for a given gas flow rate[3].

Tony Giampaolo [9] (MSME.PE,) performed a study on gas turbine problems. The study showed there are many detectable problems for gas turbines. They are Turbine blade failures account for 25.5% of gas turbine failures . Turbine blade oxidation, corrosion and erosion is normally a longtime process with material losses occurring slowly over a period of time, Common starting problems are "hot starts "and "hung starts ". Hot starts are so called because they produce excessive exhaust gas or turbine inlet temperatures. Sometimes hot starts are also associated with compressor surge. Hot starts are caused by too rich a fuel schedule and running problems associated with fuel controls consist of the inability to accelerate (or increase load), or accelerate too rapidly[9]. A hung start is often the result of insufficient power to the engine from the starter. In the event of a hung start, the engine should be shut down. If the engine fails to accelerate to the proper speed after ignition or does not accelerate to idle rpm, a hung start has occurred. A hung start may also be called a false start. A hung start may be caused by an insufficient starting power source or fuel control malfunction[16]. There are also numerous running problems (such as miscellaneous trips, aborts, etc). That are specific to the type of control method employed[9].

However, there is a trade-off with all of these methods of increasing efficiency. They all increase costs, and some reduce the available power output of the gas turbine. The final design choice will be the most appropriate compromise that balances cost, power and efficiency for each specific application .

These different approaches can be broken down into five main categories :

- increased inlet temperature for the turbine
- regeneration
- compressor inter cooling
- turbine reheat
- steam/water injection .

3.2.1 Increasing turbine inlet temperature

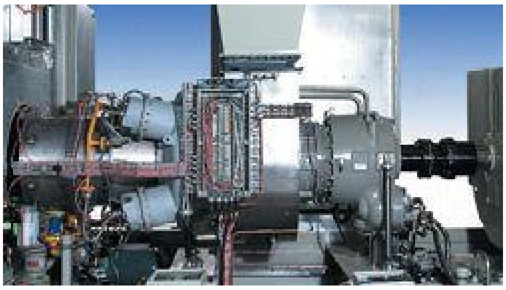


Figure 3.1: A simple-cycle gas turbine that generates 1.9 MW of shaft output power (OPRA Gas Turbines)

The most obvious way of increasing the efficiency of a gas turbine is to increase the inlet temperature of the turbine. Efficiency is related to both the inlet and outlet temperatures of the turbine; the higher the difference between the two temperatures, the greater the thermal efficiency of the turbine. There is an absolute limit to how low the outlet temperature can go, so increasing the inlet temperature is an obvious method of increasing efficiency[4].

The power of the gas turbine is directly proportional to the mass flow rate of the air passing through it, which is directly proportional to the air density. Since a high ambient temperature reduces the air-density, gas turbines designed to operate at standard conditions of 15.6^{o}C (60^{o}F) lose significant portions of their generating capacity when installed in hot climates. A high inlet-air temperature also increases the compressor work and lowers the thermal efficiency. Therefore, gas turbines operating under hot climates do not only produce less power than their design capacity, but also consume more fuel. Gas turbines produce 25-35 %less power in summer than in winter at 5-10% higher heat rate (i.e .an average increase of 6 % in fuel consumption). While in temperate climates this problem is only faced during the hot summer days, in Sudan the air temperature is relatively high all the year round[10].

However, increases in inlet temperature have already reached the point where the temperatures are actually higher than the melting point of some of the metals used in the turbine . Cooling of the first rows of turbine blades is therefore imperative, and any further increases in inlet temperature will require improvements in cooling techniques. This could involve increased use of steam cooling, increased flow of the cooling fluid, or increased effectiveness of heat transfer with the cooling fluid[4] .

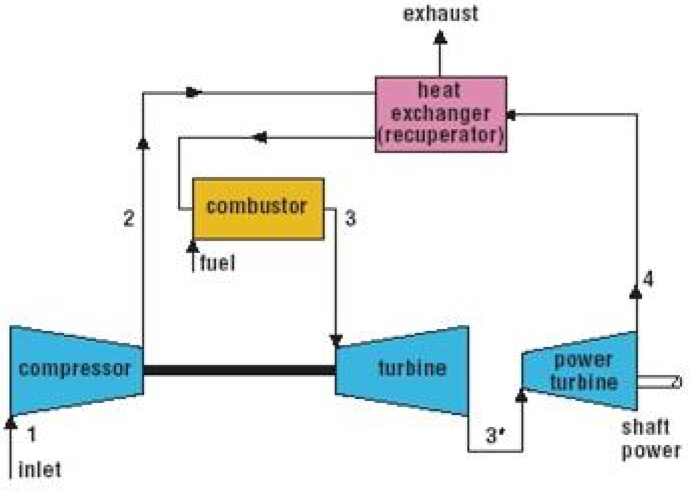
In addition, different materials with improved heat-resisting properties and improved thermal barrier coating can also assist in allowing the elevation of the inlet temperature[4] .

Nonetheless, the cost involved in increasing the efficiency of the turbine through raising the inlet temperature is becoming increasingly prohibitive. The materials involved are adding to the cost, while the addition of ever-more complex cooling techniques gives rise to more expensive production and more areas that need maintenance, which adds to O&M costs. It also introduces more items that can fail, potentially leading to greater outage time for the turbine .

The pressure ratio has to be increased alongside increases in turbine inlet temperature otherwise the turbine exit temperature will drop, resulting in a reduction in the combined cycle efficiency.

3.2.2 Regeneration

Regeneration is the internal exchange of heat within the cycle. In the gas turbine cycle, the gases leaving the turbine are at a relatively high temperature. This temperature is higher than the temperature at the compressor outlet. Therefore, a regenerator (a surface-type heat exchanger) is used to preheat the compressed gases by using heat from the exhaust gas. This reduces the amount of fuel required by the combustor. Regeneration involves the installation of a heat exchanger (recuperator) through which the turbine exhaust gases pass. The compressed air is then heated in the exhaust gas heat exchanger before the flow enters the combustor, preheating the gas before it enters the combustion chamber, thus reducing the amount of fuel required (see Figure 3.2).



Schematic of a gas turbine that employs regeneration (Energy Solution Center)

Figure 3.2: Regeneration

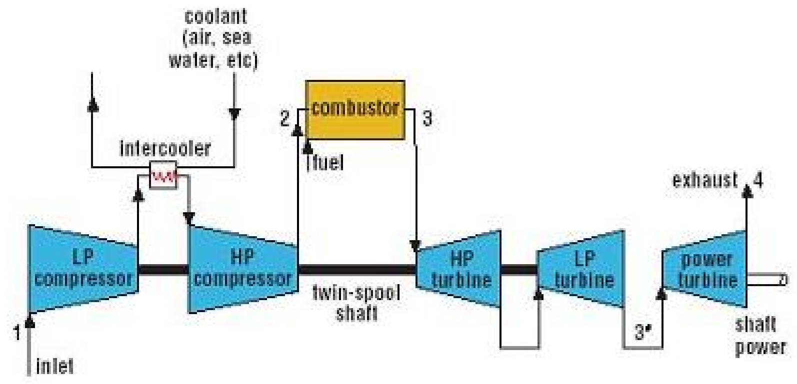
Use of a regenerator can increase the simple-cycle efficiency. However, the relatively high cost of such a regenerator is a disincentive to its use. Regeneration can improve the efficiency of simple gas turbines by 5% – 6 %. However, use of a regenerator reduces specific power output as a result of additional pressure losses in the regenerator[4] .

According to the Managing Director of OPRA Gas Turbines, Frederick Mowill, gas turbines have exit temperatures of about 560°C . This heat can be of great use in providing energy to produce process steam or heating. The use of a recuperator reduces this exit temperature to about 330°C . As a result, while the efficiency of the gas turbine is increased by the use of a recuperator, the available energy from the heat of the gas exhaust is reduced. Operators have to decide which is more important for their specific needs .

3.2.3 Compressor inter cooling

Another method of increasing the overall efficiency of a gas turbine is to decrease the work input to the compression process. The effect of this is to increase the network output. This can be achieved by cooling the gas passing through the compressor. Inter cooling involves compressing the fluid to an intermediate pressure, then passing it through a heat exchanger, or intercooler, where it is cooled to a lower temperature at essentially constant pressure. The fluid is then passed through another stage of the compressor, where its pressure is increased. This is followed by another intercooler process, and then another staging of the compressor, until the final pressure is achieved (see Figure 3.3). The overall result is a lowering of the network input required for a given pressure ratio.

Intercoolers can be air-cooled heat exchangers but are more commonly water-cooled. The output of a gas turbine is increased with an intercooler.



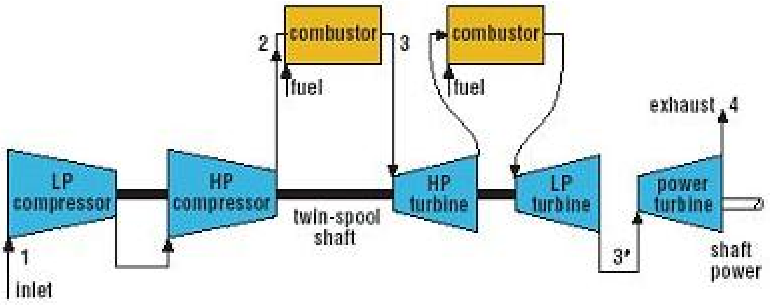
Schematic of a gas turbine that employs intercooling (Energy Solution Center)

Figure 3.3: Compressor inter cooling

3.2.4 Turbine reheat

Another method of increasing overall efficiency is to keep the gas temperature in the turbine as high as possible. This can be achieved by continuous heating of the gas as it expands through the turbine. Continuous heating as such is not practical, and reheat is carried out in stages (see Figure 3.4). Gases are allowed to partially expand before being returned to the combustion chamber, where heat is added at constant pressure until the limiting temperature is reached. The use of reheat increases the turbine work output without changing the compressor work or the maximum limiting temperature. Using the turbine reheat increases the whole cycle output. However, the final turbine exhaust temperature is above the outlet turbine temperature without reheat . As a consequence, reheating is most effective when used in conjunction with regeneration, as the quantity of heat exchanged in the regenerator can be greatly increased[4] .

If a gas turbine has a high-pressure and a low-pressure turbine at the back end of the machine, a reheater, usually another combustor, can be used to reheat the flow between the two turbines, giving an increase in efficiency of 1%–3%.



Schematic of a gas turbine that employs reheating
(Energy Solution Center)

Figure 3.4: Turbine reheat

3.2.5 Injection of steam or water

Steam or water injection is a method by which the output power of a gas turbine cycle can be increased. This has several effects : it increases the flexibility of the gas turbine during part load operations and significantly decreases emissions of carbon monoxide and unburned hydrocarbons .

Steam injection can be carried out using saturated or superheated steam. The introduced steam is usually injected into the combustion chamber.

In general, for every one percent of steam flow based on inlet air mass-flow the gas turbine’s output will increase with about 4%. This is because each kilogram of injected and heated steam contains more useful energy than a kilogram of air .The increase in power output can of course not be generated free of charge extra fuel supply is required to heat the injected steam to the turbine inlet temperature .Steam injection also can help if constant power is required for hot and cold days while the output power from a gas turbine varies with ambient conditions[8].

However, water injection is a more common method of increasing efficiency than steam injection. Water is typically injected into the system at the compressor outlet to increase the mass flow rate. The compressed air temperature falls as a consequence of water injection, but this temperature reduction can be minimized through use of a regenerator. No more fuel is consumed in this temperature compensation because the process uses heat from the gas turbine exhaust that would otherwise be wasted .

Water can also be injected at the compressor inlet. This has the following advantage compared with injection at the compressor outlet :

- Because air at the inlet air duct is at about atmospheric pressure, there is no need to use a high pressure pump .

- The inlet air temperature is atmospheric, which means there is no need to warm up the spray water to prevent thermal shock .
- There is usually a long distance between the inlet air duct and the compressor inlet, so by the time the atomized water reaches the compressor, it will be thoroughly mixed with the air, so it is a homogenized mixture of water and air that will be introduced into the compressor. This reduces impact damage and corrosion effects on compressor components[4] .

3.3 Additional methods of increasing efficiency

Several other methods of increasing efficiency are also employed .

3.3.1 Heat recovery

Gas turbines generate a large volume of very hot air. This exhaust is also high in oxygen content compared to other combustion exhaust streams because only a small amount of oxygen is required by the combustor relative to the total volume available. Depending on how much thermal energy is required, the turbine exhaust may be supplemented by a duct burner.

A duct burner is a direct-fired gas burner located in the turbine exhaust stream. It has a very high efficiency due to the high inlet air temperature and is used to boost the total available thermal energy. The turbine exhaust boosted by the duct burner is directed into the HRSG .

Turbine exhaust can also be ducted directly into hot air processes, such as kilns and material drying systems. This is the least costly first cost, as there is no boiler or steam drying system to purchase. Turbine exhaust can also be ducted directly into absorption chillers for large cooling loads .

The system will also include a diverter for those times when the waste heat is not required. The diverter vents the turbine exhaust to the atmosphere. This substantially reduces the system efficiency because only the electrical energy output of the turbine is being used .

The higher the electrical efficiency of the turbine the lower the available thermal energy in the exhaust. Newer turbines with recuperators and larger turbines tend to have higher efficiencies[4] .

3.3.2 Inlet air cooling

Another method of increasing turbine efficiency is through inlet air cooling. This is most effective in hot, dry climates because in effect it reduces the ambient temperature entering the turbine. Gas turbines operate with a constant volume of air, but the power generated depends on the mass flow of air. Warm air is less dense than cold air, resulting in lower power output. Warm air is also harder to compress than cold air, taking more work from the compressor and thus increasing internal losses .

Inlet air cooling is sensitive to ambient conditions at the site, and thus selection of the correct inlet air cooling system is site specific. For a typical gas turbine, it is possible to cool the inlet air from 15°C to 5.6°C, increasing fuel efficiency by about 2%.

There are many technologies that are commercially available for turbine air inlet cooling. These technologies can be divided into the following major categories :

- evaporative :wetted media, fogging and wet compression/overspray
- Chillers :mechanical and absorption chillers with or without thermal energy storage.
- LNG vaporization.
- Hybrid systems .

3.3.3 Reduction of leakage flow

One area of potential loss of efficiency lies in the leakage of flow through the gas turbine. This typically comes from inevitable leakage around the tips of the blades and vanes, as well as from other areas that require sealing .

Reduction of leakage is a compromise between reducing the leakage flow and allowing the leakage flow ‘necessary’ to avoid high temperature increases caused by disc friction or heat conduction or both. The most common method of optimizing blade design to achieve the best compromise is through the use of a blade tip shroud. These restrict gas leakage flow across the blade tip by using knife-edge seals designed to rub into a honeycomb seal material that is brazed onto the shroud blocks. However, centrifugal forces on the rotating stages of a gas turbine are very large. For example, an F-class blade weighing about 3.6 kg will exert a pull of over 440,000 N at operating speed. Typically, shrouds account for about 10 %of the weight of a blade.

Another method of reducing flow leakage at blade tips is through the use of abradable material. The obvious method of reducing this flow leakage is to reduce the clearance between the blade tip and the casing. Reducing this clearance can result in the blade tips rubbing against the casing. By applying an abradable coating, some rubbing can be tolerated. The abradable coatings are designed to release fine wear debris while causing no wear on the blades. A balance needs to be struck between efficiency losses due to gas flow leakage and those due to friction effects[4] .

3.3.4 Aerodynamic design of all components

The optimization of the aerodynamic design of all components can help reduce internal losses. However, there is an optimization process involved. For example, reducing pressure loss in the combustor will increase efficiency but may also result in a smaller stability range. It may also influence the possibility of having efficient cooling systems, especially in the first turbine stage .

3.3.5 Inlet air filtration

The operating conditions within a gas turbine can make particulate matter and chemical impurities in the air and fuel damage the blades of the turbine, either by collecting and sticking to the blades or by corroding and eroding them, which reduces their effectiveness and degrades the efficiency of the gas turbine. Consequently, the cleanliness of the air and fuel used has a major impact on the amount of maintenance required. There are two basic solutions to this : coating the turbine blades so that they can better resist this degradation; and preventing the impurities from entering the turbine in the first place .

Inlet air filtration will prevent a proportion of the impurities from entering the turbine. However, filtration also generally results in a pressure drop across the filtration system, resulting in a loss of efficiency. As a general rule, the more effective the filtration system is at removing particulate matter, the greater the pressure drop across the system and hence the greater the drop in efficiency .Designing a filtration system to maximize the first and minimize the second is a complex compromise. Precise selection will depend very heavily on site-specific operating conditions .

Particles that stick to blades and vanes cause blade fouling. These interfere with the smooth flow of the air stream, resulting in a reduction in the mass flow through the turbine. Fouling also degrades the effective pressure ratio, also impacting on efficiency .

Any parameter affecting the mass flow of the air entering the gas turbine will have an impact on the efficiency of the gas turbine[7].

Fouling has to be washed away by spraying a liquid from a set of nozzles installed upstream of the inlet. The liquid follows the air stream, and mechanical movements and chemical action by the washing liquid removes the deposits. Washing can be carried out either on-line or off-line. On-line washing is not as effective as off-line washing, but has the advantage that the turbine is still generating power. On-line washing has also been criticized for causing erosion damage .

It should be noted that inlet air filtration doesn’t increase efficiency; it limits and reduces the loss of efficiency from particulate degradation .

3.4 Areas of development

There are several areas of research being carried out with regard to aerodynamic improvements, particularly with relation to the use of unsteady effects. One major field is the potential of stator clocking. Numerical analyses have predicted an efficiency increase of about 0.5 % for an optimal clocking position. Experimental investigations in compressors and turbines have shown promising results, especially for axial turbines[4].

Another aspect of the unsteady flow phenomenon that is under investigation is the axial spacing between the rotor and the stator row blades. The potential increase in efficiency has been estimated at about 0.5%.

However, approaches based on periodic unsteady effects increase the risk of vibration excitation of the blades and vanes. Further investigation into these is needed before they can be used in industrial applications.

Another research activity is in 3D optimization. The vane and blade shapes combined with 3D side-wall contouring have been shown to have a great impact on secondary flow losses. Optimizing these can lead to a significant efficiency increase .

There is also work being carried out on new and improved cycles, such as ‘isothermal’ compression, involving cooling in the compressor, sequential combustion, bottoming cycles and semi-closed cycles[4].

One of the main factors complicating all this research is that none of the actions can be considered in isolation. Each effects the functioning of the gas turbine as a whole. For example, a change in the load distribution inside the turbine will influence the hot gas pressure, the exit pressure of the cooling system, the cooling air mass flow, the balance of mass flow inside the secondary air system, and the extracted mass flow from the compressor.

One also has to look at the efficiency of the gas turbine across a wide operating range and compare this with the likely operating regime that the turbine will be under. If the turbine is a base-load engine that will operate within fairly tight conditions, it can be optimized for those specific conditions. If, on the other hand, it has to operate under differing load conditions, then it may be better for the turbine to have a lower design efficiency that is more stable over the whole operating range[4].

3.5 Development of specific gas turbines

The size of industrial gas turbines has grown . They can generate up to 200 MW at 50 Hz . Turbine entry temperatures have risen to up to 1260°C, and pressure ratios have increased to up to 16:1 . Examples include :

- The ABB GT 13E2, rated at 164 MWe gross output on natural gas, has an efficiency of 35.7 %. The pressure ratio is 15:1. The combustion system is designed for low NOx production. The dry NOx-emission levels are less than 25 ppm with no water or steam injection on natural gas. The turbine entry temperature is 1100°C, and the exhaust temperature 525°C. The turbine has five stages, and the first two rotor stages and the first three stator stages are cooled. The roots of the last two stages are also cooled .

- Siemens' model V84.3, rated at 152 MWe, has an efficiency of 36.1 %. The turbine has a pressure ratio of 16:1. Each chamber has six burners designed for low NOx emissions. The turbine entry temperature is 1290°C and the exhaust temperature 550°C. The turbine has four stages, and the first three rotating stages are air cooled. The effectiveness of the cooling is improved by intercooling the cooling air after it is drawn out of the compressor[4].

GE and European Gas Turbines have jointly developed the MS9001F 50 Hz engine, which generates 215 MWe at an efficiency of 35 %. The engine uses an 18-stage compressor with an overall compression ratio of about 20:1. The gas turbine has three stages, the first two being cooled. Gas temperature at entry into the turbine is 1288°C.

In the long term, it is expected that the increasing costs for fossil fuels will allow techniques that are currently considered not to be cost effective to become economically viable. Most of these techniques will drive the gas turbine thermal cycle to more closely resemble the Carnot cycle. This introduces the possibility of using intercooling in the compressors and increasing the level of sequential combustion in the turbine[4].

The increasing cost of fossil fuel is also likely to result in towards increased use of alternative fuels, which will require greater fuel flexibility from turbines. This will in turn require turbines to be able to operate across a broader range of conditions, making it necessary to reconsider the balance of very high efficiencies at specific conditions with high efficiencies across a range of conditions[4] .

It is perhaps worth noting that a report from the EU states : Natural gas combined-cycle technology has been continuously improved over the last 20 years ...Further increases of efficiency depend on gas turbine development, such as increased turbine inlet temperature and blade cooling technologies. The water steam cycle in a combined cycle is almost at its physical limits, and only small improvements in efficiency are expected through the improvement of components such as pumps and steam turbine blades[4].

Chapter 4

Garri Power Plant

4.1 Plant Overview

4.1.1 General

The GARRI COMPLEX POWER STATION is located about 70 km north of Khartoum City, the capital of SUDAN.

The contract titled SUDAN GARRI POWER STATION -Plant 1 was generally executed by Harbin Power Engineering Co .Ltd (HPE) . The supervisor committee (consultant) was Lahmeyer International Germany.

The GARRI COMPLEX POWER STATION belongs to National Electrical Corporation (NEC) of SUDAN. The complex consist of four plants, two existing (plant 1and plant2), one is under construction (plant 4) and one future plant (plant 3) (see figure 4.1).

Figure 4.1: Layout for Garri Power Plant

Plant one consist of six units comprising of two sets of combined cycle units, including four gas turbines with type of PG6581B and rated capacity of 38MW, two units of steam turbine with rated capacity of 36MW. Total installed capacity is 224 MW.

The Heat Recovery Steam Generator (HRSG) is made by Harbin Boiler Works (China).

The steam turbine type L36-6.70 is also the product of Nanjing Turbine & Electrical Machinery Group Co .Ltd . Each has a capacity of 36MW equipped with brushless excitation and digital excitation regulator SVR-2000A (figure4.1).

The water used for all plant comes from NILE River, where a water pre-treatment station is installed to purify the dirty water by chemical dosing, and then water is delivered to the Site through a common 15 km pipeline. The waste water after treatment through a new waste water treatment plant is going back to NILE River with an acceptable quality.

For Light Diesel Oil (LDO) tanks have been installed and are receiving LDO by existing pipes from Khartoum Refinery Company (KRC) or unloading station.

There is a big electrical substation with a voltage degree of 220kV, and 6 bays as well as one coupler are ready to send electricity to grid.

There are two control centers, one located in substation where communication is available with Local Dispatching Center

(LDC) through Supervisory Control And Data Acquisition (SCADA) system and local control & protection systems are also put in. Another control room called Central Control Room (CCR) is on the 3rd floor of steam turbine hall.

The general control system Distributed Control System (DCS) is Freelance 2000 produced by ABB Company, it has an ability to communicate with other controllers, such as Programmable Logic Controller (PLC) servicing for River-side and water demineralization Station, as well as controller (MARK VI). So all the running parameters including Gas Turbine, Steam Turbine, LDO system, Demi-water plant & Rive-side can be shown on the 10 sets of Operation Station (OS).

For electrical arrangement, the power generated by gas turbine steps up through main transformer to the 220kV grid, same time steps down through a unit auxiliary transformer to 6.3kv medium voltage for house load system. The power generated by steam turbine is directly stepping up through main transformer to 220kVgrid.

An emergency diesel generator is also available here in this plant for any emergency case in order to ensure the essential load.

A newly-installed demi-water station is controlled by PLC Siemens Simapic S7-300 to generate enough capacity of demineralised water for HRSG, steam turbine and other customers.

Other main ancillary systems consist of air compressor system, firefighting system, potable water generation plant, waste water treatment plant, heating ventilation and air conditioning (HVAC) system, DC system, Uninterruptible Power Supplies system (UPS), etc.

4.1.2 Parameters of Main Equipments

4.1.2.1 No.4 Gas turbine generator unit

The gas turbine-generator unit was manufactured by Nanjin Turbine Group Company limited. The power output is 32551KW under the following design conditions:

Ambient temperature :40º C
Atmosphere pressure :0.966 bar
Ambient humidity :38%
Inlet air pressure drop :100 mmH$_2$O
Exhaust pressure drop
(under combined cycle) : 350 mmH$_2$O
Fuel : Distil (LDO)
Power factor : 0.80
Rated frequency : 50Hz

4.1.2.2 Heat recovery steam generator

The HRSG was supplied by Harbin marine boiler & turbine research institute.

It has two steam pressure levels one of which is used for deaerator and it has
No duct firing. The HRSG output parameters are:

Maximum continues output :63.78 t/h
Output steam pressure :6.9 MPa(g)
Output steam temperature : 468 (+5-/l0)º C
Exhaust gas temperature : <154º C
Feedwater temperature :104º C

4.1.2.3 Steam turbine generator unit

Steam turbine also was manufactured by Nanjin Turbine Company limited. It is a single case, condensing type turbine. The main parameters are as followings:

Main steam pressure :6.7 MPa
Main steam temperature : 465º C
Rated process steam flow :6 t/h
Process steam pressure :0.91Mpa
Process steam temperature :244.3º C
Rated main steam flow :127.56t/h
Exhaust steam pressure :0.0099Mpa
Generator power factor :0.80
Frequency :50Hz.

4.1.3 Gas Turbine

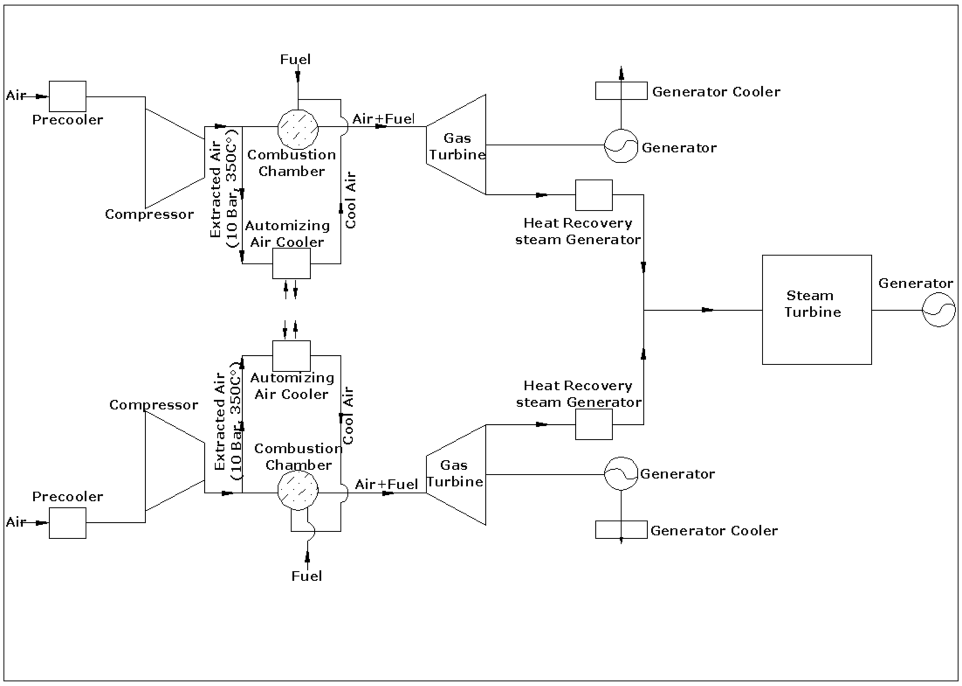


Figure 4.2: Diagram for Gas Turbines in Garri Power Plant 1

The gas turbines main body is composed of three major sections, i.e. control compartment, gas turbine compartment and generator.

The bases are equipped with weather protection enclosures for the necessities of outdoor installation and acoustic and thermal insulation. The generator is not equipped with a base plate and directly installed on the site foundation.

There is a load gearbox between the gas turbine and the generator to meet the drive necessity. The load gear is installed on a separated base plate. In addition the load gearbox and generator are equipped with enclosures in accordance with the requirement of outdoor installation [4].

Gas turbine and its accessories are installed in the gas turbine compartment with access door, ventilating openings, removable roofs to facilitate the inspection and maintenance of the equipment, as well as fire protection system. Gas turbine consists of a compressor of 17 stages, a turbine of 3 stages and 10 can type combustion chamber.

The forward side of the gas turbine compartment is the accessory compartment, of which the function is containing various accessories necessary for operating the gas turbine.

Among of them the main equipment includes lube oil system components, lube oil coolers, starting and ratchet system, accessory gearbox, fuel system, local gauge cabinet, hydraulic supply system, etc…

The load equipment includes a load gearbox enclosed circulating air cooling synchronous generator and related equipment.

In addition to the above-mentioned main body section, the unit also has:

- An inlet air system includes an inlet air filter house, an inlet air silencer and inlet duct
- An exhaust system includes an exhaust silencer, exhaust ducting and expansion joints for thermal compensation.
- An automatic fire protection system
- An compressor extraction air processing skid, a self-cleaning pressurized air source for the inlet air filters
- A backup air compressor equipment used as a backup self-cleaning pressurized air source for the inlet air filters

- A compressor water washing skid
- Two off-base blowers for cooling the turbine shell the turbine exhaust frame
- An oil mist eliminator system

The unit can be operated singly for generation power on base load or beak load, and also combined together with a heat recovery steam generator and steam turbine into a combined cycle plant.

Gas turbine section includes following main equipment and assemblies.

4.1.3.1 Intel Plenum

The air needed by gas turbine passes through the inlet air filter house, inlet silencer and transition duct and then enter the compressor inlet casing.

The inlet plenum there is a separate, by which the air flow is divided into two stream and then enter the compressor inlet casing without significant circumferential velocity.

4.1.3.2 Compressor Casing

It is composed of compressor inlet casing with No .1 bear housing, compressor casing and compressor discharge casing. The three casings are all large iron casings.

4.1.3.3 Turbine shell

It is another large casting of the gas turbine following the compressor casing. It is designed to make a contribution to support the alignment of all the moving and stationary parts of the turbine flow path section and ensure the axial and radial position dimensions of the parts and various clearances.

4.1.3.4 Turbine Exhaust Frame

It has a vertical flange, by which it is bolted to the vertical flange of the turbine shell . It consists of the out cylinder, the inner cylinder, the exhaust diffuser, the radial struts, etc…

4.1.3.5 Exhaust Plenum

It is located externally to the turbine exhaust frame, the hot gas flows into the exhaust plenum after passing the guide vanes in the exhaust frame and fills the complete space of the exhaust plenum, from which the exhaust gas is exhausted to the exhaust system.

4.1.3.6 Compressor

It can be classified as variable inlet guide vanes, compressor stator blades and exit guide vanes . The compressor rotor blades are mounted on the compressor rotor.

4.1.3.7 Turbine Blading and Turbine Flow Path Section

The turbine first stage nozzle with 16 segments receives the hot combustion gases from the combustion chamber via the transition pieces. The hot gases expand in the first-stage nozzle passage and then enter first-stage buckets.

After finishing the expansion the first-stage bucket section and doing work the hot gases enter the second-stage nozzle channels and continues to expand.

The hot gases which exit from the turbine second-stage buckets enter the third-stage nozzle channels (solid) and continues to expand.

4.1.3.8 Rotor Assembly

It is composed of the compressor rotor, the turbine rotor and the distance piece.

4.1.3.9 Bearings

Gas turbine has two journal bearings and one thrust bearing.

The No .1 bearing including the trust bearings is located at the journal of the forward stub shaft and is contained in the bearing housing inside the compressor inlet casing. The No.2 bearing in the contained in the bearing housing inside the exhaust frame inner tunnel. The No .1 and No.2 bearings have the function of supporting the rotor assemble of the gas turbine.

4.1.3.10 Combustion System

Combustion chamber is an assembly which provides the hot gas for the turbine. The highly pressurized air discharged from the compressor mixes with the fuel that is injected by the fuel nozzles to produce a combustible mixture in the combustion chamber.

Combustion system includes following components : combustion chambers, transition pieces, fuel nozzles, crossfire tubes, spring-positioning igniters, flame detectors, etc…

4.1.3.11 Gas Turbine Support

4.1.3.12 Accessory System of Gas Turbine

In order to meet the necessities of the unit for normal operation, control, monitoring, protection, cooling and lubricating, it equipped with following primary accessory systems:

- Accessory drive system
- Starting system and ratchet system
- Lube oil system
- Hydraulic supply system
- Trip oil system
- Fuel oil system
- Hazardous gas detection system
- Atomizing air system
- Cooling and sealing air system
- Cooling water system
- Fire protection system
- Ventilating system & lighting system
- Compressor washing system
- Compressor extraction air processing system
- Oil mist eliminator system

4.1.3.13 Generator Package

The QFR-38-2 type gas turbo-generator is manufactured in accordance with the technology introduced from the British BRUSH ELECTRICAL MACHINES LIMITED. It is a 3-phase synchronous generator which uses the enclosed water-to-air cooling mode.

The excitation of the generator is brushless excitation. It contains of a 3-phase AC main exciter which is coaxial with the generator and a single-phase AC permanent auxiliary exciter . The excitation current created by the main exciter is inputted to the excitation windings of the generator.

4.1.3.14 Load Gear

It is located between the gas turbine and the generator, its function is to transmit the torque of the turbine shaft to the generator shaft through speed reduction.

4.1.3.15 Control Compartment Package

Where has following devices:

- SPEEDTRONIC turbine control panel
- Generator control panel
- Generator protection panel
- Motor control center cabinet

4.1.3.16 Inlet and Exhaust Systems

It contains of the inlet filter house, the inlet silencer, inlet conduct, the elbow, etc… its function is to clean the air entering the compressor and reduce the noise level of the inlet air to acceptable value.

4.2 OPERATION OF THE UNIT

4.2.1 Responsibility of operational personnel

The operational personnel for the gas turbine must be conversant with the operation documents and other relevant information.

Consult the setting of the control system from the CONTROL SPECIFICATIONS .Look up the function of the individual accessory system and settings of the system components from

“DEVICE SUMMARY” and “PIPING SCHEMATIC DIAGRAMS”.

The operational personnel are also required to be aware of the power plant equipment either mechanically or electrically linked with the gas turbine and their effects on the normal operation of the unit.

Do not make an attempt to start the unit, either newly installed or major-overhauled, until the following conditions are met:

The requirements listed in “Checks Prior to Operation” have been met.

Perform the function inspection for the control system before starting the unit and confirm the capacity of proper operation.

Especially the operational personnel should cultivate good operation style. Here the following important points should be emphasized:

Response to annunciation indicators—Once the alarm information is displayed on the screen, investigate the cause and take correct actions particularly for the alarms from the protection systems, such as low lube oil pressure, over-temperature, high vibration level, over-speed, etc. These troubles must be corrected in time without any delay.

Check of control systems — After any type of control maintenance is finished, whether repair or replacement of parts, perform the function check of the control system for proper operation. This should be done prior to restarting the gas turbine. It must be strictly avoided that the reassembled “as taken apart” system is considered being capable of proper operation without performing the functional test.

Monitoring exhaust temperature during all phases of start-up—The operator is warned of the following:

Over-temperature can damage the turbine hot gas path parts .

It is necessary to monitor whether the exhaust temperature is under proper control upon first start-up and after any turbine maintenance is performed. If the exhaust temperature exceeds the normal trip level or rises at an unusual rate, trip the turbine. A particularly critical period for over-temperature damage to take place is during the start-up phase before the turbine reaches governing speed. This is because the air flow is low at this time and the turbine is unable to accelerate away from excess fuel.

4.2.2 Inspection and preparation prior to initial operation

The unit should be completely inspected before the initial operation either after finishing the installation of the unit or after finishing its major overhaul.

First the foreign matters and combustible substance around the unit should be cleared away. The unit must be inspected and remove away any foreign matters or tools dropped in any slots or gaps. Also inspect and clear the foreign matters in the inlet system.

Then the individual piping system must be inspected whether it is conformed to the relevant piping system schematic diagram and must ensure that the valves, sight ports, gauges and orifices are not missing and correctly installed. The various electrical connections must be checked that they are correctly against the relevant elementary circuit diagrams, also the wire leads should be securely bound up and the covers of the various junction boxes must be consistent with the requirements.

During inspection particular attention must be paid to certain special items, combined with the features of the various systems.

4.2.3 General operating precautions

4.2.3.1 Temperature limits

The CONTROL SPECIFICATIONS should be looked up to find out the actual exhaust temperature control settings, it is important for the operational personnel to define a “baseline value "of exhaust temperature spread with which to compare future data .This baseline data is established during steady state operation after each of the following conditions:

a .Initial startup of unit;

b .Before and after a planned shutdown;

c .Before and after planned maintenance.

An important point concerning the evaluation of exhaust temperature spreads is not necessarily the magnitude of the spread, but the variation in spread over a period of time. A developing problem can be found by daily accurately recording exhaust temperatures and making plots. If the exhaust temperature spread exceeds 80°F(44°c)or deviates 35°F(19°c)from the baseline value, the correction measures should be accordingly taken .

The permitted maximum values of the turbine wheel-space temperatures during unit operation is given below:

Location **Temperature**

First stage

TT-WS1FI-1&2 = 800°F(427°C)

TT-WS1AO-1 & 2 = 950°F(510°C)

Second stage

TT-WS2FO -1 & 2 = 950°F (510°C)

TT-WS2AO-1 & 2 = 950°F(510°C)

Third stage

TT-WS3FO-1 & 2 = 900°F(482°C)

TT-WS3AO-1 & 2 = 800°F(427°C)

When the average temperature in any wheel space exceeds the above-mentioned limit, it indicates the presence of some trouble in the unit. High wheel space temperature may be caused by any of the following faults:

a .Restriction in cooling air;

b .Wear of turbine seals;

c .Excessive distortion of the turbine stator;

d .Improper positioning of thermocouple;

e .Malfunctioning combustion system;

f .Leakage in external piping;

g .Excessive distortion of exhaust inner diffuser.

4.2.3.2 Pressure limits

The "DEVICE SUMMARY "should be consulted to find the actual pressure switch settings . The nominal value of lube oil pressure in the bearing feed header is 0.173 MPa . The turbine will trip when the lube oil pressure decreases to 0.055 MPa . The entrapped particulate substance within the lube oil filtering system will result in pressure variations between the above two values.

4.2.3.3 Vibration limits

The maximum overall vibration velocity of the gas turbine should never exceed 25.4mm/s in either the vertical or horizontal direction .

4.2.3.4 Fire protection system operating precautions

The fire protection system, when actuated, will cause several functions to take place in addition to actuating the extinguishing media discharge system. The turbine will trip, an audible annunciator will sound, and the alarm message will be displayed on the control (CRT) .

The fire protection system after each activation, must be replenished and reset in order to be able react to another fire. The fire protection system includes the initial discharge and extended discharge sections and the extended discharge section is used to release the extinguishing media over a prolonged period of time[14].

4.3 Maintenance

4.3.1 General

Maintenance costs and availability are two of the most important concerns to the equipment owner. A maintenance program that optimizes the owner's costs and maximizes equipment availability must be instituted. for a maintenance program to be effective, owners must develop a general understanding of the relationship between their operating plans and priorities for the plant, the skill level of operating and maintenance personnel, and the manufacturer's recommendations regarding the number and types of inspections, spare parts planning, and other major factors affecting component life and proper operation of the equipment.

4.3.2 Maintenance Principle

4.3.2.1 Maintenance Classification

Terminology and indexes are standardized for maintenance works so as to improve the efficiency, reliability, availability of performance.

Basically maintenance can be classified as:

- Corrective maintenance
- Preventive maintenance (including regular inspection, minor repair, major overhaul)
- Predictive maintenance (including problem assessment, etc)

a. Corrective Maintenance

Corrective maintenance concerns on the occasional defects found and transferred from operation department by means of defect cards, work order or fault report, etc.

All of defects can be evaluated and identified by shift engineer as following polarities according to the influence to the operation:

- Normal
- Worse
- Urgent

Normal :it presents that the defect can be cleaned by means of necessary isolations without shutdown and special opportunity.

Worse :it presents that the defect can be cleaned without shutdown, but has to wait for a special chance

Urgent :it presents that the defect possibly causes a shutdown or out of service .It should be taken into action immediately.

b. Preventive Maintenance

Preventive maintenance refers to those items which are stipulated or recommended in the equipment manual coming from manufactures or related regulation.

Definitely preventive maintenance includes:

- Regular inspection
- Scheduled Maintenance
    - Minor Repair
    - Major Overhaul

Preventive maintenance basically refers to those items which are stipulated or recommended in the equipment manual coming from manufactures or related regulation.

c. Predictive Maintenance

Reliability encompasses measures of the ability of generating units to perform their intended function

- Reliability is expressing that the preventive maintenance activities have been made in a correct way
- A low reliability will also indicate that there are improvements to be made on the maintenance side, or that the purchased equipment is a low quality.

4.3.2.2 Recommended Rules on Maintenance

For a purpose of excellent performance on maintenance, management in charge should specify following procedure necessarily needed we strongly recommend.

- Job Description

- Posts Arrangement & Boundary
- Communication
- Permit to Work
- Root Cause (Event)Analysis
- Problem Assessment
- Spare part
- Purchase
- Training

Permit to Work (PTW)

The aim of permit to work is for operators to prevent any potential disturb resulting from the man error of maintenance works.

Permit to work (PTW) is a paper on which following information should be described:

- What work
- When to begin and how long time needed
- Who is in charge and from which department of this work
- What isolation is required by maintenance
- Who shift engineer that provides confirmation and permission
- Who & when operator finishes the isolation
- Who confirms and cancels the PTW

Normally, plant management has to decide a name list on which dominated & well-experienced engineers coming from maintenance departments are authorized to apply for PTW, not all. It is taken to ensure a full and understanding isolation measures can be shown on the PTW for the safety reason.

Shift engineer must mainly be concerned with the following issues when receiving a PTW:

- Clearly understand this work?
- Time is suitable for this work?
- Any effect or hidden danger to normal performance?
- Isolation measures required by maintenance are enough for safety and performance?

Operator in charge must know about:

- How and the steps to complete isolation measures
- Any change on operation resulting from this PTW
- What lock-out or tag-out needed
- Fill in form after completion

Maintenance staffs on work should:

- Know well about the objective
- Know well about the isolation measures explained by engineer
- Bring paper (PTW)always in work
- Clean after work finishes
- Engineer should give a final check after work completion

4.3.3 General Description on Maintenance of Main Equipments

4.3.3.1 Maintenance on No.4 Gas Turbine

4.3.3.1.1 General

Maintenance of the gas turbine, as of any rotating power equipment, must include a planned program of periodic inspection, with accompanying repair and replacement of parts as necessary, to ensure the maximum availability and reliability of the unit.

The object of this Maintenance Section is threefold:

- To aid the user in becoming familiar with the unit by separating the inspections according to specific systems and, where appropriate, describing briefly the reason for the inspection and the action to be taken.
- To identify those components and parts that should be periodically examined between the initial start-up tests and the designated inspection.
- Inspection intervals herein are based on engineering judgment and experience gained with gas turbine units. The actual time interval established for any particular gas turbine should be based on the user's operating experience and on ambient conditions, such as humidity, dust, and corrosive atmosphere.

Prior to scheduled inspections or taking operating data, clean the compressor per the gas turbine compressor cleaning procedure in the Standard Practices, out of this maintenance instruction. Before and after any inspection a complete set of operating data including vibration readings should be taken and recorded for reference. A record of the inspections made and the maintenance work performed will be most valuable in helping to establish a good maintenance program for the gas turbine units. It is expected that the maintenance program, will start with minor work, and increase in magnitude over a period of time to a major overhaul, and then repeat the cycle. The performance of inspections can be optimized to reduce unit outage time and maintenance cost for a particular mode of operation, and still maintains maximum availability and reliability of the unit.

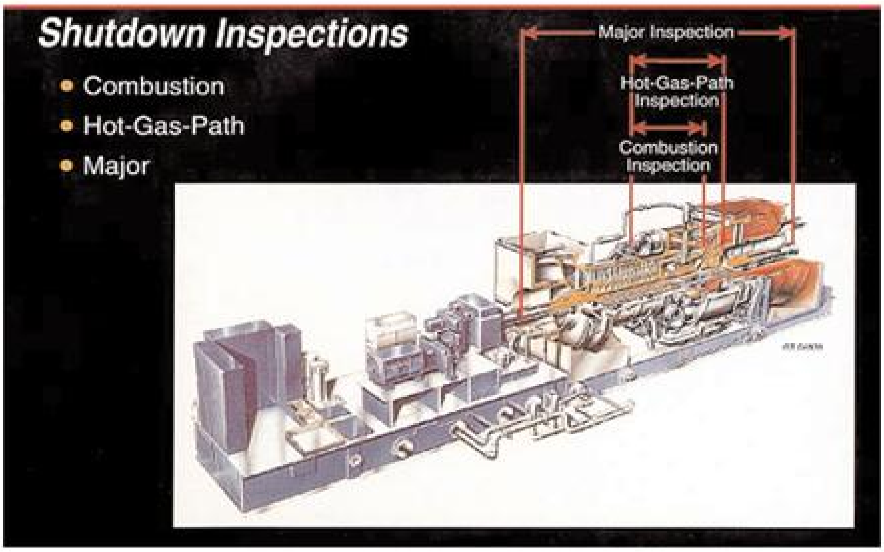
4.3.3.1.2 Estimated Maintenance Cycle of GT4 Maintenance

The types of inspections may be broadly classified in terms of unit "running” and unit "shutdown "inspections (figure 4.2).

The running inspection is performed during start-up and while the unit is operating. This inspection indicates the general condition of the gas turbine unit and its associated equipment.

The shutdown inspection is performed while the unit is at a standstill. The shutdown inspections include "Combustion Inspection", "Hot Gas Path Inspection "and "Major Overhaul”.

These latter inspections require disassembly of the turbine in varying degrees .



Shutdown Inspections

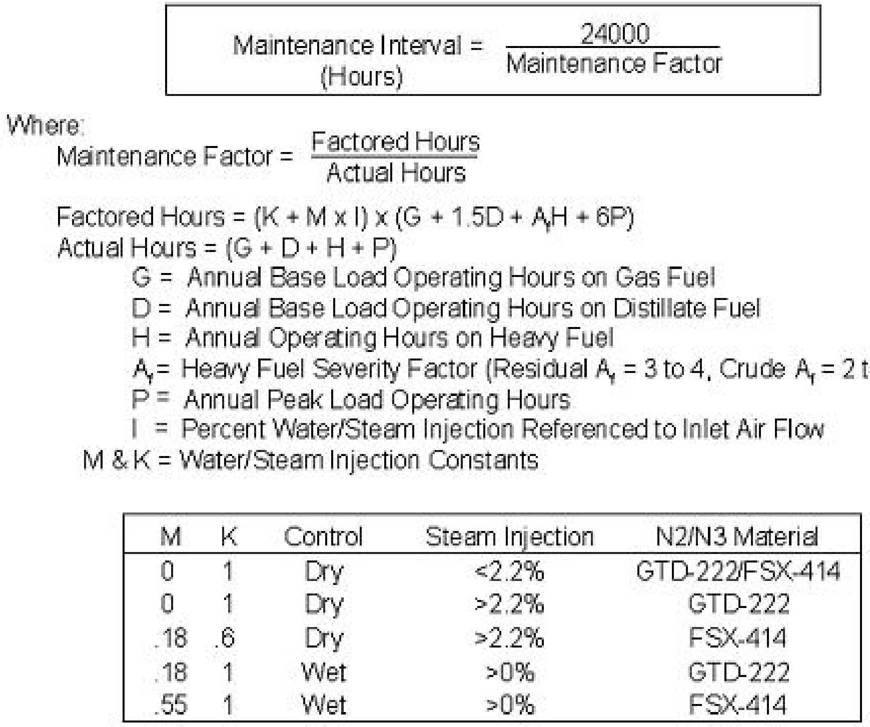
- Combustion
- Hot-Gas-Path
- Major

Figure 4.3: Gas turbine.Shutdown Inspection

Turbine starting reliability can be aided by conducting a "standby "inspection while the unit is shut down. Routine servicing of the battery system, changing of filters, checking oil and water levels, cleaning relays, checking device settings and calibrations, lubrication and other general preventative maintenance can be performed in off-peak hours without interrupting the availability of the turbine .Certain designated accessories in need of repair or replacement may be returned to the factory on either a repair and return basis or an exchange basis (figure 4.3).

General Electric Company Field Service Representatives are available to provide technical direction or consultation for repair and replacement.

Periodic test runs are also an essential part of a good maintenance program. It is highly recommended that the unit be operated at load for at least 1 hour bimonthly, and data recorded, it is recommended that gas turbines on extended shutdown)three weeks or more should be operated on turning gear or ratchet for one hour each day to prevent the buildup of corrosive deposits in the turbine wheel dovetail or the gas turbine should be operated at full speed, no load for one hour per week to dry the turbine out and thereby prevent moisture condensation in the turbine dovetail crevices. Special inspections such as bore scope and eddy current probe maintenance without interrupting availability. It is also recommended that visual inspections be performed whenever there is personnel at the unit[15].



$$\text{Maintenance Interval (Hours)} = \frac{24000}{\text{Maintenance Factor}}$$

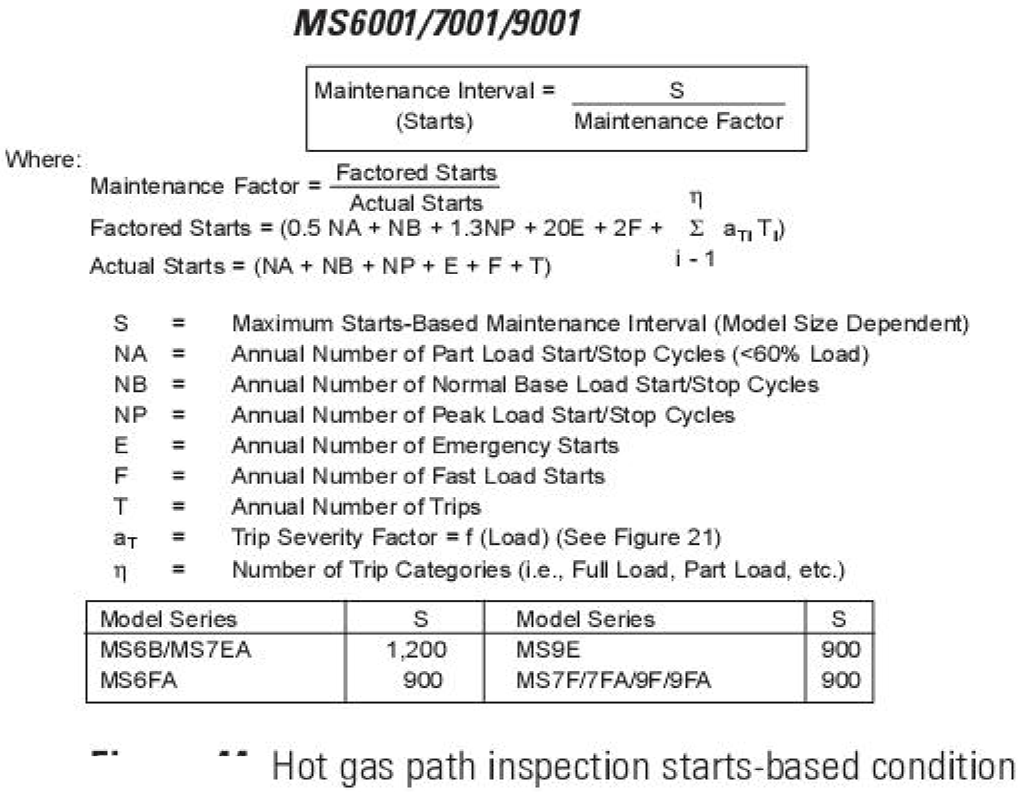
Where:

$$\text{Maintenance Factor} = \frac{\text{Factored Hours}}{\text{Actual Hours}}$$

Factored Hours = $(K + M \times I) \times (G + 1.5D + A_f H + 6P)$
Actual Hours = $(G + D + H + P)$
G = Annual Base Load Operating Hours on Gas Fuel
D = Annual Base Load Operating Hours on Distillate Fuel
H = Annual Operating Hours on Heavy Fuel
A_f = Heavy Fuel Severity Factor (Residual A_f = 3 to 4, Crude A_f = 2 to 3)
P = Annual Peak Load Operating Hours
I = Percent Water/Steam Injection Referenced to Inlet Air Flow
M & K = Water/Steam Injection Constants

M	K	Control	Steam Injection	N2/N3 Material
0	1	Dry	<2.2%	GTD-222/FSX-414
0	1	Dry	>2.2%	GTD-222
.18	.6	Dry	>2.2%	FSX-414
.18	1	Wet	>0%	GTD-222
.55	1	Wet	>0%	FSX-414

Hot gas path inspection: hours-based criterion



MS6001/7001/9001

$$\text{Maintenance Interval (Starts)} = \frac{S}{\text{Maintenance Factor}}$$

Where:

$$\text{Maintenance Factor} = \frac{\text{Factored Starts}}{\text{Actual Starts}}$$

$$\text{Factored Starts} = (0.5\,NA + NB + 1.3NP + 20E + 2F + \sum_{i-1}^{\eta} a_{Ti} T_i)$$

$$\text{Actual Starts} = (NA + NB + NP + E + F + T)$$

S = Maximum Starts-Based Maintenance Interval (Model Size Dependent)
NA = Annual Number of Part Load Start/Stop Cycles (<60% Load)
NB = Annual Number of Normal Base Load Start/Stop Cycles
NP = Annual Number of Peak Load Start/Stop Cycles
E = Annual Number of Emergency Starts
F = Annual Number of Fast Load Starts
T = Annual Number of Trips
a_T = Trip Severity Factor = f (Load) (See Figure 21)
η = Number of Trip Categories (i.e., Full Load, Part Load, etc.)

Model Series	S	Model Series	S
MS6B/MS7EA	1,200	MS9E	900
MS6FA	900	MS7F/7FA/9F/9FA	900

Hot gas path inspection starts-based condition

4.3.3.1.3 Effects to the Maintenance Cycle

The effect of maintenance factors for fuel, starts and load duty are cumulative if all the above factors are present. It should also be understood that as the maintenance factor increases the time between inspections and components repairs decreases and it is possible that component replacement frequency will increase.

a. Fuel

The effect of the type of fuel on parts life is associated with the radiant energy in the combustion process and the ability to atomize the various liquid fuels .Therefore, natural gas, which does not require atomization, has the lowest level of gradient energy and will produce the longest life of parts.

Natural gas has been the traditional fuel for use with gas turbines in industrial applications. Limitations on the available supply of natural gas, with the resultant increase in costs, have led to the consideration of liquid fuels to a greater degree than at any time in the past.

Of the liquid fuels, distillate fuel will produce the next highest life, and crude oil and residual oils, with the attendant higher radiant energy and more difficult atomization, will produce shorter parts life.

Contaminants in the fuel also affect maintenance intervals. This is particularly true for liquid fuels in which dirt results in accelerated replacement of pumps, metering elements, and fuel nozzles. Contaminants in fuel gas can erode or corrode control valves and fuel nozzles.

The limiting item to continuous operation on liquid fuels is the fuel nozzles .

Exceptionally "clean” fuel can increase this interval, while "dirty” fuel will decrease it accordingly.

b. Starting Frequency

Each stop and start of a gas turbine subjects the hot gas path to significant thermal cycles. Control systems are designed and adjusted to minimize this effect.

However, a gas turbine with frequent starting and stopping requirements will demonstrate parts lives that are shorter than those for a similar unit in continuous duty service.

c. Load cycle

The load cycle of the gas turbine, up to it's continues rating, will have little effect on parts lives, provided it does not require frequent and rapid load changes.

d. Environment

The condition of the inlet air to the gas turbine can have a significant effect on maintenance costs and intervals if it is either abrasive or corrosive. If abrasives are in the inlet air (e.g., as from sand storms), more attention should be paid to inlet filters in order to minimize this effect.

If the gas turbine is to be operated in a corrosive atmosphere (for example, one with salts), careful attention should be paid to the location of the inlet air arrangement and the application of correct materials and protective coatings. It is essential during the planning stages of an application to recognize any abrasive or corrosive contaminants and to take the necessary steps to minimize them.

e. Maintenance Process

Parts condition information is based on estimates only, and will vary with machines and specific operating conditions. However, estimates are based on previous experience and can be very useful in planning a maintenance program. As actual operating data is accumulated on a specific application, adjustments of inspection cycles should be the next step in a well-planned program.

Initial inspection planning can be based on the combustion inspection schedule.

It must be recognized that the foregoing estimated outage requirements can be used for estimating maintenance cycles, however, these numbers will vary depending upon the many factors which establish the operating conditions for a specific installation. The inspection cycles will vary depending upon fuel, duty cycle and maintenance philosophy of the owner. The inspection man-hours will vary depending upon preplanning, availability of parts, productivity, weather conditions, union regulations, supervision, etc.

Precise estimates of the outage duration resource requirements and costs associated with the inspection of a specific installation may be obtained from your General Electric Company Apparatus and Engineering Services Operation Representative.

Good maintenance planning for minimum down-time requires the availability of replacement parts, either new or previously repaired, that can be exchanged with existing parts. The exchanged parts can then be repaired without extending shut -down time.

To ensure optimum performance of the gas turbine, the minimum stock of spare parts should be able to support the service inspection. A predetermined central location can stock spare parts that are adequate for hot gas path inspection. Many gas turbine plants stock capital spare parts on-site, recognizing that this parts availability minimizes the turn-around time required for major overhauls.

The planned maintenance program anticipates the needs of the equipment and is tailored to meet the requirements of the system for utilization, reliability, and cost.

4.3.3.1.4 Combustion Inspection

A brief shutdown inspection is required to change out fuel nozzles and to check the combustion liners transition pieces and crossfire tubes. These parts require the most frequent attention, as continued operation with a deteriorated combustion system can result in much shortened life of the downstream parts, such as turbine nozzles and buckets. It is also inherent in the gas turbine design that these parts are the first to require repair or replacement. Therefore, the importance of this inspection in the maintenance program must be emphasized.

A visual inspection of the leading edge of the first-stage turbine nozzle partitions and buckets should be made during the combustion inspection to note any wear or deterioration of these parts. This inspection will help to establish the schedule for the Hot Gas Path inspection.

The combustion liners, transition pieces, crossfire tubes, and fuel nozzles should be removed and replaced with new or repaired liners, transition pieces, crossfire tubes and new or cleaned fuel nozzles. This procedure reduces downtime to a minimum and the removed liners, transition pieces, crossfire tubes, and fuel nozzles can be cleaned, inspected and repaired later when it is more convenient.

After the combustion inspection is completed and the turbine has been returned to service, the removed liners, and transition pieces can be bench inspected and repaired if necessary, by competent service personnel, or off-site at a qualified service facility. Off-site cleaning inspection and repair of the liners and transition pieces is recommended, since this activity can best be performed where specialized equipment and fixtures are available .

The removed fuel nozzles can be cleaned site. Liquid fuel nozzles should be stored in sets, all by the same manufacturer, for use at the next inspection

4.3.3.1.5 Hot Gas Path Inspection

The Hot Gas Path inspection includes the Combustion Inspection just described and, in addition, a detailed inspection of the turbine nozzles and turbine buckets. To perform this inspection, the top half of the turbine case (shell), and the first-stage nozzle must be removed. The second-stage nozzle, the third--stage nozzle, and the turbine buckets will be inspected visually while still in place in the unit. A complete set of turbine clearances should also be taken during any inspection of the hot gas path.

As with the combustion inspection, it is recommended that replacement combustion liners, fuel nozzles and transition piece be available for installation at the conclusion of the visual inspection. The removed parts can then be inspected at a qualified service facility and returned to stock for use during the next inspection. It is also recommended that the Rot Gas Path inspection be conducted under the technical direction of the General Electric Company Field Service Representative for accurate analysis of inspection data and most effective use of outage time.

4.3.3.1.6 Major Overhaul

The major overhaul involves inspection of all the major “flange-to flange” components of the gas turbine which are subject to wear during normal turbine operation. This inspection includes elements of the Combustion and Hot Gas Path inspections. In addition, casings are inspected for cracks and erosion, rotor and stator blades are to be checked for tip clearance, rubs, bowing, cracking, and warpages. Shrouds are checked for clearance, erosion, rubbing and build-up. Seals and hook fits of nozzles and diaphragms are inspected for rubs, erosion, fretting or thermal deterioration. The compressor and inlet are inspected for fouling, erosion, corrosion, and leakage .

Bearings and seals are inspected for clearance and wear. All clearances are checked against their original values.

4.3.3.2 Maintenance on combustion and inspection (C&I) Instruments

4.3.3.2.1 The overhaul of C&I instruments is carried out along with the relevant unit .The items are listed as following:

a) Check over all the instruments and device, record the status in log book
b) Backup the configuration and program.
c) Verify the symbol and address.
d) Clean the power supply and modules.
e) Clean the dustproof filter.
f) Inspect and tighten the terminal connection and screws.
g) Inspect the earthed system.
h) Dust prevention and sealing of the control panel.
i) Replace the cooling fan.

j) Performance test of power supply.
k) Isolation test of loops.
l) Recorder, printer, alarm and HMI maintenance.
m) Eliminate the pending defects in the operation.
n) Recover and optimize the symbol and tag.
o) Hardware function performance test.
p) Configuration software download and upload inspection.
q) Measuring modules calibration.
r) Interlock protection test.
s) Cable, pipeline and accessories inspection and replacement.

4.3.3.2.1 Minor Overhaul

The minor overhaul of C&I instruments is carried out along with the relevant Unit. The items are listed as following:

a) Clean the dustproof filter
b) Check the cooling fans
c) Inspect the power supply and modules
d) Hardware function test
e) Eliminate the pending defects in the operation.
f) Partial repair of the local instrument
g) Partial calibration of the I/O modules
h) Communication inspection

4.3.3.3 Maintenance on Electrical Equipments

4.3.3.3.1 General

Electrical equipments are very important components in the power plant. Their service condition (technical status) is deemed significant to guarantee the safe and economical operation of the power plant. discovered and eliminated in time, and potential accident can be prevented as well. As all these have been done, the safety and operation skill of the repairmen must be enhanced during examine and repair.

Normally maintenance of electrical equipments is carried out at the same time with mechanical scheduled program because electrical equipments can be stopped at that moment.

4.3.3.3.2 Classification of Electrical Maintenance

- Corrective Maintenance : Repairing during the operation, deal with casual failure at any moment to keep safe operation of equipment and reduce loss .
- Preventive maintenance : Local and preventive inspection & checks on electrical equipment according to the program
- Overhaul : Checks and tests of electrical equipments according to the plan

4.3.3.3.3 Cycle of Electrical Maintenance

Normally maintenance of electrical equipments is carried out at the same time with mechanical scheduled program because electrical equipments can be stopped at that moment.

Maintenance of electrical equipment should follow the maintenance instruction coming from the manufacture and related standards stipulated by IEC or others.

Check sheets must be established as a way of preventive maintenances and carried out timely by maintenance staff so as to find any defect or abnormal thing earlier, therefore to avoid any disturbance to performance.

Cleaning, functional checks & necessary tests on electrical equipments are usually adopted for yearly maintenance, called as minor repair. Records should be carefully stored and compared with original data for any difference, from which defects, such as loosen wires, can be seen and found to be a method of preventive maintenance.

Any defect found through maintenance, quick action should be taken as:

- Analyze the reason, report & make a final solution
- It is highly recommended to be changed by new one if spare part available
- If no available spare parts in store, make ordering immediately

4.3.3.3.4 Minor Repair

One year is normally selected for electrical and to be with mechanical maintenance together, there gives a list of main electrical equipments on scheduled maintenance cycle in power plant according to IEC standards & China Standards.

4.4 Operation Problems

1. Cooling System:

Usage of raw water in heat exchangers for cooling caused fouling and blockage of water tubes (see figure 4.4). This caused the gas turbine to trip due to oil and atomizing air high temperature.

This problem was solved by the usage of dematerialized water in the cooling system.

Figure 4.4: fouling of generator cooling system

2. Fuel Potassium Content:

Fuel Potassium Content is affecting the turbine hot parts. And this problem can be solved by fuel purification (mixing the fuel with water and the impurities will settle down with the water droplets and leave the fuel pure) (figure4.5).

Figure4.5: Corrosion due to fuel potassium content effect on blades

3. Impurities:

Fuel impurities and dust from intake may cause erosion for the turbine internal parts as in figure4.6.

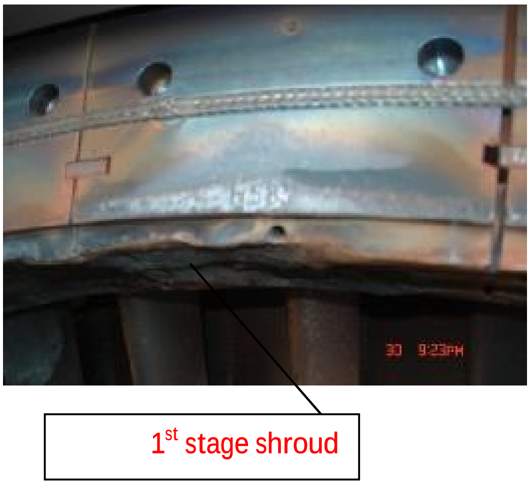


Figure4.6: Erosion due to impurities

4. Hot gas path parts

There is heavy erosion especially in the sealing ring which connected the transition piece. Hot corrosion in the leading edge root and slight wear in the trailing edge (figure4.7).

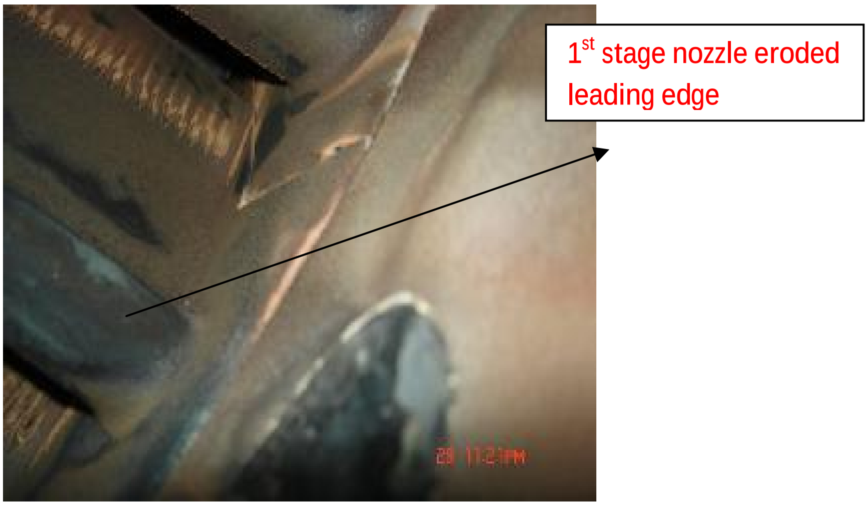


Figure4.7: The hot gas path parts

5. Inlet air filtration system:

Usage of old filter type (course mesh) caused dust to enter to the turbine internal parts and silicon to settle on the turbine internal parts (figure4.8).

This problem was solved by the usage of fine air filters (F9 instead of F7).

Figure4.8: The inlet air filters

Chapter 5

Analysis and discussion

1. In general, efficiency is influenced by :

- Energy used by the air compressor – if less energy is used to compress the air, more energy is available at the output shaft.
- Temperature of the gas leaving the combustors and entering the turbine – the higher the temperature, the greater the efficiency.
- Temperature of the exhaust gases from the turbine – the lower the temperature, the greater the efficiency.
- Mass flow through the gas turbine – in general, higher mass flows result in higher efficiencies.
- Pressure drop across inlet air filters – increased pressure loss decreases efficiency.
- Pressure drop across exhaust gas silencers, ducts and stack – increased pressure loss decreases efficiency .

There has been considerable work done to improve the efficiency of gas turbines, mainly on increasing turbine entry gas temperatures and increasing the efficiency and capability of the compressor . Various methods have been used to improve efficiency in these areas .

These include :

- Using the exhaust gas to heat the air from the compressor – this is most effective in cold weather.

- Dividing the compressor into two stages and cooling the air between the two stages.
- Dividing the turbine into two stages and reheating the gas between the two stages.
- Cooling the inlet air – this is mainly used in hot weather.
- Reducing the humidity of the inlet air.
- Increasing the pressure of the air at the discharge of the air compressor.
- Regularly washing or otherwise cleaning the fouling of turbine and air-compressor blades .

In this chapter; all the findings obtained from Garri power station were analyzed. The analysis showed results which have been plotted in graphs . The graphs provide and depict results of the calculated power output and efficiency. The calculations took into account the average reading of nine days for each month.

2. The effect of ambient temperature on efficiency

Example of calculations

Using the readings of the temperatures and generator output power [5] for the years 2006 and 2007, efficiencies of the plant were calculated as shown in table (5.3) and table (5.4) with the aid of related equations and using the data in (table5.1) and (table5.2)

Table 5.1: Garri Power Plant Performance Data, year 2006

Date	m_f(Tons) LDO	m_f(Tons) LPG	P_{out} MWh
1/3/2006	943.9	145.74	4436.245
2/3/2006	937.18	147.54	4656.018
3/3/2006	3823.23	149.06	3832.745
14/3/2006	914.89	178.05	4982.527
15/3/2006	845.10	201.27	4819.418
16/3/2006	949.01	201.83	5234.473
29/3/2006	991.87	210.92	5294.582
30/3/2006	1053.03	179.04	5188.291
31/3/2006	691.53	42697.2	4180.164
1/4/2006	954.61	200.80	5237.564
2/4/2006	854.09	201.38	5236.645
3/4/2006	1055.44	205.72	5197.418
14/4/2006	966.8	79.88	4686.355
15/4/2006	1160.12	0	4998.482
16/4/2006	1199.57	0	5146.636
28/4/2006	1087.45	0	4694.436
29/4/2006	1178.14	0	4931.582
30/4/2006	1169.76	0	4924.864
1/5/2006	1043.02	111.69	4879.036
2/5/2006	1011.09	180.95	5041.464
3/5/2006	809.97	150	4044.700
14/5/2006	1162.26	195.23	5448.300
15/5/2006	1024.86	190.68	5045.200
16/5/2006	971.07	192.16	4891.873
29/5/2006	886.39	188.28	4335.377
30/5/2006	877.2	177.47	4481.036
31/5/2006	878.47	184.22	4063.145
1/6/2006	717.57	173.97	3779.791
2/6/2006	518.49	160.97	3186.473
3/6/2006	761.5	176.66	3792.100
14/6/2006	826.49	177.8	4488.445

15/6/2006	879.84	173.27	4550.564
16/6/2006	521.84	173.27	3301.918
28/6/2006	1014.16	191.18	5031.418
29/6/2006	1017.42	191.34	5080.109
30/6/2006	694.59	180.51	3705.418
1/7/2006	899.85	178.97	4614.48
2/7/2006	1072.9	179.79	5217.345
3/7/2006	936.95	189.61	4831.664
14/7/2006	834.21	192.52	4578.355
15/7/2006	979.10	197.79	5133.736
16/7/2006	955.11	182.48	4959.827
29/7/2006	1194.25	0	4849.618
30/7/2006	1227.37	0	5104.636
31/7/2006	955.11	182.51	5129.418
1/8/2006	1239.6	0	5352.745
2/8/2006	1268.72	0	5482.673
3/8/2006	1184.02	0	5125.864
15/8/2006	857.31	23.62	3476.418
16/8/2006	860.36	203.87	4271.709
29/8/2006	1080.38	180.05	5248.355
30/8/2006	1071.44	186.6	5425.118
1/9/2006	695.99	177.99	4212.127
2/9/2006	961.75	179.27	5025.373
3/9/2006	1032.55	179.01	5130.491
14/9/2006	1208.41	0	4986.455
15/9/2006	1075.15	0	4314.545
16/9/2006	1167.03	92.5	5021.400
28/9/2006	957.61	133.53	4914.255
29/9/2006	986.23	0	4477.673
30/9/2006	1107.44	0	4821.982
1/10/2006	1096.66	0	4819.845
2/10/2006	1052.73	0	4577.382
3/10/2006	1115.83	0	4887.528
14/10/2006	1093.27	0	4726.545

15/10/2006	1103.66	0	4862.200
16/10/2006	1068.22	0	4700.527
29/10/2006	933.34	0	3929.245
30/10/2006	764.29	59.45	3724.918
31/10/2006	629.69	203.82	3796.273
1/11/2006	520.4	219.7	3407.136
2/11/2006	543.01	382.51	4375.282
3/11/2006	433.01	339.69	3818.773
14/11/2006	396.02	326.91	3702.509
15/11/2006	434.63	189.83	2691.355
16/11/2006	360.14	187.41	2461.891
28/11/2006	316.93	154.05	2119.109
29/11/2006	290.92	147.31	1931.800
30/11/2006	264.61	146.41	1858.500
1/12/2006	159.01	0	419.273
2/12/2006	363.98	0	1103.073
3/12/2006	431.6	0	1866.445
14/12/2006	444.23	0	2097.055
15/12/2006	429.13	0	2025.000
16/12/2006	539.85	0	2399.236

Table 5.2: Garri Power Plant Performance Data, year 2007

Date	m_f(Tons) LDO	m_f(Tons) LPG	P_{out} MWh
1/1/2007	122.85	0	290.945
2/1/2007	135.7	0	349.955
3/1/2007	138.7	0	380.718
14/01/2007	466.74	155.93	3052.636
15/01/2007	655.02	170.26	3872.382
16/01/2007	470.52	277.04	3372.473
29/01/2007	1109.0	0	5048.982
30/01/2007	1102.45	0	4949.891
31/01/2007	1000.06	0	4575.927
1/2/2007	817.58	7.67	3754.964

2/2/2007	632.11	0	2775.545
3/2/2007	764.49	0	3136.345
14/02/2007	881.97	53.22	3856.582
15/02/2007	1037.24	56.2	4944.964
16/02/2007	566.92	159.5	3693.536
26/02/2007	912.07	182.39	4956.273
27/02/2007	877.34	181.72	5036.800
28/02/2007	847.66	161.28	4813.655
1/3/2007	811.97	118.96	4183.073
2/3/2007	7419.51	0	2931.173
3/3/2007	1050.25	0	4219.000
14/03/2007	1037.81	184.57	5738.118
15/03/2007	872.34	182.29	5112.127
16/03/2007	872.34	182.29	3894.036
29/03/2007	1349.83	0	6123.945
30/03/2007	1032.41	0	4661.336
31/03/2007	1119.05	0	5045.027
1/4/2007	1135.51	0	5133.355
2/4/2007	1237.51	0	5349.464
3/4/2007	1338.89	0	5359.182
14/04/2007	1221.14	0	5101.945
15/04/2007	1299.02	0	5450.973
16/04/2007	1316.3	0	5981.809
28/04/2007	1292.36	0	5198.109
29/04/2007	1231.43	0	5145.564
30/04/2007	1030.71	0	4448.018
1/5/2007	1387.6	0	6102.436
2/5/2007	1381.31	0	6132.036
3/5/2007	1348.93	1	5716.018
14/05/2007	1291.79	0	5916.727
15/05/2007	1347.13	0	6162.345
16/05/2007	1401.89	0	6340.473
29/05/2007	1279.27	0	6326.063
30/05/2007	1280.4	0	6322.364

31/05/2007	1269.77	0	6131.064
1/6/2007	970.27	0	4704.355
2/6/2007	1211.05	0	5475.200
3/6/2007	1186.82	0	5435.982
14/06/2007	1171.17	0	5711.009
15/06/2007	896.11	0	4514.864
16/06/2007	1256.37	0	6148.555
28/06/2007	1274.67	0	7041.435
29/06/2007	1231.52	0	6067.436
30/06/2007	1152.08	0	5389.373
1/7/2007	1266.89	0	6200.009
2/7/2007	1399.67	0	7171.018
3/7/2007	1351.75	0	6970.882
14/07/2007	1207.96	0	6058.191
15/07/2007	1347.24	0	6971.900
16/07/2007	1371.14	0	7107.755
29/07/2007	1518.87	0	7699.573
30/07/2007	1458.38	0	7422.255
31/07/2007	1352.67	0	6859.418
1/8/2007	1315.31	0	6348.064
2/8/2007	1325.21	0	6696.619
3/8/2007	1122.87	0	5139.382
14/08/2007	1427.42	0	7342.300
15/08/2007	1391.97	0	7077.236
16/08/2007	1482.41	0	7405.927
29/08/2007	1458.75	0	7465.936
30/08/2007	1514.64	0	7674.455
31/08/2007	1178.14	0	5750.445
1/9/2007	997.14	0	5119.155
2/9/2007	1349.18	0	6661.773
3/9/2007	1433.93	0	7331.755
14/09/2007	1489.08	0	7430.000
15/09/2007	1480.28	0	7934.436
16/09/2007	1561.13	0	7994.918

28/09/2007	1293.93	0	6428.155
29/09/2007	1369.36	0	7047.627
30/09/2007	1407.46	0	7084.064
1/10/2007	1309.64	0	6689.200
2/10/2007	1387.7	0	6803.200
3/10/2007	1344.45	0	6714.218
14/10/2007	783.83	0	3377.909
15/10/2007	820.93	0	3645.918
16/10/2007	1076.84	0	5321.636
29/10/2007	1188.58	0	5135.018
30/10/2007	1212.39	0	5399.236
31/10/2007	1117.97	0	5418.127
1/11/2007	1118.05	0	5391.191
2/11/2007	863.61	0	4283.464
3/11/2007	1116.14	0	5914.509
14/11/2007	1177.41	0	5741.891
15/11/2007	1100.79	0	5364.927
16/11/2007	873.23	0	4121.927
28/11/2007	1041.27	0	4977.600
29/11/2007	1128.71	0	5637.809
30/11/2007	898.10	0	4112.509
1/12/2007	1133.3	0	5806.836
2/12/2007	1133.3	0	6213.882
3/12/2007	1250.33	0	6227.018
14/12/2007	1011.73	0	4966.009
15/12/2007	1097.89	0	4926.955
16/12/2007	1164.37	0	5439.482
29/12/2007	1037.28	0	5384.409
30/12/2007	1106.23	0	5713.373
31/12/2007	1505.65	0	6632.975

Sample of efficiency calculations:-

The input heat Q_{in} (kJ) = mass of fuel (m) *calorific value (CV)

Ton =1000 kg

$Q_{out}(kJ) = Q_{out}\,(MWh)*10^3*3600$

The input heat Q_{in} (kJ) $=m_{LPG} *\text{Low CV}_{LPG} + m_{LDO} *\text{Low CV}_{LDO}$

Low CV $_{LPG}$ = 45125 kJ /kg

Low CV $_{LDO}$ =42697.2 kJ/kg

$$\eta_{th} = \frac{Q_{OUT}}{Q_{in}} = \frac{Q_{OUT}}{m_{LPG} *\text{Low CV}_{LPG} + m_{LDO} *\text{Low CV}_{LDO}}$$

Where: Q_{in} is input heat (kJ)

Q_{out} is output heat (kJ)

m is mass of fuel in (kg)

CV is calorific value (kJ/kg)

2006

1 March

$Q_{out}\,(kJ) = Q_{out}\,(MWh)*10^3*3600$

$Q_{out} = 4436.3245 *10^3 *3600 =1.5970482E+10$ kJ

$Q_{in} = (m_{LPG} *\text{Low CV}_{LPG} + m_{LDO} *\text{Low CV}_{LD})$

$Q_{in} = [(943.9*10^3*42697.2) + (145.74*10^3*45125)]$

$Q_{in}=4.697840458E+10$ kJ

$$\eta_{th} = \frac{Q_{OUT}}{Q_{in}} = \frac{1.5970482E+10}{4.697840458E+10} = 0.33995$$

2 March

$Q_{out} = 4656.018*10^3*3600 =1.67616648E+10$ kJ

$Q_{in}[(937.18*10^3*42697.2) + (147.54*10^3*45125)]$

$Q_{in} = 4.66727044E+10$ kJ

$\eta_{th} = 0.359132$

3 March

Q_{out} = 3832.745*10^3*3600=1.3797882E+10 kJ

Q_{in} = [(3823.23*10^3*42697.2) + (149.06*10^3*45125)]

Q_{in} = 4.187594846E+10 kJ

η_{th} = 0.329494

14 March

Q_{out} =4982.52*10^3*3600=1.79370972E+10 kJ

Q_{in} =[(914.89*10^3*42697.2) + (178.05*10^3*45125)]

Q_{in} =4.70974756E+10 kJ

η_{th} = 0.380848

15 March

Q_{out} = 4819.418*10^3*3600=1.73499048E+10 kJ

Q_{in} =[(845.10*10^3*42697.2) + (201.27*10^3*45125)]

Q_{in} =4.516571247 E+10 kJ

η_{th} = 0.3841388

16 March

Q_{out} = 5234.473*^{103}*3600=1.88441028E+10 kJ

Q_{in} =[(949.01*10^3*42697.2) + (201.83*10^3*45125)]

Q_{in} =4.962764852E+10 kJ

η_{th} = 0.3797097659

29 March

Q_{out} = 5294.582*10^3*3600=1.90604952E+10 kJ

Q_{in} =[(991.87*10^3*42697.2) + (210.92*10^3*45125)]

Q_{in} =5.186783676E+10 kJ

η_{th} = 0.3674819771

<u>30 March</u>

Q_{out} = 5188.291*10^3*3600=1.86778476E+10 kJ

Q_{in} =[(1053.03*10^3*42697.2) + (179.04*10^3*45125)]

Q_{in} =5.304040191E+10 kJ

η_{th} = 0.3521437796

<u>31 March</u>

Q_{out}= 4180.164*103*3600=1.50485904E+10 kJ

Q_{in} =[(691.53*10^3*42697.2) + (201.07*10^3*45125)]

Q_{in} =3.859967847E+10kJ

η_{th} = 0.3898631024

Average of the efficiency for a month = [0.33995 + 0.359132 + 0.329494 + 0.380848 + 0.3841388 + 0.3797097659 + 0.3674819771 + 0.3521437796 + 0.3898631024] /9 = 36.48 %

Table 5.3: Calculated heat input (Q_{in}) and heat output (Q_{out}) using Garri Power Plant data, for year 2006:

Date	Q_{in}(kJ)	Q_{out}(kJ)
1/3/2006	4.70E+10	1.60E+10
2/3/2006	4.67E+10	1.68E+10
3/3/2006	4.19E+10	1.38E+10
14/3/2006	4.71E+10	1.79E+10
15/3/2006	4.52E+10	1.73E+10
16/3/2006	4.96E+10	1.88E+10

29/3/2006	5.19E+10	1.91E+10
30/3/2006	5.30E+10	1.87E+10
31/3/2006	3.86E+10	1.50E+00
1/4/2006	4.98E+10	1.89E+10
2/4/2006	4.56E+10	1.89E+10
3/4/2006	5.43E+10	1.87E+10
14/4/2006	4.49E+10	1.69E+10
15/4/2006	4.95E+10	1.80E+10
16/4/2006	5.12E+10	1.85E+10
28/4/2006	4.64E+10	1.69E+10
29/4/2006	5.03E+10	1.78E+10
30/4/2006	4.99E+10	1.77E+10
1/5/2006	4.96E+10	1.76E+10
2/5/2006	5.13E+10	1.81E+10
3/5/2006	4.14E+10	1.46E+10
14/5/2006	5.84E+10	1.96E+10
15/5/2006	5.24E+10	1.82E+10
16/5/2006	5.01E+10	1.76E+10
29/5/2006	5.63E+10	1.56E+10
30/5/2006	4.55E+10	1.61E+10
31/5/2006	4.58E+10	1.46E+10
1/6/2006	3.85E+10	1.36E+10
2/6/2006	2.94E+10	1.15E+10
3/6/2006	4.05E+10	1.37E+10
14/6/2006	4.33E+10	1.62E+10
15/6/2006	4.54E+10	1.64E+10
16/6/2006	4.78E+10	1.89E+10
28/6/2006	5.19E+10	1.81E+10
29/6/2006	5.21E+10	1.83E+10
30/6/2006	3.78E+10	1.33E+10
1/7/2006	4.65E+10	1.66E+10
2/7/2006	5.39E+10	1.88E+10
3/7/2006	4.86E+10	1.74E+10
14/7/2006	4.43E+10	1.65E+10

15/7/2006	5.07E+10	1.85E+10
16/7/2006	4.90E+10	1.79E+10
29/7/2006	5.10E+10	1.75E+10
30/7/2006	5.24E+10	1.84E+10
31/7/2006	5.08E+10	1.85E+10
1/8/2006	5.29E+10	1.93E+10
2/8/2006	5.42E+10	1.97E+10
3/8/2006	5.06E+10	1.85E+10
15/8/2006	3.77E+10	1.25E+10
16/8/2006	4.59E+10	1.54E+10
29/8/2006	5.43E+10	1.89E+10
30/8/2006	5.42E+10	1.95E+10
1/9/2006	3.77E+10	1.52E+10
2/9/2006	4.92E+10	1.81E+10
3/9/2006	5.22E+10	1.85E+10
14/9/2006	5.16E+10	1.80E+10
15/9/2006	4.59E+10	1.55E+10
16/9/2006	5.40E+10	1.81E+10
28/9/2006	4.69E+10	1.77E+10
29/9/2006	4.21E+10	1.61E+10
30/9/2006	4.73E+10	1.74E+10
1/10/2006	4.68E+10	1.74E+10
2/10/2006	4.49E+10	1.65E+10
3/10/2006	4.76E+10	1.76E+10
14/10/2006	4.67E+10	1.70E+10
15/10/2006	4.71E+10	1.75E+10
16/10/2006	4.56E+10	1.69E+10
29/10/2006	3.99E+10	1.41E+10
30/10/2006	3.53E+10	1.34E+10
31/10/2006	3.61E+10	1.36E+10
1/11/2006	3.21E+10	1.23E+10
2/11/2006	4.04E+10	1.58E+10
3/11/2006	3.38E+10	9.30E+09
14/11/2006	3.17E+10	1.33E+10

15/11/2006	2.71E+10	9.69E+09
16/11/2006	6.95E+09	8.86E+09
28/11/2006	2.05E+10	7.63E+09
29/11/2006	1.91E+10	6.95E+09
30/11/2006	1.79E+10	6.69E+09
1/12/2006	6.79E+09	1.51E+09
2/12/2006	1.55E+10	3.97E+09
3/12/2006	1.84E+10	6.72E+09
14/12/2006	1.92E+10	7.55E+09
15/12/2006	1.83E+10	7.00E+07
16/12/2006	2.31E+10	8.64E+09

Table 5.4: Calculated heat input (Q_{in}) and heat output (Q_{ou}t) using Garri Power Plant data, for year 2007:

Date	Q_{in}(kJ)	Q_{out}(kJ)
1/1/2007	5.25E+09	1.05E+09
2/1/2007	5.79E+09	1.26E+09
3/1/2007	5.92E+09	1.37E+09
14/01/2007	2.70E+10	1.10E+10
15/01/2007	3.57E+10	1.39E+10
16/01/2007	3.26E+10	1.21E+10
29/01/2007	4.74E+10	1.82E+10
30/01/2007	4.71E+10	1.78E+10
31/01/2007	4.27E+10	1.65E+10
1/2/2007	3.53E+10	1.35E+10
2/2/2007	2.70E+10	9.99E+09
3/2/2007	3.26E+10	1.13E+10
14/02/2007	4.01E+10	1.39E+10
15/02/2007	4.68E+10	1.78E+10
16/02/2007	3.14E+10	1.33E+10
26/02/2007	4.72E+10	1.78E+10
27/02/2007	4.72E+10	1.81E+10
28/02/2007	4.44E+10	1.73E+10

1/3/2007	4.00E+10	1.51E+10
2/3/2007	3.17E+11	1.06E+10
3/3/2007	4.48E+10	1.52E+10
14/03/2007	5.26E+10	2.07E+10
15/03/2007	4.55E+10	1.84E+10
16/03/2007	4.55E+10	1.84E+10
29/03/2007	5.76E+10	2.20E+10
30/03/2007	4.41E+10	1.68E+10
31/03/2007	4.78E+10	1.82E+10
1/4/2007	4.85E+10	1.85E+10
2/4/2007	5.28E+10	1.93E+10
3/4/2007	5.72E+10	1.93E+10
14/04/2007	5.21E+10	1.84E+10
15/04/2007	5.55E+10	1.46E+10
16/04/2007	5.62E+10	2.15E+10
28/04/2007	5.52E+10	1.87E+10
29/04/2007	5.26E+10	1.85E+10
30/04/2007	4.40E+10	1.60E+10
1/5/2007	5.92E+10	2.20E+10
2/5/2007	5.90E+10	2.21E+10
3/5/2007	5.76E+10	2.06E+10
14/05/2007	5.52E+10	2.30E+10
15/05/2007	5.75E+10	2.22E+10
16/05/2007	5.99E+10	2.28E+10
29/05/2007	5.46E+10	2.28E+10
30/05/2007	4.14E+10	2.28E+10
31/05/2007	5.42E+10	2.21E+10
1/6/2007	4.14E+10	1.69E+10
2/6/2007	5.17E+10	1.97E+10
3/6/2007	5.07E+10	1.96E+10
14/06/2007	5.00E+10	1.06E+10
15/06/2007	5.83E+10	1.63E+10
16/06/2007	5.36E+10	1.21E+10
28/06/2007	5.44E+10	2.53E+10

29/06/2007	5.26E+10	2.18E+10
30/06/2007	4.92E+10	1.94E+10
1/7/2007	5.41E+10	2.23E+10
2/7/2007	5.98E+10	2.58E+10
3/7/2007	5.77E+10	2.51E+10
14/07/2007	5.16E+10	2.18E+10
15/07/2007	5.75E+10	2.05E+10
16/07/2007	5.85E+10	2.56E+10
29/07/2007	6.49E+10	2.77E+10
30/07/2007	6.23E+10	2.67E+10
31/07/2007	6.23E+10	2.47E+10
1/8/2007	5.62E+10	2.29E+00
2/8/2007	5.66E+10	2.41E+10
3/8/2007	4.79E+10	1.85E+10
14/08/2007	6.09E+10	2.64E+10
15/08/2007	5.94E+10	2.55E+10
16/08/2007	6.33E+10	2.67E+10
29/08/2007	6.23E+10	2.69E+10
30/08/2007	6.47E+10	2.77E+10
31/08/2007	5.03E+10	2.07E+10
1/9/2007	4.26E+10	2.84E+10
2/9/2007	5.76E+10	2.40E+10
3/9/2007	6.12E+10	2.64E+10
14/09/2007	6.36E+10	2.67E+10
15/09/2007	6.32E+10	2.86E+10
16/09/2007	6.67E+10	2.88E+10
28/09/2007	5.52E+10	2.31E+10
29/09/2007	5.96E+10	2.54E+10
30/09/2007	6.01E+10	2.55E+10
1/10/2007	5.59E+10	2.41E+10
2/10/2007	5.92E+10	7.97E+15
3/10/2007	5.74E+10	2.42E+10
14/10/2007	3.35E+10	1.21E+10
15/10/2007	3.51E+10	1.31E+10

16/10/2007	4.60E+10	1.92E+10
29/10/2007	5.07E+10	1.85E+10
30/10/2007	5.18E+10	1.94E+10
31/10/2007	4.77E+10	1.95E+10
1/11/2007	4.77E+10	1.94E+10
2/11/2007	3.69E+10	1.54E+10
3/11/2007	4.96E+10	2.13E+10
14/11/2007	5.03E+10	2.07E+10
15/11/2007	4.70E+10	1.93E+10
16/11/2007	3.73E+10	1.48E+10
28/11/2007	4.45E+10	1.79E+10
29/11/2007	4.82E+10	1.03E+10
30/11/2007	3.83E+10	1.48E+10
1/12/2007	4.84E+10	2.09E+10
2/12/2007	4.84E+10	2.24E+10
3/12/2007	5.34E+10	2.24E+10
14/12/2007	4.32E+10	1.79E+10
15/12/2007	4.69E+10	1.77E+10
16/12/2007	4.97E+10	1.96E+10
29/12/2007	4.43E+10	1.94E+10
30/12/2007	4.72E+10	2.06E+10
31/12/2007	6.43E+10	2.39E+10

Table 5.5: Effects of Ambient Conditions on Performance 2006

Month	Ta°C Average	Heat input (kJ) Average	Heat output(kJ) Average	EFF %
3	28.5	4.68E+10	1.54E+10	36.48
4	31.5	4.91E+10	1.80E+10	36.77
5	34	5.01E+10	1.69E+10	34.5
6	34	4.30E+10	1.55E+10	36.25

7	32	4.97E+10	1.78E+10	35.79
8	31	5.00E+10	1.77E+10	35.32
9	32.5	4.74E+10	1.72E+10	36.35
10	32	4.33E+10	1.60E+10	37.17
11	27.5	2.55E+10	1.01E+10	36.75
12	24.5	1.69E+10	4.74E+09	37

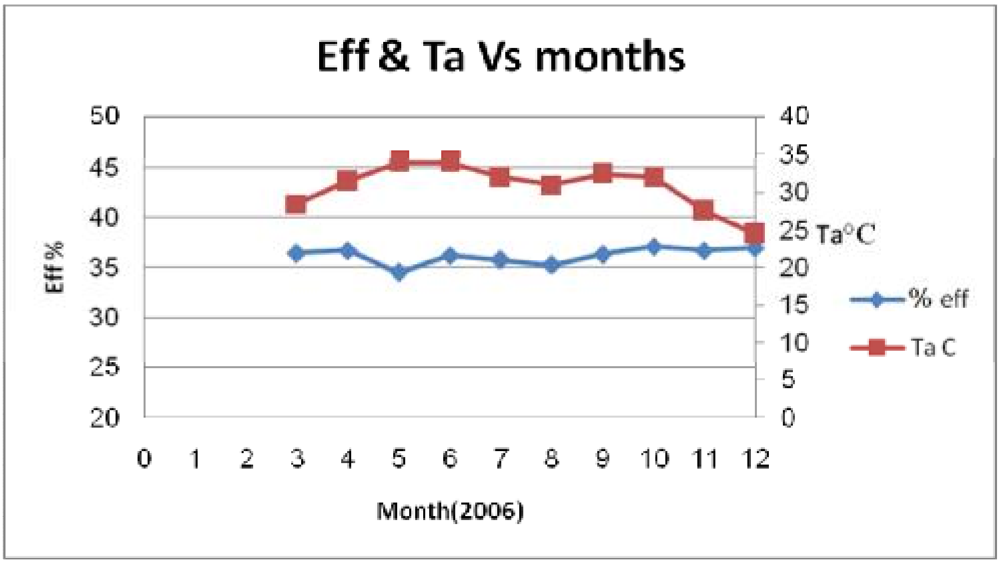


Figure 5.1: Thermal efficiency and ambient temperature during the year 2006

From figure 5.1: illustrates the variation of temperature and efficiency during the whole year. In March when the temperature is 28.5 ◦C the corresponding efficiency is 36.48%, for April the efficiency is nearly same as March, but when the temperature increases the efficiency decreases as shown in May and June (summer season) . For the remaining months of the year;

September, October, November, and December the efficiency is observed to as the temperatures drops in those months.

It is clear from the above that the efficiency is greatly affected by the ambient temperature of the air entering the compressor.

Table 5.6: Effects of Ambient Conditions on Performance 2007

Month	Ta°C Average	Heat input (kJ) Average	Heat output(kJ) Average	EFF %
2	22	3.91E+10	1.48E+10	38.02
3	25	7.72E+10	1.73E+10	37.87
4	31	5.27E+10	1.83E+10	35.86
5	35	5.54E+10	2.22E+10	36.84
6	33	5.13E+10	1.80E+10	40.11
7	27	5.87E+10	2.45E+10	42.89
8	28	5.80E+10	2.18E+10	41.92
9	31	5.89E+10	2.63E+10	38.01
10	31	4.86E+10	1.87E+10	38.42
11	30	4.44E+10	1.71E+10	40.94
12	26	4.95E+10	2.05E+10	41.09

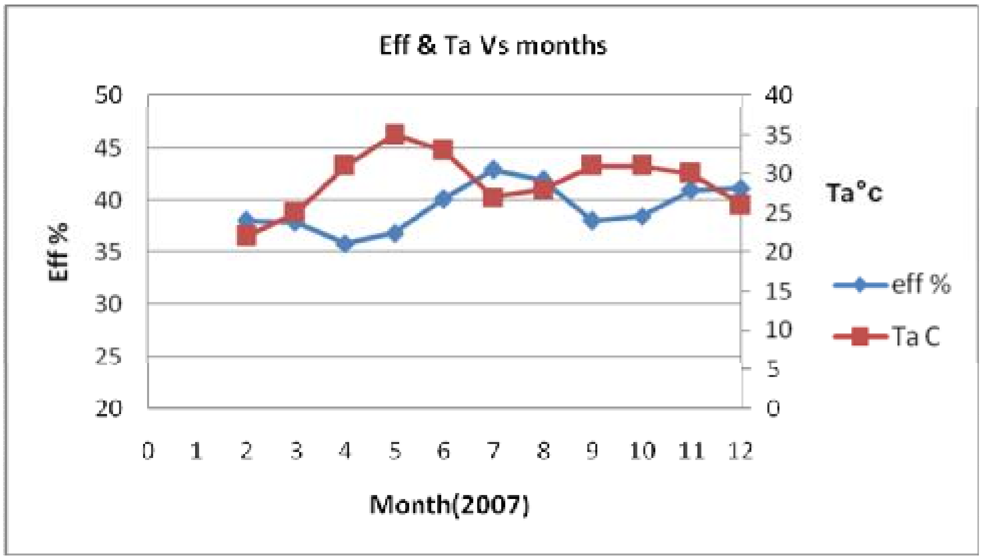


Figure 5.2: Thermal efficiency and ambient temperature during the year 2007

From figure 5.2: the effeciency decreases gradualy as the average temperature increase form February to April. In July the effeciency reach a maximum value when the ambient temperature is 27◦C due to the rainy season. The effecincey then decrease again in September and October due to increasing in the temperature .The efficiency again rises as the temperature drops in December (winter season).

In general the ambient conditions under which a gas turbine operates have a noticeable effect on both the power output and efficiency. The power decreases due to reduction in air mass flow rate (the density of air declines as temperature increases) and the efficiency decreases because the compressor requires more power to compress air of higher temperature. Conversely, the power and efficiency increase when the inlet air temperature is reduced .There is variation in power and efficiency for a gas turbine as a function of ambient temperature compared to the reference International

Organization for Standards (ISO) condition at sea level and 32.78°C .

At an inlet air temperature of near 55.55°C, power output can drop to as low as 90 %of ISO-rated power for typical gas turbines. At cooler temperatures of about 4.44 to 10°C, power can increase to as high as 105 %of ISO-rated power[1].

3. The effect of the fuel type on efficiency:

The effect of the type of fuel on part life is associated with the radiant energy in the combustion process and the ability to atomize the various liquid fuels. Therefore, natural gas, which does not require atomization, has the lowest level of gradient energy and will produce the longest life of parts.

Natural gas has been the traditional fuel for use with gas turbines in industrial applications. Limitations on the available supply of natural gas, with the resultant increase in costs, have led to the consideration of liquid fuels to a greater degree than at any time in the past.

Of the liquid fuels, distillate fuel will produce the next highest life, and crude oil and residual oils, with the attendant higher radiant energy and more difficult atomization, will produce shorter parts life.

Contaminants in the fuel also affect maintenance intervals. This is particularly true for liquid fuels in which dirt results in accelerated replacement of pumps, metering elements, and fuel nozzles. Contaminants in fuel gas can erode or corrode control valves and fuel nozzles. The limiting item to continuous operation on liquid fuels is the fuel nozzles. Exceptionally "clean” fuel can increase this interval, while "dirty” fuel will decrease it accordingly.

Effect of fuel type on efficiency:-

1. **Sample calculations:**

a. **In 15 February 2006**

For GT_4 (working with LDO)

Q_{out} = 329.91*10^3*3600 = 1187607600 kJ

Q_{in} = 103.66*10^3*42697.2 = 4425991752kJ

η_{th} = 0.26832

In 1 March 2006

For GT4 (working with LPG)

Q_{out} = 549.005*10^3*3600 = 1976418000 kJ

Q_{in} = 145.74*10^3*45125 = 6576517500 kJ

η_{th} = 0.300526

b. **In 25 March 2006**

For GT_4 (working with LPG)

Q_{out} = 794.182*10^3*3600 =2859055200 kJ

Q_{in} = 202.03*10^3*45125 =9116603750 kJ

η_{th} = 0.313609

In 15 Apri 2006

For GT_4 (working with LDO)

Q_{out} = 563.564*10^3*3600 = 2028830400 kJ

Q_{in} = 172.38*10^3*42697.2 = 7360143336 kJ

η_{th} = 0.27565

c. In 15 October 2006

For GT_2 (working with LDO)

Q_{out} = 610.909*10^3*3600 = 2199272400 kJ

Q_{in} = 190.15*10^3*42697.2 = 8118872580 kJ

η_{th} = 0.27088

In 3 November 2006

For GT_2 (working with LPG)

Q_{out} = 596.4*10^3*3600 = 2147040000 kJ

Q_{in} = 167.7*10^3*45125 = 7567462500 kJ

η_{th} = 0.2837

All the above results showed that the efficiency of the plant is quite high when using Liquid Petroleum Gas (LPG) and the difference is vary from 5 to 8% which indicates that the type of fuel being used has a significant effect on gas turbine thermal efficiency and consequently power output.

Chapter 6

Conclusion and Recommendations

6.1 Conclusion

The power of the gas turbine is directly proportion to the mass and density of the inlet air which directly affected by the ambient air temperature. As gas turbines designed to operate at standard conditions of 15.6°C so it loses significant portions of their generating capacity when installed in hot climate conditions like Sudan.

6.1.1 The efficiency is affected by the ambient temperature where it increases due to temperature decrease and these phenomena appear due to season’s variation

6.1.2 The efficiency of the plant is high while using LPG when compared to LDO this is due to two reasons, the first is the high LHV of LPG, the second is the mixture of LPG and the air is more homogenous than LDO mixture during the combustion process.

6.1.3 In comparison of using LPG and LDO as fuels for operating the plant it was found that the efficiency using small amount of LPG than LDO is higher, but due to shortage in the production of LPG locally it cannot be used continuously.

6.1.4 From economical side the price of LPG (600 pounds /ton) is very low compared to the price of LDO (1200 pounds/ton).

6.1.5 LPG is more pure than LDO which results in less failure of the turbine hot parts.

6.2 Recommendations

6.2.1 It is therefore highly recommended in a hot country like Sudan to use a cooling method to lower the temperature of ambient air entering the compressor.

6.2.2 Filters types and configurations should be selected on the basis of the environment in which they are expected to operate, the amount of maintenance available, and the degree of protection expected.

6.2.3 As LPG is more efficient and cheaper, so it should be used**.**

6.2.4 Using clean water in evaporative cooling to maintain the air filter clean.

6.2.5 Use of demineralized/deionized (DI) water to wet the evaporative cooler pads .

6.2.6 Use evaporative coolers or fogger systems in the gas turbine inlet to reduce temperature and increase power output and efficiency.

6.2.7 To reduce the turbine hot parts failures more concentration on fuel treatment are required when using the LDO.

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