

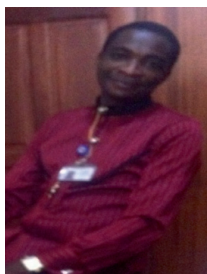
The major push for this work is insufficient power generation issues and the need to improve the availability of power. The work focuses on using pertinent performance parameters to evaluate the performance characteristics of a gas turbine: MS5001Nuovopignone engine of 25MW capacity. Also, the work deals explicitly on how the effects of ambient temperature on the gas turbine. However, comparison of the performance criteria of the actual working parameters of the gas turbine were made with the industry values and it was found that they prove significant correspondence. Hence, this work will be useful to post-graduate and undergraduate students who have interest in the field of gas turbine engineering as maintenance.



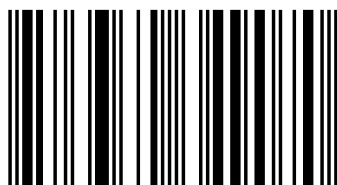
Emughiphel Nelson Igoma

Evaluation Performance Characteristic of a Gas Turbine

Evaluations, Effects and Comparison



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CHAPTER 1

INTRODUCTION

1.1 General

Gas turbines are widely used for generating electricity, operating airplanes and for sundry industrial applications such as refineries and petrochemicals.

Meanwhile, the basic gas turbine cycle has low thermal efficiency, so it is pertinent to look at improved gas turbine base cycle (Mahmood and Gadmahdi, 2009).

Therefore, enhancing the value of performance of gas turbine became successful through raising the turbine inlet temperature (TIT) and the compressor - pressure ratio. Moreover, gradual improvement in cooling technology and material science has caused high turbine inlet temperature obtainable (Ibrahim et al, 2010).

Gas turbine has been used for electricity generation in most countries around the world. In the past, their use has been generally limited to generating electricity in periods of peak electricity demand. Similarly, the gas turbines are ideal for this application as they can be started and be stopped quickly, enabling them to be brought into service as required thereby meeting energy demand at the most dependable level (Jaber et al, 2007). However, due to the availability of natural gas at very cheap prices compared to distillate fuels, many countries around the world (for example, Nigeria) use large conventional Gas Turbines as base load units. The small conventional Gas Turbines are used to meet any shortages in availability of electricity supplies occurring during peak load demand periods (Abam et al, 2012).

The overall efficiency of the gas turbine cycle depends primarily upon the pressure ratio of the compressor. It is pertinent to note that in the gas turbine, the processes of compression, combustion and expansion do not occur in a single component as they occur in a reciprocating engine. Consequently, a parametric study on the effect of operation conditions requires managing the operation conditions of the system (Rahman et al, 2012).

The gas turbine in its most common form is a heat engine operating by means of series of processes consisting of compression of air taken from the atmosphere, increase of gas temperature by constant – pressure combustion of fuel in the air, expansion of hot gases and discharge of the gases to the atmosphere. It is thus similar to the spark ignition (S.I) and internal combustion (I.C) engines in working medium but is akin to the steam turbine in its aspect of the steady flow of the working medium (Khajuria and Dubey, 1992). The gases expand during the work process and finally exhaust outside. The only difference is, instead of the reciprocating motion as in the case of the internal combustion (I.C) engines, gas turbine uses a rotary motion throughout (Zactruba, 2009).

Precisely, the Trans – Amadi gas turbine power station Phase II (GT-II) operates on an open cycle consisting of a compressor, a combustor and a turbine combined in series as shown in Figure 1.1. The combustor increases both the temperature and the specific volume of air. A net positive shaft work transfer is produced because the negative shaft work transfer required for powering the compressor is less than the positive work transfer produced by the turbine (Tsai, 2004).

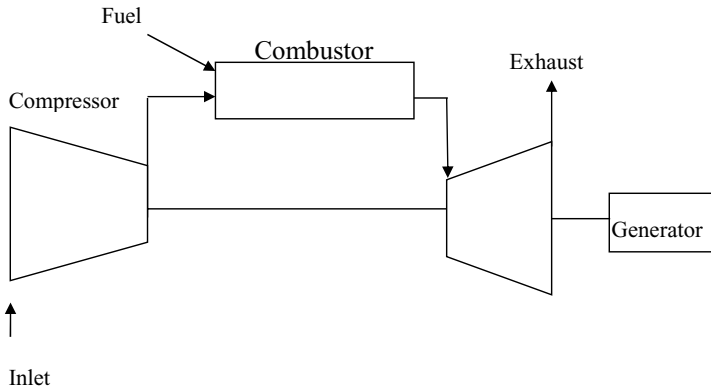


Figure 1.1: Simple-Cycle, Single-Shaft Gas Turbine

(Source: Trans-Amadi Gas Turbine Manual)

Precisely, the major sections of the Trans-Amadi gas turbine power station phase II are shown in Figure 1.2.

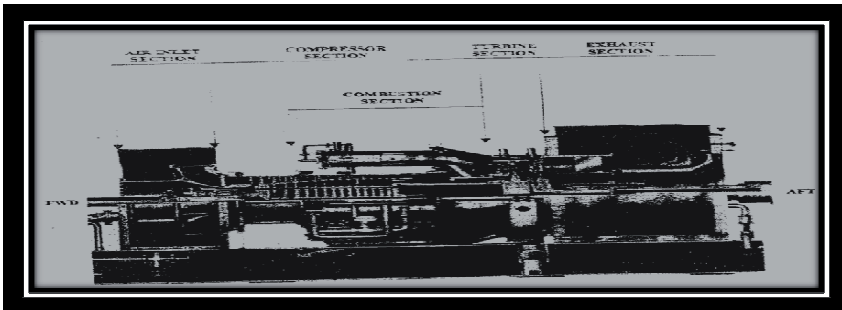


Figure 1.2: Major Sections of the Trans-Amadi Gas Turbine Assembly

(Source: Trans-Amadi Gas Turbine Manual)

Compressors: Compressors are samples of negative shaft work machines. They increase both the temperature and pressure of the air. Meanwhile, increasing the pressure of the air requires a negative work transfer.

Turbines: Turbines are samples of positive shaft work machines. They decrease both the temperature and pressure of the working fluid. This decrease creates a positive work transfer.

Combustor Chambers: The combustor chamber is the component of the gas turbine in which the fuel is combined with the air from the compressor and burned. The combustion chamber is modeled as a constant pressure device. The combustion process raises the temperature of air in the system by converting the chemical potential energy of the reactants to thermal energy. There is no work transfer involved in the reaction.

Air Inlet Package: The air inlet package is situated at the front of the generating set and provides filtered air to the gas generator because clean air is of great importance for the compression process. A two-stage filtration system is located at the top of the inlet tower supplemented by by-pass doors in the event of filter blockage. Noise attenuator splitters are fitted in the upper part of the inlet tower which leads to the main combustion chamber from where the gas turbine takes its air for compression.

Exhaust System: It is a rectangular system that vents off the residual gases from the gas generator through the power turbine exhaust volute. It is located between the gas generator and the power turbine and is made up of sound-proof materials to reduce the sound of the rushing gas.

1.2 Background of the Study

The Trans-Amadi gas turbine power station Phase II was commissioned in 2011 and was meant to complement power in the Phase I which is a steam turbine of lower capacity and the national grid which is inefficient.

Precisely, the power station of Trans-Amadi Gas Turbine, Phase II has four turbines each of which is installed to a capacity of 25 megawatts. Meanwhile, a combination of the four turbines would serve 100 megawatts and electricity generation capacity of about 85 megawatts at optimum productivity (85% of total installed capacity). Because of operational irregularities such as stepping up and down of voltages at the sub-stations; which is likely to cause thermal stress, the turbine had failed to produce the rated output. The consequential wear and tear had also reduced efficiency and warranted unwanted overhauls and repairs. The overall effect is that the turbine had not worked to its purpose since service areas still experience frequent power outages.

The frequent power outages in the areas the plant is programmed to service in the state prompted this study. The study would evaluate the performance characteristics of the Trans-Amadi gas turbine phase II and also seek ways of improving them.

1.3 Statement of the Problem

The major drive that prompted the evaluation of the performance characteristics of the Trans-Amadi gas Turbine power station Phase II are insufficient power generation issues and the need to improve the availability of power to meet the power needs of the nation.

Also, the effects of internal factors on gas turbine like pressure losses in the compressor inlet suction and turbine exhaust system, fuel type, air exhaustion from the axial compressor, steam and water injection, and evaporation cooling are also some of the basic problems that necessitated this study.

1.4 Aim of the Research

The main objective of this research is to evaluate the performance characteristics of Trans-Amadi gas turbine with a view of finding ways to improve the performance of the turbine. So that it will serve the purpose for which it was initiated.

1.5 Specific Objectives

The specific objectives of this research work include:

- i. To review the performance parameters of the Plant from January 2012 to January 2013.
- ii. To determine the performance parameters (turbine inlet temperature, total heat supplied, compressor work, turbine work , net work output, thermal efficiency , specific fuel consumption , heat rate, work ratio, thermal discharge index, net plant heat rate and net work ratio) of the plant.
- iii. To determine the effects of ambient temperatures and pressures on the performance of the power plant and how to control them.
- iv. To compare the design parameters to the actual working parameters of the plant.

- v. To make recommendations in order to achieve efficient performance of the plant.

1.6 Significance of the Research

The following are the significant contributions of the study:

- i. Determination of the power output and efficiency of the gas turbine power plant based on the parameters from daily readings and environmental conditions.
- ii. The research will help to determine the quantity of power generated by the gas turbine plant and provide information to area of potential improvement that when adequately effected, will go a long way in improving the efficiency of the turbine.
- iii. Contribution to knowledge.

1.7 Scope of the Research

This research will base its findings on the Trans-Amadi gas turbine power station Phase II. However, the environmental conditions of where the gas turbine is located would influence the performance of the gas turbine; so measures to improve the thermal efficiency would be the overall desired result.

This research is limited to the evaluation of the performance characteristics of the Trans-Amadi gas turbine power station Phase II: Gas Turbine(GT)-Unit II: MS5001 Nuovopignone engine using ambient temperature and performance criteria in the determination of the performance of the gas turbine.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Gas turbine plant which operates on Brayton's cycle in the simplest form, has undergone tremendous modifications. These different forms of gas turbine plants were made possible as a result of researches that were carried out on gas turbine performance parameters as well as solution on how to improve the performance of gas turbine over the years. All of these researches among other things considered the compressor inlet temperature and pressure; compressor discharged pressure and turbine inlet temperature as well as turbine exhaust temperature and also proffered solutions on how to improve the performance of the gas turbine plant. Some recent works on gas turbine performance are discussed below.

2.2 Recent Work on Gas Turbine Performance

Kwambai (2005) employed exergy based method in the analysis of Olkaria I Geothermal Power Plant (Kenya). The aim of the study was to determine the overall second law efficiency for the power plant, identify locations and processes where exergy is wasted or lost. The system and sub-system; were simplified to the control volumes and the flow process approximated to standard of quasi-steady flow processes. The desired exergy output was compared to the necessary exergy input.

The study investigated the total exergy of fluids received from the production geothermal wells as the available exergy for the system and that from the connected wells as the primary exergy input into the system. The net electrical energy sup-

plied to the transmission grid was the desired exergy output. Only physical exergy was considered, ignoring other forms of it because they were insignificant to the processes involved. The results were presented in a Grassman diagram and they showed that the major exergy losses occur in the steam transmission, turbines, condensers and gas ejectors with the condenser taking the lead with a 14 MW energy loss.

In conclusion, the high losses in the condenser were due to non-insulation, turbulence and possible blockages of nozzles which affect spraying. The causes of the exergy wastage in the turbines were linked to inter-stages leakages and frictional losses.

Ebadi and Gorji-Bandpy (2005) used exergy based methods to analyze the performance of a 116 MW gas turbine at Mahshahr (Iran). In this research, mass and energy conservation laws were applied to each component of the system. Quantitative exergy balance was done for each component and for the whole system; the exergy of the system was decomposed into thermal, mechanical, chemical exergy and entropy generated. The effect of the change in the inlet temperature on the exergetic efficiency and exergy in the plant was evaluated.

The results from the analysis revealed that the largest amount of exergy occurs in the combustion chamber by the air compressor and the turbine. The analysis also revealed that 60.97% of the total inlet exergy is in the plant of which the combustion chamber contributes more than 55%.

Yadav et al (2009) investigated the gas turbine characteristic by varying tip clearance and axial gap. In this investigation, turbine efficiency has been calculated for the difference combinations of axial gap and tip clearance. The results show that the efficiency is maximum for the combination of 3.5 mm axial gap and tip clearance of 3%.

But for the actual case, the tip clearance is 3% is not sufficient because of thermal expansion of rotor blade working under high temperature and rotating with high speed. Losses are less in case of minimum tip as compared to maximum tip clearance.

The second highest efficiency is for the set of axial gap of 3.5 mm and 5% tip of the optimum set value axial gap and tip clearance for the maximum performance. The maximum loss was for the case of 7% tip clearance. The leakages flow mixing with the rotor passage flow also reduces turbine stage efficiency; hence, the maximum tip efficiency is less.

Thus, from the result for different axial gap, efficiency increases with increase in axial gap to a maximum point and after then falls with further increase in axial gap. The torque on the first rotor is maximum as compared to second rotor because of large pressure drop. Mach number for rotor is maximum for the first rotor and it decreases for the second rotor.

Shih and Liu (2009) studied the combustion characteristic of a Can-Type gas turbine combustor with rotating casing for an innovative micro gas turbine. In this study, a can-type combustor with rotation casing for the innovative micro gas tur-

bine has been modeled, and the combustion characteristics are investigated. The simulation are performed using commercial code STAR-CD in which three-dimensional compressible k- ϵ turbulent flow model and one-step overall chemical reaction between propane and air were used.

The results include the detailed flame structure at different rotation speed outside casing; ranging from stationary to the maximum 58000 rpm of the design point. Comparing the cases with and without rotating casing, the airflows are prevented from entering liner holes due to the centrifugal force and the inlet angle is inclined as casing rotates.

When the rotation is in the opposite direction of the swirling flows driven by the designed swirled, a shorter but broader recirculation and flame are found. At full speed of rotation (58000 rpm), the swirling direction follows the rotating direction of the casing, no matter what the designed swirled produced.

Higher combustion efficiency and higher exit temperature at higher rotating speed, which benefits the performance of gas turbine, but the cooling and possible hot spots for turbines are the primary concerns.

Lebele-Alawa (2010) studied the axial-thrust responses due to a gas turbine's Rotor Blade distortions. In this study, the axial thrust is imposed on the shaft gas turbine which depends upon its rotor blade inlet inclination to the turbine's axial direction. This inclination can change due to the distortions resulting from the tip rubbing, erosion, and corrosion, aging, foiling and thermal – fatigue cracks. Meanwhile, relevant influential parameters while the gas turbine was operating were

measured. Moreso, theoretical predictions for the behavior of the same turbine were obtained from computer simulations. The results of both measured parameters and theoretical prediction parameters were compared to each other and it showed qualitative correspondence. The rotor blade profile distortions result increased significantly when the axial thrust is on the compressor which unfavourably affects the gas turbine's thermodynamic performance, reliability and operational life.

Adegboyega and Odeyemi (2011) investigated the performance of a thermal power Station: A case study of Egbin power Station, Lagos, Nigeria. The station consists of six units of steam turbines with installed capacity of 1320 MW and the investigation looked at the efficiency and reliability. Field data were collated from the station records of operations and maintenance for the period of 2000 to 2010. The data collated reveals that the average overall efficiency of the plant for the period under review was 34.67% as against the standard of 40-50% while the average reliability was 80.92% as against best practice of 98-100%. The shortage in the performance level of the plant is caused by the low plant availability due to frequent breakdown-failure, overdue overhauling of some units, out – of - date technology, epileptic power supply: unstable supply of the national grid system, aging plant equipment and disruption in gas supply among others.

Finally, measures to improve the performance indices like the efficiency and reliability of the plant were enumerated and they include: overhaul maintenance of due equipment-units, training and re-training of operation and maintenance (O&M)

personnel, upgrading of out - of - date technology, ensuring security of gas supply and rehabilitation of failed equipment.

Asuo (2011) carried out an exergy analysis of a gas turbine power plant at Imiringi, Bayelsa State (that is, the Kolo Creek gas turbine plant). In his study, readings were taken based on the operating parameters for the Kolo Creek power plant for a period of two years (2004-2006) from the control room logbooks. The results taken were statistically analyzed and computed.

Relevant plant and working fluid parameters were obtained from appropriate thermodynamic tables. The volume and mass flow rates of air and natural gas were computed from standard thermodynamic equations and energy values at prescribed locations in the plant and these values were used to evaluate the plant performance in terms of thermal efficiency, fuel utilization and heat rate. The results revealed that the exergy efficiencies of the turbine were between 91.2% and 19%. The results also showed that as the system operates close to its maximum rated capacity (20 MW), the efficiency and hence other performing parameters were improved.

Finally, it was suggested that water should be injected in the compressor to act as intercooler between compressor stages so as to reduce the compressor work.

Obodeh and Isaac (2011) carried out the performance analysis for Sapele thermal power station. In their work, the performance indices were measured with percentage of shortfall of energy generated, load factor and utilization factor of the Sapele thermal station within the period 1997 to 2006. The thermal station consists of six units of steam turbine and four units of gas turbine with total installed capacity of 1020

MW. But it was found out through their study that less than 17% of the installed capacity was available. The percentage shortfall of power generated for the period under review ranged from 27.4% to 49.1%. This was due to aging of the plant equipment, improper operation and maintenance.

The load factor was between 39.9% and 64% as against international best practice of 80%. The inability of Power Holding Company of Nigeria (PHCN) to do turn-around maintenance on the plant advance for the dismal value. Average plant availability of the plant during the studied period was less than 21% as against industry best practice of over 95%. The capacity factor was even more dismal for the period under review; which ranked from as low as 5.49% in 2006 to the highest value of 17.19% in 1997. The low capacity factor indicates excessive plant failure. Measures to improve the performance indices of the plant has been suggested to include the following: training of personnel; operation and maintenance (O & M) regularly and improvement in O and M practices; proper spare parts inventory; organizing regular management meeting and improvement of general housekeeping of the plant.

Ibrahim et al (2011) studied ways of improving gas turbine performance using inlet air cooling systems. In this study, the performance of the gas turbine depends mainly on the inlet air temperature, while the power output of gas turbines depends on the flow of mass through it. This is precisely the reason for on hot days - when air is less dense, power output falls off. A rise of 1°C temperature of inlet air decreases the power output by 1%.

Also, the paper review up to date techniques that were developed to cool inlet air of gas turbine. Meanwhile, the development that occurred in the inlet air cooling system had been used to improve the performance of the gas turbine power plants. The conclusions of the study are summarized below:

- i. However, multiple use of this system to achieve cooling efficacy reflects the necessity of this technique in the improvement of the performance of the gas turbine plants.
- ii. The success of evaporative cooling in reducing the high air temperature depends on relative humidity of the ambient air. These types of systems are economical and suitable for hot and dry climates rather than hot and humid ones.
- iii. Absorption system is similar to vapour-compression air conditioning systems, except the pressurization stage. The adsorption cooling technique demonstrated a higher gain in power output and efficiently than evaporative cooling for a simple cycle gas turbine which is independent of the ambient conditions.
- iv. The adsorption chiller system for inlet air cooling of the gas turbine increases the peak capacity of the gas turbines during the hot ambient operation.

Ighodaro and Aburime (2011) conducted exergetic appraisal of Delta State IV power station, Ughelli. In their study, exergy is used to measure the performance of Delta IV power station. The choice of exergy as a means for performance appraisal was based on the fact that it provides as bird's eye view of the sources of the major losses in the system, thus giving us a clue on how best to optimize the system under observation.

Analyses are done using the first and second law of thermodynamics and results are presented in tables. In the exergetic analyses, mass and energy conservation laws were applied to each component. The compressor was seen to have the largest exergetic efficiency while that of the total plant was 45.7%. The combustion chamber had the largest exergy loss (56%) while that of the total plant was 58.5%.

The exergetic balance applied to the power station revealed how much of the unstable work potential or energy; supplied as the unit to the system under which consideration has been consumed (irretrievably lost). The process of the energy losses or irreversibilities will provide a general applicable quantitative measure of process inefficiencies thus enhancing optimization.

Rahman et al (2012) studied the thermodynamic performance analysis of a gas turbine power plant. In this study, the variation of operating conditions like compressor pressure ratio, turbine inlet temperature, exhaust temperature, air /fuel ratio, isentropic efficiencies of turbine and compressor, and the compressor inlet temperature on the performance of the gas turbine thermal efficiency, compressor work, power output, specific fuel consumption and heat rate are looked at. Analytically, the formula for the specific work and the efficiency were derived and possible analysis taken in consideration. The MATLAB Software was used in developing the programme of the performance model of the gas turbine. The results show that the compressor pressure ratio, turbine inlet temperature, air/ fuel ratio as well as the isentropic efficiency have a great influence on the thermal efficiency. In addition, the thermal efficiency and the power output drop linearly with an appreciable rise of the com-

pressor inlet temperature and the air /fuel ratio. In addition, the specific fuel consumption and heat rate rise linearly with the increase of both the compressor inlet temperature and the air/ fuel ratio.

Finally, the thermodynamic parameters on the cycle performance are economically possible and advantageous to the gas turbine operations.

Ibrahim and Rahman (2012) investigated on the effect of compressor-pressure ratio on the performance of combined cycle gas turbine. Thermodynamically, the work used a complete model of combined cycle gas turbine power plant with effect of the gas turbine peak pressure ratio. The simulated modeling results include the following:

- i. The compressor pressure ratio, air/ fuel ratio as well as the isentropic efficiencies influences greatly on the overall thermal efficiency of the combined cycle gas turbine plant.
- ii. The overall thermal efficiency rises and the total power output drop linearly with the rise of the compressor pressure ratio with constant turbine inlet temperature.
- iii. Higher efficiencies for combined cycle gas turbine compared to gas turbine plants was quoted to be about 61%
- iv. The peak overall efficiency happened at the higher compression ratio with the higher cycle peak temperature as well as higher isentropic compressor and turbine efficiencies.

The overall efficiency for gas-steam combined cycle after the utilization of the MATLAB software in the simulation model was found to be higher when compared to gas turbine plants.

Oyedepo and Kilanko (2014) carried out the thermodynamic analysis of gas turbine power plant modeled with evaporative cooler. In this study, the performance of the gas turbine plants was improved or enhanced by cooling the compressor intake air with evaporative cooler and the effect of the inlet cooling system on the performance of a gas turbine power plant with installed capacity of 33.5 MW- barge types. The results show that for every 5°C drop of inlet air temperature, the power output was raised between 5-10% and the efficiency rises between 2-5%. It was found that the amount of this increment is higher when the pressure ratio is high and turbine inlet temperature is low.

In a nutshell, the findings of this study show that adding the cooling system: evaporative cooler(retrofitting) of the existing gas turbine plant gives a much better system performance which is economically appreciable and advisable.

Poku and Ogbonnaya (2014) investigated on the Effect of Evaporative Cooling on the Performance of a Gas Turbine Plant located in Bayelsa State, Nigeria. The investigation critically and holistically examined the influence of an evaporative cooler on turbine inlet air and how the turbine reacts to the compressor inlet temperature. The examination make used of the actual operational conditions like compressor inlet temperature and relative humidity in the determination of the following: compressor work, turbine net work, specific fuel consumption and the thermal effi-

ciency of the plant. The findings showed that as the compressor inlet temperature was lowered, the compressor work reduces, the turbine net work increases, specific fuel consumption drops and the efficiency of the plant increases appreciably. It was perceived that the effect of the evaporative cooling is that a decrease in the compressor inlet temperature of 2.4°C leads to a rise of about 0.14% and 2.02kJ/kg in the efficiency and net work respectively. It was found that the specific fuel consumption decreased by 0.002kg/KWh.

Hence, the investigation shows that gas turbine plants work well in temperate region than tropical. To enhance the performance of the existing gas turbine plants in high temperature climates, retrofitting of an air cooler that will always reduce the temperature back or close to the design temperature before compression is necessary. Lebele-Alawa and Jo-Appah (2015) studied the thermodynamic performance analysis of a gas turbine in an Equatorial Rain Forest Environment. In the study, steady state monitoring and direct collection of data from MKIV speedtronics system and collation of logbooks data from the control room was done. The variation of operating conditions(ambient temperature, compressor discharge temperature, turbine inlet temperature, exhaust temperature and mass flow rate) on the performance of the gas turbine thermal efficiency, net power output, heat rate, specific fuel consumption and compressor work were investigated utilizing various appropriate thermodynamic equations and formulae. The results show that 1°C rise of the ambient temperature will cause the following: 1.37% drop in the net power output, 1.48% rise in the power drop, 1.49% drop in the thermal efficiency, 2.16%

rise in the heat rate, 2.17% rise in the specific fuel consumption and 0.3% rise in the compressor work. The thermal efficiency drops by 0.006% for every 1 Kcal/KWh rise in the heat rate and the heat transfer in the hot gas part was found to rise by 0.16% for every 1°C rise in the compressor inlet temperature.

The study revealed that the gas turbine had a large decrease in the power output due to the influence of site parameters which is not in agreement with designed parametric data.

2.3 Summary of Literature Review

The researches reviewed have shown the importance of evaluation of gas turbine performance and its characteristics. Also it shows the importance of exergy analysis in evaluating performance of gas turbine. Moreover, it elucidates the importance of the continuation of study of gas turbine plant. Studies emphasized that the gas turbine plant as vital as it is depends on various modifications for it to perform efficiently. These modifications became necessary as a result of the gas turbines operating parameters that were reviewed.

Furthermore, this study will go further to look at the evaluation of the performance characteristics of the Trans-Amadi gas turbine power station Phase II and relate it to some of the studies reviewed.

CHAPTER 3

MATERIALS AND METHODS

3.1 Introduction

The research methodology involved collection of data through direct measurements and from actual plant operational logsheets: turbine logsheets, plant-auxiliaries logsheets and generator logsheets for the months of January 2012 to January 2013 (Appendices A and B). The salient reason for these months is pertinent to the actual field investigation of the individual component performance may be efficiently carried out. Parameters which could not be directly measured or determined were derived utilizing appropriate thermodynamic relationships.

The parameters investigated and considered during the data collection were the pressures, temperatures and mass flow rates at various temperature and pressure points in the gas turbine.

However, in the evaluation and treatment of the data, statistical methods were used to calculate the mean values of daily parameters. This was done for every month that was under consideration and the average was taken at the end of each month. The actual performance of the power plant over the period of its installation was determined from their average parameters: inlet pressures, outlet pressures, inlet temperatures, outlet temperatures, mass flow rates and compressor works.

Precisely, the performance of the plant was also investigated using the performance criteria such as work ratio, net work ratio, thermal discharge index and net plant heat rate (Lebele-Alawa and Asuo, 2011).

3.2 Analytical Method

The Trans-Amadi power plant Phase II is a simple Brayton cycle comprises a Gas Generator (GG), the power turbine and auxiliary equipment required to operate it. The power station is made up of four gas turbine units (Unit I, Unit II, Unit III and Unit IV). It is of the dual fuel type - that is, it can operate on liquid fuel or gas supplied by the Shell Petroleum Development Company (SPDC) of Nigeria. The process of fuel supply is accomplished through a gas conditioning skid.

In this study, the Gas Generator contains the axial flow compressor driven by single axial flow turbine and the combustor (Figure 1.1). The power station uses the simple gas turbine cycle called the Brayton (Joule) cycle and it operates as an open system: since it takes air from the atmosphere and exhausts into the atmosphere after compression and expansion.

Furthermore, the Brayton (Joule) cycle which is the thermodynamic cycle upon which this gas turbine operates can further be analysed using the Figure 3.1:

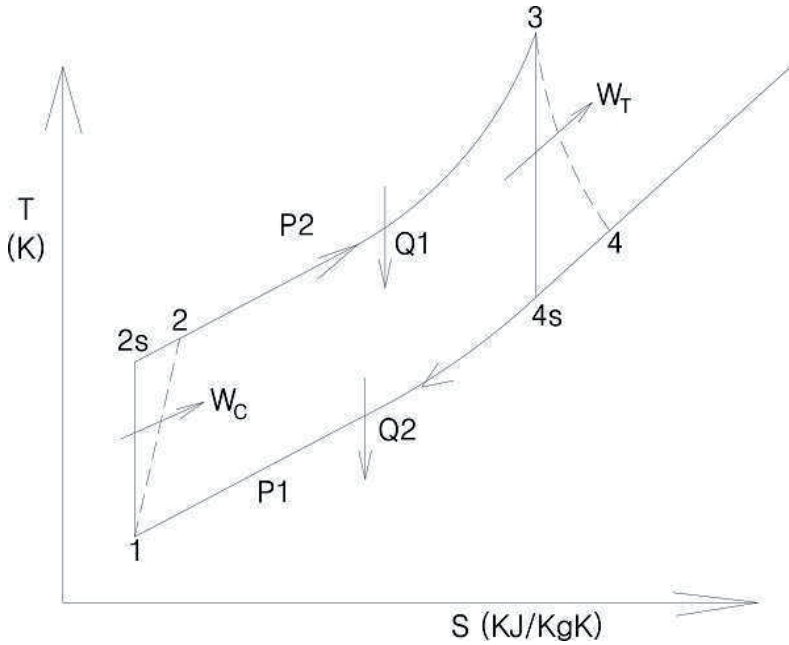


Figure 3.1: T-S Diagram for Brayton Cycle

(Source: Nag, 2001)

In evaluating the performance of the plant, the following assumptions and considerations were made:

- (i) Compression and expansion processes are reversibly adiabatic and thus involve no changes in entropy (Isentropic Process).
- (ii) Where no actual data were available, design values were used in calculations.
- (iii) Values of specific heats vary throughout the cycle due to changes in temperature.

(iv) Pressure drops across the combustion system are assumed constant.

(v) The Trans-Amadi Gas Turbine Power Plant is a simple system comprising the Gas Generator and the auxiliary equipment and it operates in an open cycle.

In view of the fact that some parameters like the intermediate temperature and pressure between compressors and turbines were not readily available the system was simplified to comprise a single compressor driven by a turbine and combustor in order to ease the analysis as shown in Figure 1.1. By applying steady flow energy equation:

$$\dot{Q} - \dot{W}_{SHAFT} = \sum_{in} \dot{m} \left(h + \frac{v^2}{2} + gz \right)_{in} - \sum_{out} \dot{m} \left(h + \frac{v^2}{2} + gz \right)_{out} \quad (3.1)$$

$$\dot{Q} + P_1 v_1 + u_1 + \frac{v_1^2}{2} + gz_1 = \dot{W} + P_2 v_2 + u_2 + \frac{v_2^2}{2} + gz_2 \quad (3.2)$$

But

$$h = Pv + u \quad (3.3)$$

where Pv = Flow work

By neglecting changes in K.E and P.E, the steady flow energy equation reduces to:

$$0 = \dot{Q} - \dot{W}_{SHAFT} + \dot{m}(h_{in} - h_{out}) \quad (3.4)$$

The first process of the Brayton cycle is adiabatic compression from P_1 to P_2 .

$$\dot{W}_{1-2} = \dot{m}(h_1 - h_2) \quad (3.5)$$

The second process is an addition of heat at constant pressure.

$$\dot{Q}_{2-3} = \dot{m}(h_3 - h_2) \quad (3.6)$$

The third process is isentropic expansion:

$$\dot{W}_{3-4} = \dot{m}(h_3 - h_4) \quad (3.7)$$

Finally, the fourth process is constant pressure exhaustion of hot gases.

$$\dot{Q}_{4-1} = \dot{m}(h_1 - h_4) \quad (3.8)$$

Mathematically, we have the following from Figure 3.1:

Heat Supplied or Heat Added:

$$\dot{Q}_1 = \dot{m}_a C_p (T_3 - T_2) \quad (3.9)$$

The Heat Rejected or Heat Removed will be;

$$\dot{Q}_2 = \dot{m}_a C_p (T_4 - T_1) \quad (3.10)$$

The Turbine Work is equal to:

$$\dot{W}_T = \dot{m}_p C_p (T_4 - T_3) \quad (3.11)$$

The Net Power Output is the power generated by the generator and is given as;

$$\dot{W}_{Net} = \dot{W}_T - \dot{W}_C \quad (3.12)$$

Therefore, the energy balance in the combustion chamber is expressed as (Rahman et al, 2011):

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \times LHV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \times T_3 \quad (3.13)$$

where: LHV = 47541.6kJ/kg (Oyedepo and Kilanko, 2014)

After manipulating Equation 3.11; the fuel ratio ‘f’ is expressed as:

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} \times T_3 - C_{pa} T_2}{LHV - C_{pg} T_3} \quad (3.14)$$

The Total Heat Supplied is expressed as (Lebele - Alawa and Jo-Appah, 2015):

$$\dot{Q}_{added} = (\dot{m}_a + \dot{m}_f) \times C_{pa} (T_3 - T_2) = \dot{m}_f \times CV \quad (3.15)$$

Meanwhile, the Isentropic Efficiency of the Turbine will be:

$$\eta_{ts} = \frac{\text{Actual Work}}{\text{Isentropic Work}} = \frac{T_3 - T_4}{T_3 - T_{4s}} \quad (3.16)$$

Therefore, T_{4s} can be defined as:

$$T_{4s} = \frac{T_3}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}} \quad (3.17)$$

$$P_2 = P_3 \text{ and } P_4 = P_1$$

Turbine Pressure Ratio = P_3/P_4 and the Compressor Pressure Ratio = P_2/P_1 .

The Compressor Work is calculated from the mass flow rate and enthalpy change across the compressor as follows:

$$W_c = \dot{m}_a C_{pa} (T_2 - T_1) = \rho_a V_a C_{pa} (T_2 - T_1) \quad (3.18)$$

Thermal Efficiency: The gas turbine efficiency is the percentage of the total fuel energy input that appears as the net work output of the cycle.

$$\eta_{th} = \frac{\text{Net Work}}{\text{Total Heat Supplied}} = \frac{W_{NET}}{Q_{added}} \quad (3.19)$$

Where: the net work is the power output and is given by Equation (3.12)

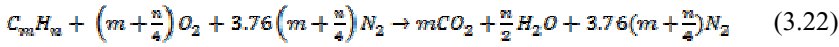
Specific Fuel Consumption: The ratio of fuel used by a machine to a certain force such as the amount of power in the machine produced. And it can be determined by the equation:

$$SFC = \frac{3600 \times \dot{m}_f}{W_{net}} \quad (3.20)$$

Heat Rate: This is a measure used to determine how efficiently a device uses heat energy. It can be expressed as:

$$HR = \frac{\text{Heat Supplied}}{\text{Power Generated}} = \frac{1}{\eta_{th}} \quad (3.21)$$

Stoichiometric equation: This is the ideal combustion process in which minimum amount of air (Stoichiometric or theoretical air) is needed to completely burn a fuel.



Also from equation (3.18) the Specific Fuel Consumption can also be calculated with the following formula:

$$SFC = \frac{3600}{AFR \times W_{net}} \quad (3.23)$$

Therefore, Air Fuel Ratio:

$$AFR = \frac{LHV}{Q_{added}} \quad (3.24)$$

Performance Criteria (Lebele-Alawa and Asuo, 2011)

$$(i) \quad \text{Work Ratio} = \frac{\text{Net Work Output}}{\text{Gross Work Output}} \quad (3.25)$$

Gross work = sum of work done by Compressor and Turbine

(ii) Thermal discharge index: This is the number of thermal energy units' discharge to the environment for every unit of electrical energy generated by a power plant. According to EL Wakill (1985), thermal discharge index is strongly dependent on the thermal efficiency of the plant, the lower it is: the more efficient is the power plant. Thus a lower value of thermal discharge index needs to be achieved for efficient power plant.

$$TDI = \frac{\text{Thermal Discharge to the Environment (MW)}}{\text{Electrical Output (MW)}} = \frac{P_{th}(1-\eta_{th})}{P_{th}\eta_{th}} = \frac{1}{\eta_{th}} - 1 \quad (3.26)$$

(iii) Net Plant Heat Rate: This is an important measure of plant performance and it is defined as the amount of fuel energy input required to generate one KWh and deliver it to the transmission lines. According to (Nag, 2001);

$$NPHR = \frac{3600}{\eta_{th}} \quad (3.27)$$

(iv) Net Work Ratio: This is the ratio of the net work to the heat that is removed in order to return the turbine exhaust to its initial temperature and pressure of state 1 (Figure 3.1).

$$NWR = \frac{W_{net}}{Q_{removed}} \quad (3.28)$$

Total Heat Removed: $Q_{removed} = (m_a + m_f) C_{pg} (T_3 - T_4)$

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Results

The parameters in Table 4.1 were obtained directly from the operational manual of MS5001 Nuovopignone Gas Turbine. The values that are in Table 4.2 are values for the compressor exit temperature, exhaust temperature, power output or average power generated and mass flow rate of the fuel which were obtained directly from the turbine logsheets, generator logsheets and plant-auxiliaries logsheets respectively. Precisely, the values for the compressor exit temperatures were determined from the direct reading of the turbine logsheets: wheelspace temperatures. At the wheelspace temperatures; the row that has the least temperature was chosen and the average temperature taken and recorded. Similarly, the exhaust temperatures were obtained from the direct reading of the turbine logsheets: exhaust thermocouples. At the exhaust thermocouples; the row that has the highest temperature was chosen and the average temperature taken and recorded. The values for the power output or average power generated were obtained from direct reading of the generator logsheets and the average power generated calculated and recorded while the values for the compressor exit pressure and mass flow rate of the fuel were obtained from direct reading of the plant-auxiliaries logsheets and their respective average values calculated and recorded (Appendices A and B); while the values for turbine inlet temperature, compressor work, turbine work, net work, total heat supplied or added, air fuel/ ratio, thermal efficiency, specific fuel consumption

and heat rate were obtained by calculation: using appropriate equations

(3.14,3.18,3.10,3.12,3.15,3.24,3.19,3.23,and 3.21 respectively).

Table 4.3 contains the values of the performance criteria such as work Ratio, thermal discharge index, net plant heat rate and net work ratio. Their values were obtained by calculation using Equations 3.25, 3.26, 3.27, and 3.28 respectively.

Table 4.4 contains the values of the actual parameters such as power output, thermal efficiency, specific fuel consumption and heat rate compared to their respective design values.

Finally, the effects for every 1°C rise in the ambient temperature with the power output, power differential, thermal efficiency ratio, specific fuel consumption and heat rate were determined and plotted as shown in Figures 4.1-4.5 respectively while Figure 4.6 shows the effect of the heat rate on the thermal efficiency.

Table 4.1: Design Parameters and their Values

S/N	Parameters	Unit	Design Data
1	Power Output	MW	25
2	Thermal Efficiency	%	26.6
3	Heat Rate	Kcal/W.h	2.833
4	Specific Fuel Consumption	kg/KW.h	0.308
5	Ambient Temperature	$^{\circ}\text{C}$	25.0-45.0
6	Specific Heat at Constant Pressure of gas (C_{pg})	kJ/kgK	1.155
7	Specific Heat at Constant Pressure of Air (C_{pa})	kJ/kgK	1.005
8	Isentropic Constants for air(γ_{air})	None	1.40
9	Isentropic Constants for gas(γ_{gas})	None	1.33
10	Mass Flow Rate of air	kg/s	122.9

Table 4. 2: Turbine Working Parameters (From January 2012 to January 2013)

S/N	T,°C Ambient Temperature	T,°C Compressor Exit Temperature	T,°C Turbine Inlet Temperature	T,°C Exhaust Temperature	ḡ _{in} (kg/s) Fuel Supply	W _c (KW) Compressor Work	W _t (KW) Turbine Work	ḡ _{exhaust} (KW) Heat Supplied	W _{net} (KW) Net Work	AFR Air Fuel Ratio	η _{th} (%) Thermal Efficiency	SFC (kg/KWh) Specific Fuel Consumption	HR (Kcal/W.h) Heat Rate	(MW) Power Output
1	25	240	1017	378	2.60	26556	1670	98002	24886	0.485	25.39	0.298	3.94	11.14
2	26	242	1025	382	2.62	26679	1693	98774	24986	0.481	25.30	0.300	3.95	11.14
3	27	244	1032	384	2.64	26803	1719	99420	25084	0.478	25.23	0.300	3.96	11.13
4	28	246	1041	385	2.66	26926	2015	100319	24911	0.474	24.83	0.305	4.03	11.13
5	29	247	1045	387	2.67	26926	2029	100706	24897	0.472	24.72	0.306	4.05	11.13
6	30	248	1049	390	2.68	26926	2040	101093	24886	0.470	24.62	0.308	4.06	11.13
7	31	250	1054	392	2.69	27050	2054	101479	24996	0.470	24.60	0.308	4.06	11.12
8	32	254	1092	394	2.80	27420	2257	105863	25163	0.449	23.77	0.319	4.20	11.11
9	33	257	1101	388	2.82	27667	2322	106638	25345	0.445	23.77	0.319	4.20	11.10
10	34	258	1127	400	2.90	27667	2435	109867	25232	0.432	22.97	0.330	4.35	11.09
11	35	260	1135	389	2.92	28451	2516	110643	25935	0.430	23.44	0.322	4.27	11.08
12	36	262	1156	388	2.98	28591	2643	113099	25948	0.420	22.94	0.330	4.36	11.07
13	37	265	1165	379	3.00	28849	2724	113877	26125	0.418	22.94	0.330	4.36	11.04

Table 4.3: Performance Criteria

S/N	T ₁ ^o C Ambient Temperature	Work Ratio	Thermal Discharge Index (TDI)	Net Plant Heat Rate (NHPR) MJ/KW	Net Work Ratio (NWR)	Thermal Efficiency (η_{th}) (%)
1	25	0.882	2.94	14.18	0.27	25.39
2	26	0.881	2.95	14.22	0.27	25.30
3	27	0.880	2.96	14.27	0.27	25.23
4	28	0.860	3.03	14.50	0.26	24.83
5	29	0.860	3.05	14.56	0.26	24.72
6	30	0.859	3.06	14.62	0.26	24.62
7	31	0.859	3.06	14.63	0.26	24.60
8	32	0.848	3.20	15.15	0.25	23.77
9	33	0.845	3.20	15.15	0.25	23.77
10	34	0.838	3.35	15.67	0.24	22.97
11	35	0.838	3.27	15.36	0.24	23.44
12	36	0.831	3.36	15.69	0.23	22.94
13	37	0.828	3.36	15.69	0.23	22.94
AVERAGE		0.855	3.13	14.90	0.25	24.19

Table 4.4: Parameters Compared to Percentage of Design

S/N	Ambient Temperature (^o C)	Percentage Of Power Output (%)	Power Differential (%)	Percentage Of Thermal Efficiency (η_{th}) (%)	Percentage of Specific Fuel Consumption(SFC) (%)	Percentage of Heat Rate(HR) (%)
1	25	44.56	55.44	95.45	96.75	139.08
2	26	44.56	55.44	95.11	97.40	111.90
3	27	44.52	55.48	94.85	97.40	139.78
4	28	44.52	55.48	93.35	99.03	142.25
5	29	44.52	55.48	92.93	99.35	142.96
6	30	44.52	55.48	92.56	100.00	142.31
7	31	44.48	55.52	92.48	100.00	142.31
8	32	44.44	55.56	89.36	103.57	148.25
9	33	44.40	55.60	89.36	103.57	148.25
10	34	44.36	55.64	86.35	107.14	153.55
11	35	44.32	55.68	88.12	104.54	150.72
12	36	44.28	55.72	86.24	107.44	153.90
13	37	44.16	55.84	86.24	107.44	153.90

4.2 Discussion

Figure 4.1 represents the effect of ambient temperature on the power output. It shows that the power output decreases as the ambient temperature increases. It also shows that as the ambient temperature increases from 29°C to 35°C, the power output decreases from 44.52% to 44.33%. But in real terms, the power output decreases from 4.96MW to 4.91MW.

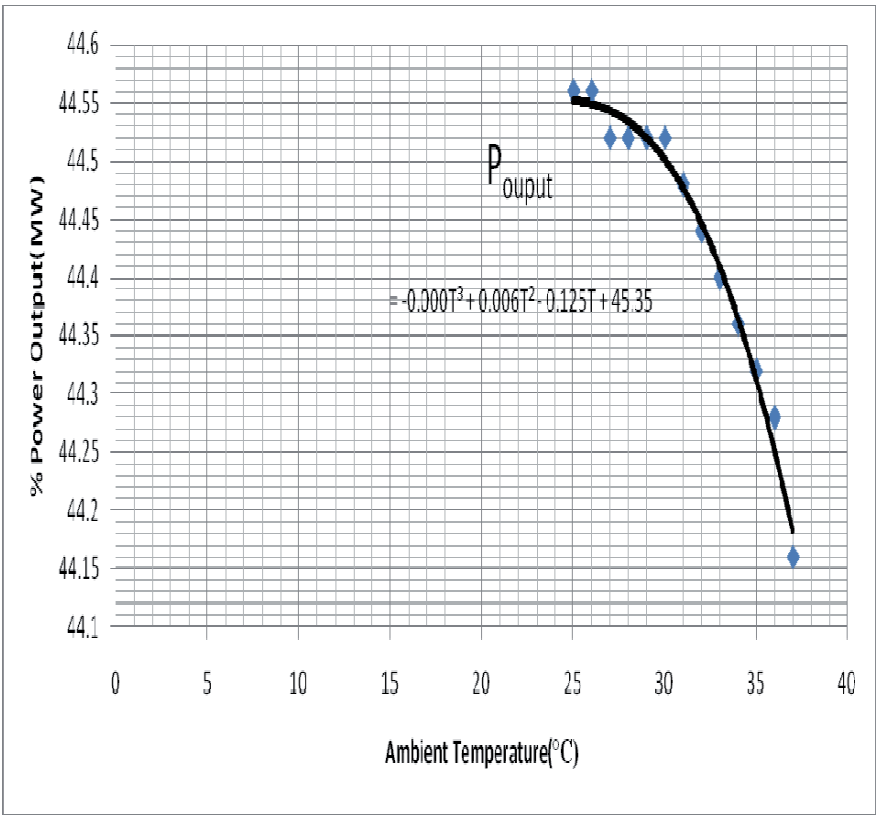


Figure 4.1: Effect of Ambient Temperature on the Power Output

Figure 4.2 represents the effect of ambient temperature on the power differential. It shows that the power differential increases as the ambient temperature increases. It also depicts that as the ambient temperature increases from 29°C to 35°C, the power differential increases from 58.44% to 58.79% and in real terms, the power drop increases from 6.17MW to 6.20MW.

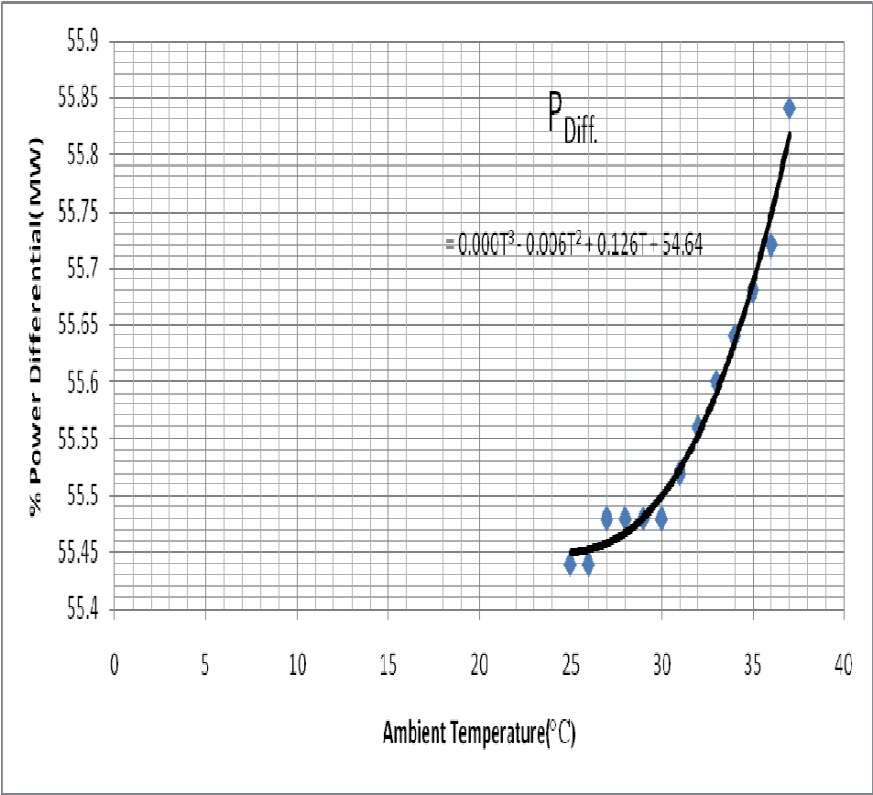


Figure 4.2: Effect of Ambient Temperature on the Power Differential

Figure 4.3 shows the effect of ambient temperature on the thermal efficiency ratio. It shows that the thermal efficiency ratio decreases as the ambient temperature increases. It also illustrates that as the ambient temperatures increase from 28°C to 33°C, the thermal efficiency ratio decreases from 94.60% to 88.60%. But in real terms, the thermal efficiency increases from 21.06% to 23.49%.

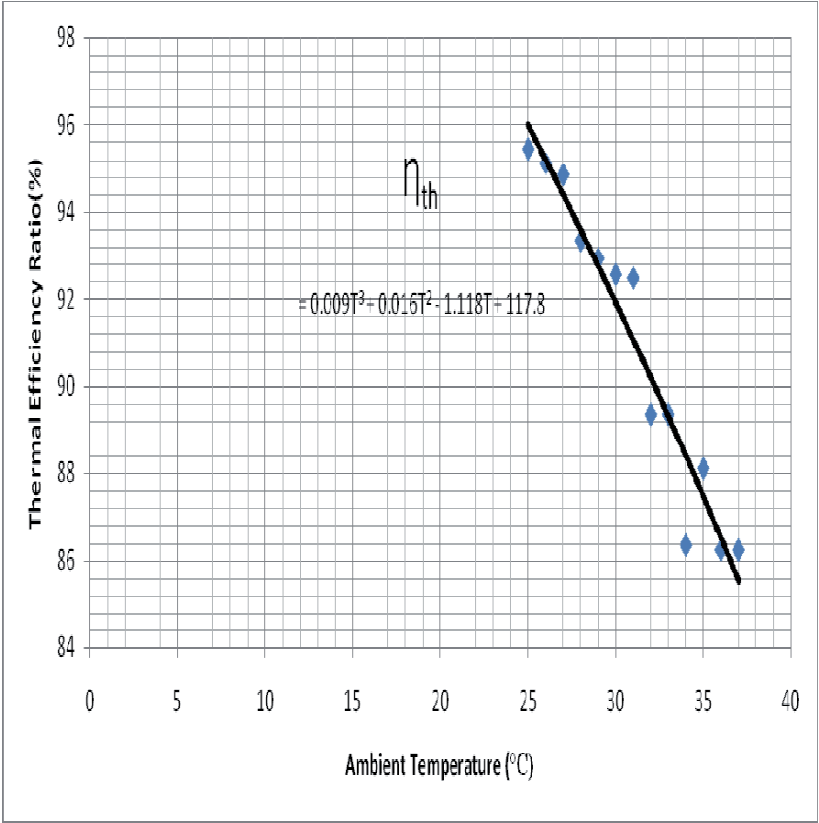


Figure 4.3: Effect of Ambient Temperature on the Thermal Efficiency Ratio

Figure 4.4 depicts the effect of ambient temperature on the specific fuel consumption. It shows that the specific fuel consumption increases as the ambient temperature increases. It also shows that as the ambient temperature increases from 30°C to 33°C, the specific fuel consumption increases from 98.60% to 102.80% and in real terms, the specific fuel consumption increases from 0.304kg/KWh to 0.328kg/KWh.

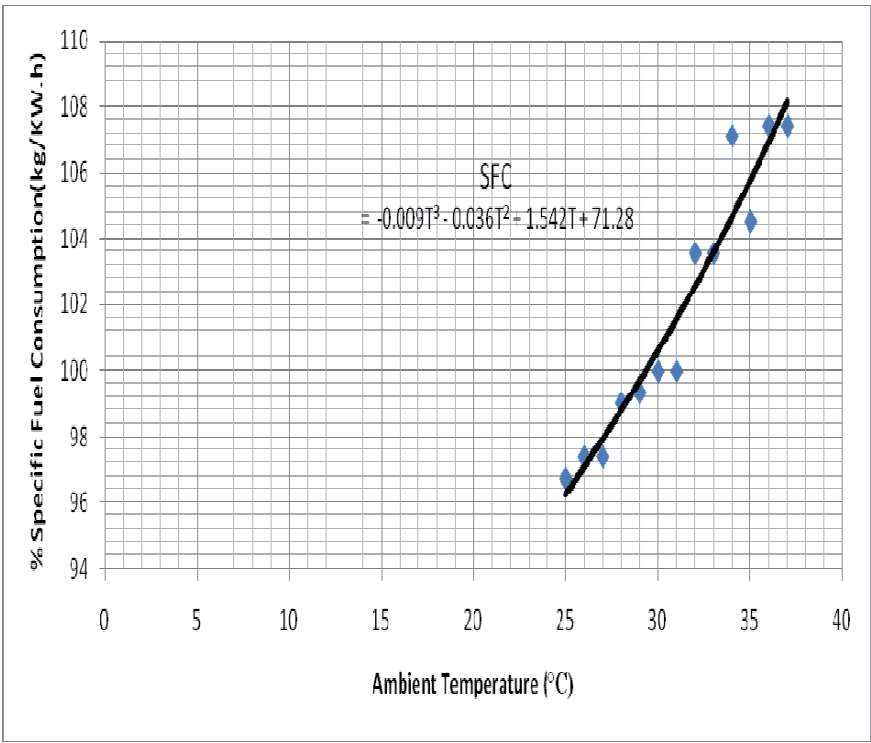


Figure 4.4: Effect of Ambient Temperature on the Specific Fuel Consumption

Figure 4.5 shows the effect of the ambient temperature: compressor inlet temperature on the heat rate. It shows that the heat rate increases as the ambient tem-

perature increases. The Figure 4.5 further shows that as the ambient temperature increases from 30°C to 37°C, the heat rate increases 144% to 156%. In actual analysis, the heat rate increases from, 5.85Kcal/W.h to 6.80Kcal/W.h.

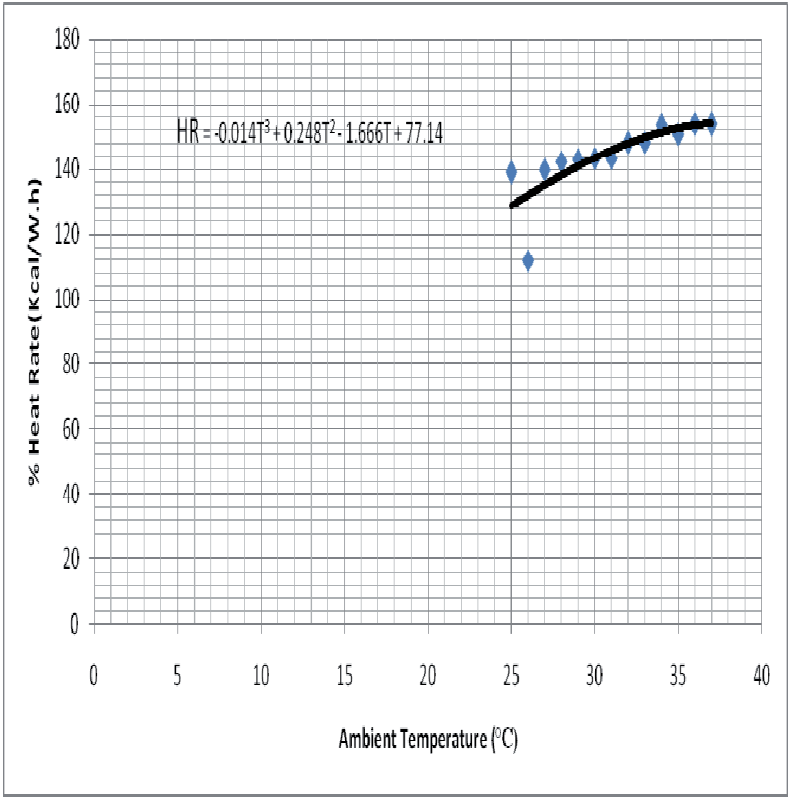


Figure 4.5: Effect of Ambient Temperature on the Heat Rate

Figure 4.6 represents the effect of the heat rate on the thermal efficiency. It shows that the thermal efficiency decreases as the heat rate increases. The Figure 4.6

also depicts in real terms that as the heat rate increases from 3.94Kcal/W.h to 4.36Kcal/W.h, the thermal efficiency decreases from 25.4% to 22.58%.

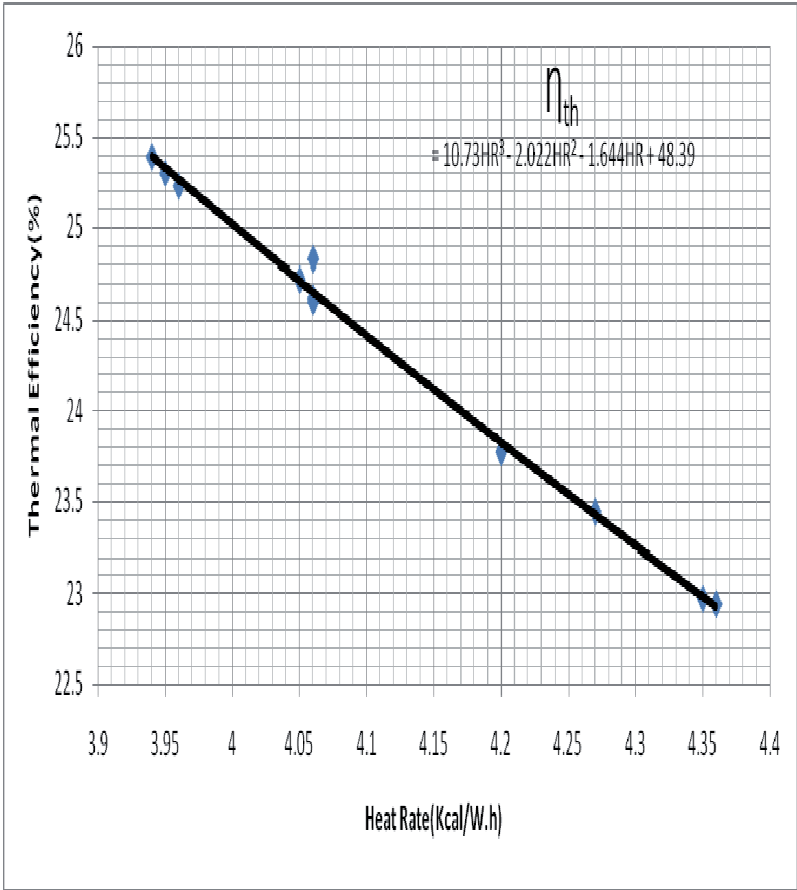


Figure 4.6: Effect of Heat Rate on the Thermal Efficiency

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

The results after the study of the various performance parameters and its effects on the Trans-Amadi Gas Turbine Power Station Phase II (GT-II) show that the ambient temperature affects the performance of the turbine.

Comparison between the percentage actual average power output generated by the turbine (Table 4.2:11.11MW) to the design power output (Table 4.1:25.0 MW) shows that 44.44% power generated and 55.56% power differential.

The average of the performance parameters like thermal efficiency (24.19%), work ratio (0.855), thermal discharge index (3.13), net plant heat rate (14.90MJ/KW) and net work ratio (0.25) were computed as shown and compared favourably with their respective design values (26.6% , 0.4-0.8,3.42, 10-20MJ/KW and 0.25) .

Furthermore, this study also shows that as the ambient temperature increases, the power output decreases, the power differential increases, the thermal efficiency ratio decreases, the specific fuel consumption increases and the heat rate increases. Also, for every increase in the heat rate, there is a decrease in the thermal efficiency.

5.2 Recommendations

The following recommendations are hereby projected:

- i. The thermal efficiency of the turbine needs an average ambient temperature of 29⁰C for more efficient performance and this ambient temperature can be achieved by the use of evaporative cooler.

ii. The specific fuel consumption needs an average ambient temperature of 29°C for minimal fuel consumption of the turbine and this ambient temperature is achievable using evaporative cooler.

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APPENDIX A

PROTOTYPE LOGSHEETS READINGS

Prototype of Turbine Logsheets Readings from the Months of January, 2012-2013.

MINISTRY OF POWER
GAS TURBINE POWER STATION
TRANS-AMADI

TURBINE LOGSHEET

07/01/12

GT-2

DATE 08/01/12

GT-2

	07.00	11.00	15.00	18.00	22.00	01.00	05.00	09.00	13.00	17.00	21.00	02.00	06.00	10.00	14.00	18.00	22.00	03.00	07.00	11.00	15.00	19.00	23.00	04.00	08.00	12.00	16.00	20.00
INLET TEMPERATURES (°C)	37.1	38.8	37.8	37.5	37.3	37.2	37.3	37.4	37.4	37.3	37.2	37.3	37.4	37.4	37.3	37.2	37.3	37.4	37.4	37.3	37.2	37.3	37.4	37.4	37.3	37.2	37.3	37.4
EXHAUST TEMPERATURES (°C)	310	302	317	310	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312
WHEELSPACE TEMPERATURES (°C)	112	384	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374	374
LUBE OIL TEMP (°C)	115	385	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
TURBINE BRG DRUM TEMPERATURES (°C)	114	385	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
VIBRATION LEVELS (mm/s)	114	385	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375

NAME	DAY	NIGHT
SIGN.		
DATE		

Prototype of Generator Logsheets Readings of the Months of January 2012 - 2013.

MINISTRY OF POWER GAS TURBINE POWER STATION TRANS-AMADI

		07/01/12												08/01/12											
		GT-2												GT-2											
		07.00	11.00	15.00	18.00	19.00	21.00	01.00	06.00	07.00	11.00	15.00	18.00	19.00	21.00	01.00	06.00	07.00	11.00	15.00	18.00	19.00	21.00	01.00	06.00
GENERATION	RV	11.23	11.23	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24	11.24
	NW	11.5	13.5	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0	11.0
GENERATION	NVAR	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5
	RZ	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5
PHASE	P-F	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9
	NW	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9
CURRENT	A	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	B	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
EXCITATION	C	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	W	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
WINDING	A	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	B	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
TEMP	C	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	D	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
BEARING	E	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	F	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
VIBRATION	DE	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	NOE	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
GLARBOX	DE	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	NOE	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
MONITORING	A	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	B	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
TEMP	C	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	D	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
GLARBOX	E	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	F	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
MONITORING	G	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	H	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
TEMP	I	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	J	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
GLARBOX	K	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	L	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
MONITORING	M	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
	N	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6

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