RIVERS STATE UNIVERSITY
PORT HARCOURT

ROTOR-BLADE PROFILE:
INFLUENCE ON THERMAL POWER AND ENERGY

AN INAUGURAL LECTURE

BY

PROFESSOR BARINAADAA THADDEUS LEBELE-ALAWA

B. Tech, M. Tech, Ph. D (RSU), MNSE, MNI MechE

Professor of Thermal Power And Energy Engineering

SERIES NO. 89

Wednesday, 28th Jun, 2023

DEDICATION

This Inaugural Lecture is dedicated to my Lord and Saviour Jesus Christ who saved me, called me, lifted me up, and made me a Professor.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF PLATES

PROTOCOL

The Vice Chancellor
Deputy Vice Chancellor (Administration)
Deputy Vice Chancellor (Academic)
Registrar and other principal officers of the University
Former Vice Chancellors and Emeritus Professors
Members of RSU Governing council here present
Heads of various campuses of the University
Provost College of Medical Sciences
Dean of the Postgraduate School
Deans of Faculties and Directors of Institutes and Centres
Heads of Departments and Units
Distinguished Professors and Members of Senate
Staff and Students of RSU
Ministers of God
Distinguished Colleagues
All members of my family, brethren, friends and Well wishers
Gentlemen of the press
Ladies and Gentlemen

1.0 PREAMBLE

It is a great privilege for me to stand here today to deliver the 89th inaugural lecture of this great University. I am indeed grateful to God who has watched over me as the apple of His eyes and ordered my steps to this enviable height in life. To Him be all glory forever. I therefore stand here today with a deep sense of joy and fulfilment to present my inaugural lecture.

I am very grateful to the Vice Chancellor and the Chairman of the Senate Lectures Committee Professor N.H. Ukoima and other committee members for making this possible. It is a day every professor looks forward to as it provides an opportunity to present to the world some key areas of the Professor's academic research and contributions to knowledge. For me, today's inaugural lecture represents a significant milestone in my life. Although this is my seventh year as a Professor of Thermal Power and Energy Engineering I could have delivered my inaugural lecture before today but I see this day as God's set time. This 89th Inaugural Lecture therefore makes me the 11th presenter in the great Faculty of Engineering and the 3rd in the Department of Mechanical Engineering since the inception of the University. I therefore consider it a great honour and privilege.

My inaugural lecture today is titled **"Rotor-blade profile:**

Influence on Thermal Power and Energy". Thermal power and energy is the bedrock of industrialization and human civilization. For instance, everyone in this place came either by road, air or sea. The car can move only with thermal power. The plane cannot take off without thermal power, neither can the ship move without thermal power. In fact, the world will be in darkness in the absence of thermal power as there may be near absence of electricity generation. What would have happened in this hall today if a cooler powered by a compressor was not placed between two thermal reservoirs. As I deliver this lecture I want to crave the indulgence of this audience to bear with me on the technical content of this presentation because it might be quite difficult to explain some very important aspects of my research without recourse to some technical terms.

Vice Chancellor Sir, distinguished ladies and gentlemen, this inaugural lecture will highlight the phenomenon of the axial compressor (its rotor blades) influences on the availability and utilization of thermal power and energy. Engineering solutions, redesigns and recommendations are also part of this presentation.

2.0 INTRODUCTION

Energy forms the basis of human life. It is the capacity to do work. There is hardly any activity that is independent of energy. Energy is fundamental to the development and advancement of any human society. It affects the standard of living and industrialization of any country. Early man used muscle power, then fire, and animal power. Later, man learnt to harness energy, convert it to useful form and put it to various use

There are many forms of energy:

(i) Chemical Energy: Energy stored in the bonds of atoms and molecules. It is potential in nature. Natural gas, gasoline, coal and biomass are examples of stored chemical energy.

(ii) Nuclear Energy: Energy stored in the nucleus of an atom. The nucleus of a uranium atom is an example of nuclear energy. The nucleus consists of protons (having positive charge) and neutrons (having negative charge). Nuclear energy exists only as stored energy. This type of energy is produced in nuclear power plants.

(iii) Mechanical Energy: Energy stored in objects by tension. It is manifested when energy results from bulk movement of a system. Mechanical energy in transition is known as work.

(iv) Thermal Energy: Thermal Energy (or heat) results from vibrations and movements of atoms and molecules within substances.

(v) Electrical Energy: Energy that comes from the movement of tiny charged particles called electrons.

(vi) Electromagnetic Energy: The term is used to describe the various energies that travel as wavelengths through space.

2.1 Thermal Power and Energy

Thermal Engineering is a specialized sub-discipline of Mechanical Engineering that deals with the movement of heat energy and transfer. The energy can be transferred between two mediums or transferred into other forms of energy. It involves the knowledge of thermodynamics and the process to convert generated energy from thermal sources into Chemical, Mechanical or Electrical energy.

Industrial thermal systems include automotive engines, jet engines, rockets, power plants, refrigeration and air-conditioning systems.Thermal power plants cover thermodynamic energy conversion systems(coal-fired power plants, gas turbines and steam power plants, combined gas and steam turbine plants as well as nuclear power plants).

Thermal energy is also obtained from renewable energy sources for immediate use or for storage for later use. The most popular forms of thermal renewable energies are:

Solar Energy: Solar energy is derived from the sun through the form of solar radiation. Solar powered electrical generation relies on photovoltaics and heat engines. The solar photovoltaic system provides a direct conversion of solar radiation to electrical energy using semiconductor materials, which is a very reliable power source. Photovoltaic cell is a solar cell, which is solid state electrical device that converts energy of light directly into electricity. Assemblies of these cells are known as solar modules or solar panels.

Geothermal Energy: Energy stored in the ground in the form of heat which can be extracted and utilized. Geothermal energy is the heat originating from the original formation of the planet, from radioactive decay of minerals, from the volcanic activity and from solar energy absorbed at the surface. The geothermal gradient, which is the difference in temperature between the core of the planet and its surface, drives a continuous conduction of thermal energy in the form of heat from the core to the surface.

Ocean Thermal Energy: Energy is obtained by harnessing the temperature differences (thermal gradients) between ocean surface waters and deep ocean waters. In tropical regions, when the sun heats the ocean surface water, it becomes warmer than the deep water. This temperature difference can be used to produce electricity. Ocean thermal energy conversion systems use a temperature difference (of at least 20^{o}C) to power a turbine to produce electricity. Warm surface water is pumped through an evaporator containing a working fluid. The vaporized fluid drives a turbine or generator. The vaporized fluid is turned back to a liquid in a condenser cooled with cold ocean water pumped from deeper depth of the ocean.

2.2 Production of thermal power from a heat engine

The production of mechanical work from the combustion of fuel is generally accomplished by a thermodynamic cycle involving the processes of induction, compression, heat addition, expansion, and heat rejection/exhaust. This is typical of a gas turbine plant (figure1).

Air is drawn in through the air intake system into the compressor from the atmosphere. The compressor discussed here is the axial flow type. The axial flow compressor compresses the working fluid (air) by first accelerating the fluid and then diffusing it to obtain a pressure increase. The fluid is accelerated by a row of rotating airfoils (blades) called the rotor and then diffused in a row of stationary blades (the stator). The diffusion in the stator converts the velocity increase gained in the rotor to a pressure increase. A compressor consists of several stages. One rotor and one stator make up a stage in a compressor. Importantly, the changes in the total conditions of pressure, temperature and enthalpy occur mainly in the rotating component where energy is imputed into the system. Inlet guide vanes are frequently used at the compressor inlet to ensure that air enters the first stage rotors at the desired angle. At the exit of the compressor is a diffuser that diffuses the fluid and controls the velocity entering the combustors. The compressed air then mixes with the gas (fuel) and is burnt in the combustion chamber. The products of this combustion are allowed to expand and the mechanical work produced drives the turbine which in turn drives the compressor.

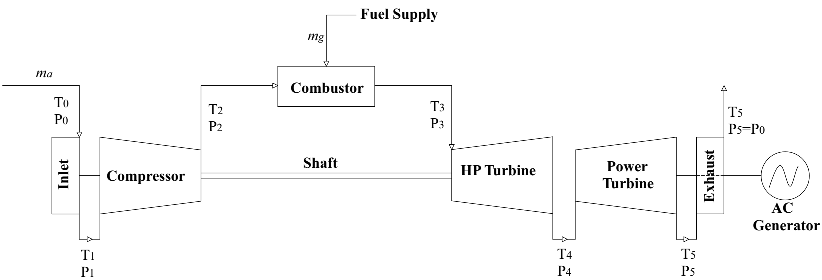


Figure 1: Schematic Diagram of a Simple Gas Turbine Plant

Of all these processes, compression is the most difficult as fluid admitted at a low pressure region is compressed and delivered to a higher pressure region. In order for the gas turbine to produce an expansion, a pressure ratio must be provided and the first necessary step in the cycle must therefore be compression of the working fluid. If after compression the working fluids were to be expanded directly in the turbine, and there were no losses in either component, the power developed by the turbine would just equal that absorbed by the compressor. But the power developed by the turbine is increased by heat addition to raise the temperature of the working fluid before expansion. When the working fluid is air, a very suitable means of doing this is by combustion of fuel in air which has been compressed.

The simple cycle gas turbine can be classified into five broad groups (Boyce, 2002):

Group1: Frame type heavy-duty gas turbines. The frame units are the large power generation units ranging from 3MW to 480MW in a simple cycle configuration, with efficiencies ranging from 30-46%.

Group2: Aircraft-Derivative gas turbines (Aeroderivative). As the name implies, these are power generation units which have origin in the aerospace industry as the prime mover of aircraft. These units range in power from 2.5MW to about 50MW and their efficiencies range from 35-45%.

Group3: Industrial type gas turbines. This type of turbine is used extensively in many petrochemical plants for compressor drive trains and other industrial applications.

Group4: Small gas turbines. These gas turbines are in the range from about 0.5MW to 2,5MW. Efficiencies in simple cycle application range from 15-25%.

Group5: Micro-turbines. These turbines are in the range from 20KW-350KW.

2.3 The Compressor

Gas turbine engines are designed to be highly efficient, producing the required power output while maintaining high thrust. Central to achieving this goal is the gas turbine compressor component which consists of blades in rows referred to as cascades.

The primary purpose of the compressor is to raise the pressure of the inlet air. The compressor must increase the pressure of the mass of air received from the inlet duct and then discharge it at higher (required) pressure to the combustor (Cohen, *et al*). Sustenance of the design shape is fundamental to the compressor performance as it is quite sensitive to surface roughening and profile deterioration. For instance, a change in blade profile resulting in a drop of compressor mass flow of about 3 percent can result in a reduction in output of 6 percent and an increase in fuel consumption of 1.7 percent.

Let us consider two compressor stages with inlet guide vane (IGV) as shown in figure 2.

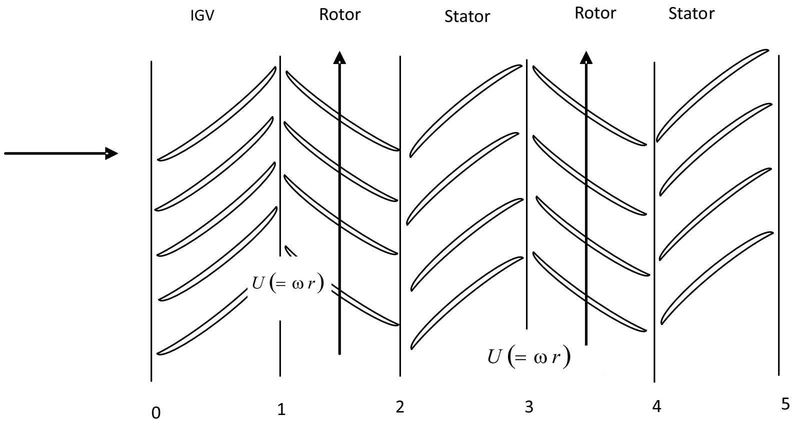
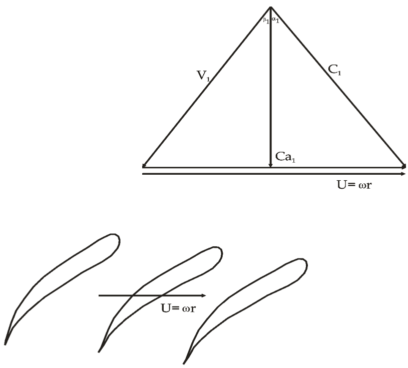


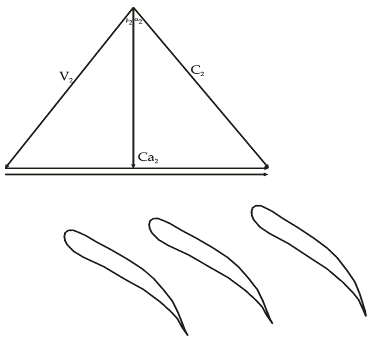
Figure 2: Two Compressor Stages with Inlet Guide Vane

The flow in the rotor and stator passages of a compressor is diffusing whereas it is accelerating in a turbine. The air flows axially through the moving and fixed blades in turns. The compressor blades are arranged so that the space between them forms diffuser passages and hence the velocity of air relative to the blades is decreased as the air passes through them and there is an increase in pressure. It is always more difficult to arrange for an efficient deceleration of flow than for an efficient acceleration. In a diffusion process there is a natural tendency for the fluid to break away from the wall of the diverging passage, reverse its direction, and flow back in the direction of the pressure gradient. The flow enters the rotor with velocity C_1 (relative Velocity V_1) and leaves with velocity C_2 (relative

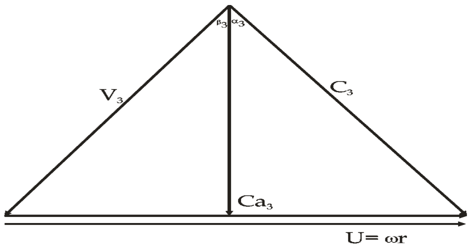
velocity V_2). The rotor is moving upward at velocity $U(=\omega r)$. The flow enters the stator with velocity C_2 and leaves with Velocity C_3. The flow enters the second stage rotor with a velocity C_3(relative velocity V_3) and leaves with velocity C_4. The second stage rotor is moving at a velocity $U(=\omega r)$ and the flow from the rotor enters the stator with velocity $C_4$and leave
With velocity C_5. Figure 3 shows the velocity diagrams.



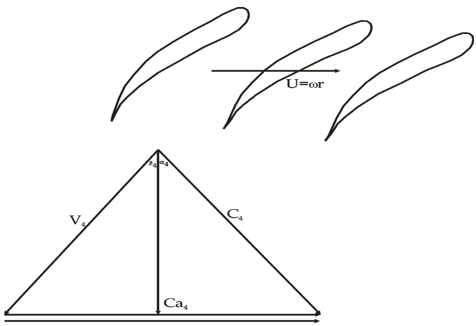
Station (1)
1st stage rotor



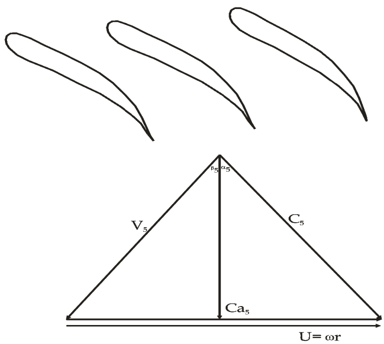
Station (2)
Exit from 1st stage rotor and entry to 1st stage stator.



Station (3)
Entry to second stage rotor



Station (4)



Station (5)
Exit from second stage rotor

Figure 3:Velocity Diagrams for First and Second Stages of a Compressor

Also, considering the change in angular momentum of air in passing through the rotor, the power input can be expressed in terms of the rotor blade air angles β_1 and β_2 as shown in figure4. This is important because a lot will be said in this lecture about the effects of the rotor-blade angle deviations on the thermal power availability and also the economic implications.

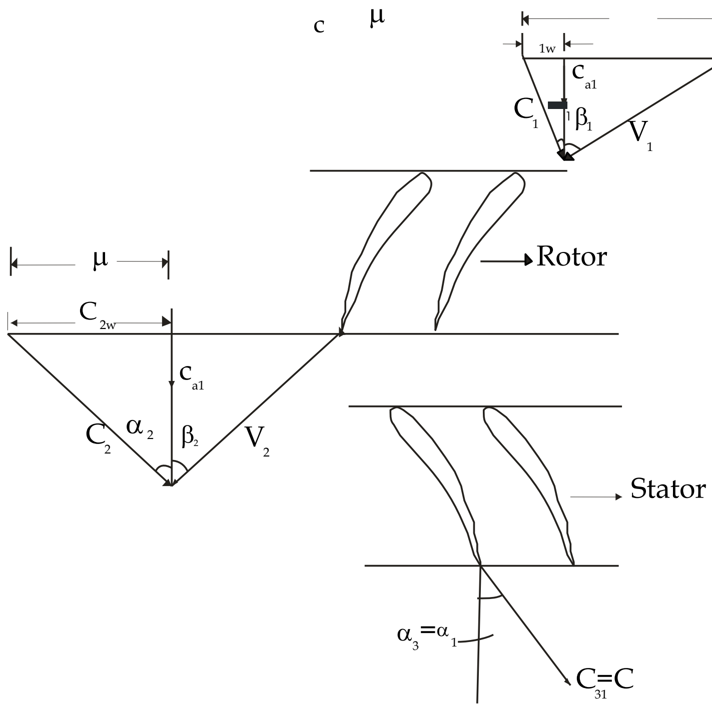


Figure 4: Axial flow compressor stage in terms of rotor blade air angles

where C_1 = Air inlet velocity,
C_2 = Air outlet velocity,
α_1 = Air inlet angle,
α_2 = Air outlet angle
V_1 = Air velocity relative to blade; $\acute{\alpha}_3 = \acute{\alpha}_1$
U = Blade speed
β_1 = Rotor blade inlet angle
β_2 = Rotor blade outlet angle

$$W = mUC_a(\tan\beta_1 - \tan\beta_2) \quad (1)$$

where W = The power input and $C_a(\tan\beta_1 - \tan\beta_2) = \Delta c_w$

The pressure ratio across stage is given by:

$$R_c = \frac{P_{03}}{P_{01}} = \left[1 + \frac{\eta_S \Delta T_{0S}}{T_{01}}\right]^{r/(r-1)} \quad (2)$$

Where $$\Delta T_{0S} = T_{03} - T_{01} = \frac{UC_a}{C_p}(\beta_1 - \tan\beta_2) \quad (3)$$

And $$\eta_S = \left(T_{03}^1 - T_{01}\right)/\left(T_{03} - T_{01}\right)$$

Compressor blade profile change leads to blunting of the air foil leading edges, it changes the air incidence angles and leads to the thinning of the blade trailing edges. The effect of these changes is a significant deterioration in performance, eventually leading to the requirement that the compressor be re-bladed. In addition, thinning of the trailing edges is always detrimental to the fatigue strength and is therefore most undesirable. All these point to the delicate nature of the compressor and its sensitivity to the slightest distortion of the designed blade shape. Thus, compressor blade profile change has a significant effect on the thermodynamic performance of the gas turbine plant.

3.0 THE ROTOR BLADE PROFILE

The performance of the compressor depends to a great extent on the blade profiles. The basic function of the blades is to turn the air to the required angle. Along this process, undesired losses are recorded. Therefore, the goal of any blade design is to achieve the desired flow turning with minimum losses, within the constraint of geometric orientation of the blade row. Compressor and turbine blades are designed and shaped to create very precise aerodynamic flow patterns and any change in profile may result in the air moving through paths different from what was designed which affects the performance.

The design specification of the compressor blade is of major importance in the overall compressor design. The axial compressor blade is an airfoil shape and is designed for compression of air efficiently at a high tip blade speed. To achieve this blade design objective, a blade to blade surface calculation is used to define the blade shape and is a way to describe the flow unlike in the past where correlations from

series of standard profile were used (Aziaka *et al*, 2014).
The blade must be designed not to fail due to gross over stressing or high cycle fatigue, hence a dovetail which is precise in size and position are used to keep the blades in their desired position and locations on the wheel. Both the stator and rotor blade is mounted in similar dovetail arrangement. The rotor blades can be seen in figures 5 and 6.

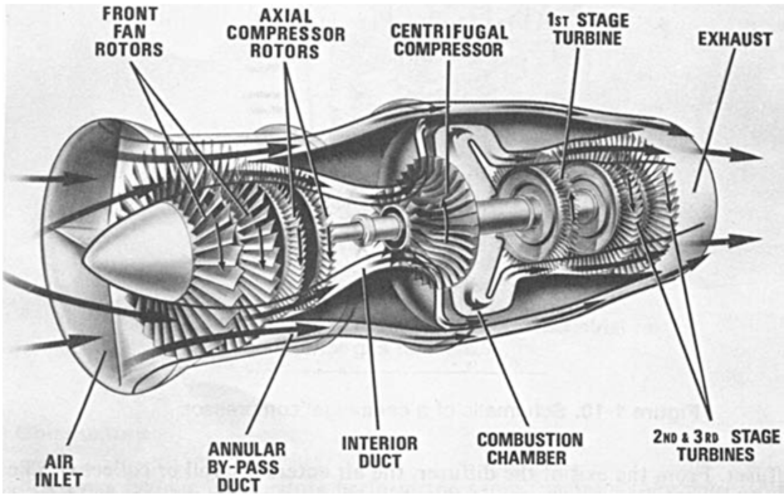


Figure 5: A Schematic of a Cutaway of a Small Gas Turbine for Helicopter or Vehicular Applications (Source: Boyce, 2002)

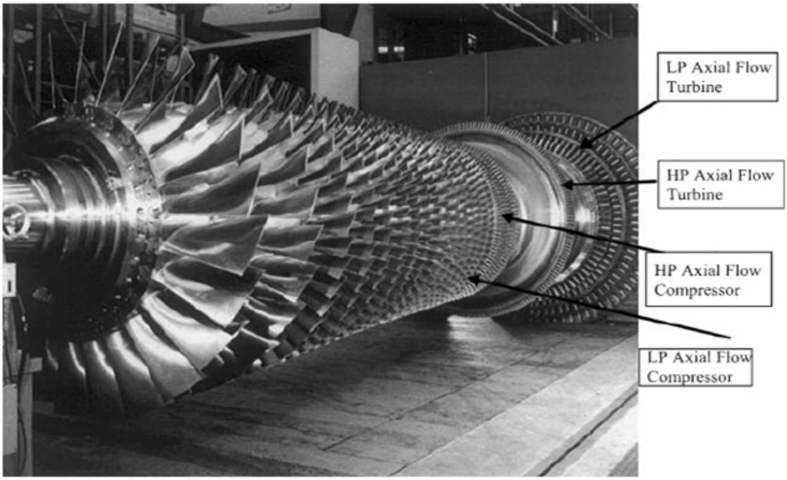


Figure 6: Diagram Showing a High-Pressure Ratio Gas Turbine Rotor Blades(Source: Boyce, 2002)

The aerodynamic performance of a compressor cascade is basically determined by the shape of the blades, Cascade stagger angle (φ), Cascade solidity $\left(\sigma = c/s\right)$ Inlet flow angle (β_1)Inlet Mach number (M_1).The aerodynamic blade profile design gives the stagger, solidity, camber angle, camber line shape (chord ratio) and thickness distribution. The chord line is the line passing through the trailing and leading edges while the camber line is curved and runs through the middle of the blade profile. Having obtained the blade radius, inlet and outlet angles and the nominal deflections, the next step is to determine the chord length which depends on the pitch and the number of blades in the row. During the choice of the number of blades the aspect ratio is considered because of its effect on secondary losses. The number of blades to be chosen is usually odd number to reduce the impact of vibration frequency. Lashing wires are also used on the rotor blades to solve blade vibration stress problems. The basic resonance and direct excitation source should be considered using stator or splitter and guide vanes to pass the frequencies (Hanlon, 2001).

Figure 7 and 8 shows comparison on thickness distribution and pressure distribution among the blade profiles.

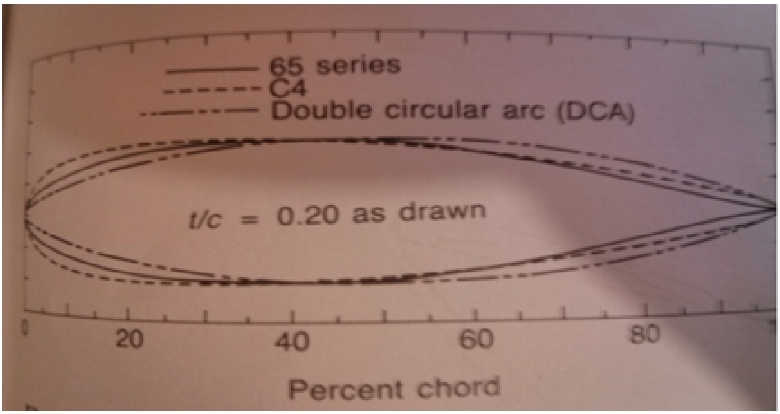


Figure 7: Comparison of Thickness Distribution on Three Common Profiles (Royce,1997)

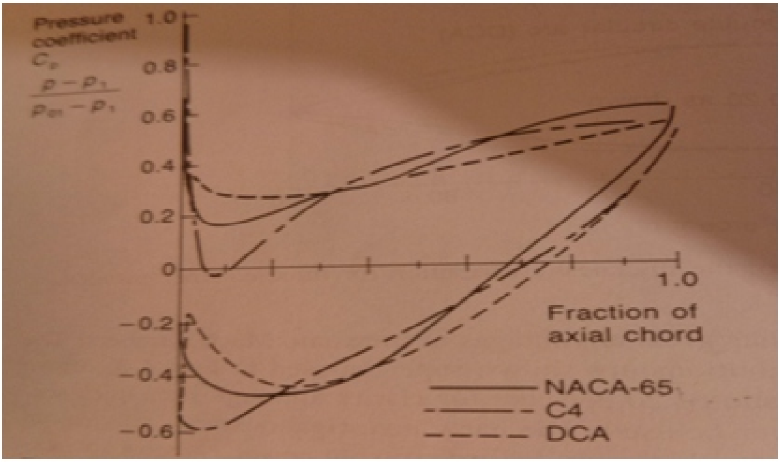


Figure 8: Pressure Distribution on the Three Blade Profile (Royce,1997)

While it is the design goal to generate blades which meet the geometrical and flow turning requirements of the compressor, of equal importance is how much losses are generated due to profile changes and the corresponding effect on the Isentropic efficiency. Unlike a turbine cascade through which the flow is accelerated, the adverse pressure gradient due to the flow diffusion in a compressor cascade (which is a consequence of the profile distortion) imposes unfavourable force on the boundary layer. Correspondingly, it is more difficult to achieve a thinner boundary layer and control the boundary layer from separation, which is the major factor determining the aerodynamic losses of the blades (Song, 2003). A particular area of interest in the behaviour of flow distortion is the manner in which it propagates through a blade row. As a distortion pattern exits a stage, it serves as a driving force for the next blade row or engine component immediately following. The cumulative result of these distortions is a drop in the isentropic efficiency of the gas turbine plant. Also for a radial clearance of 2mm due to compressor blade profile distortion, the efficiency of the gas turbine would decrease by about 0.9 percent, and the resultant loss of power output from an 80MW gas turbine would be 0.73 mW. This would mean an annual loss of 3.6 GWh of electrical energy based on 5000 full load operating per day.

In addition to these, flow distortions present periodic blade passing frequencies to rotor blades, which are capable of exciting these blade vibratory modes. Even with modern calculation methods, the alternating stress response of blades to potential blade vibration excitation cannot be reliably determined (Vahdati *et al*, 2000). The mechanical vibration of the machine is always monitored and the machine is tripped if the vibration exceeds a certain level. A much higher vibration response occurs in rotating blade rows when they move through a non-uniform flow field and when they are excited by alternating bending forces. Such non-uniformities can be generated by flow distortions, by trailing-edge wakes of struts in the compressor or inlet casing etc. The most intensive vibration response occurs if the flow distortions are equally spaced over the circumference, thus forming harmonic multiples of the rotational speed, and if these harmonics coincide with the natural frequencies of the rotating blades. Blade passing frequencies combining with the relative rotation of blades can lead to a condition known as high cycle fatigue (HCF), which is a primary mechanism of blade failure caused by vibrations at levels exceeding material endurance limits. The issues discussed above make the problem of compressor blade profile change a major source of concern as it ultimately affects the gas turbine thermodynamic performance and reliability.

3.1 Physical inspection/ Observations

Physical inspection of some first stage compressor blades which has been in operation at Ughelli GT 17 for a period of 27 months were made during my research when the sets were opened for maintenance. The presence of contaminants was very prominent on the compressor blades and in some cases the blade profile has been distorted (plates 1, 2, and 3). It was obvious that the build-up of particles on the compressor blades would result in the thickening of the boundary layer airstream. This results in the engine operating at a reduced pressure ratio which in turn reduces the compressor isentropic efficiency and the entire plant performance.

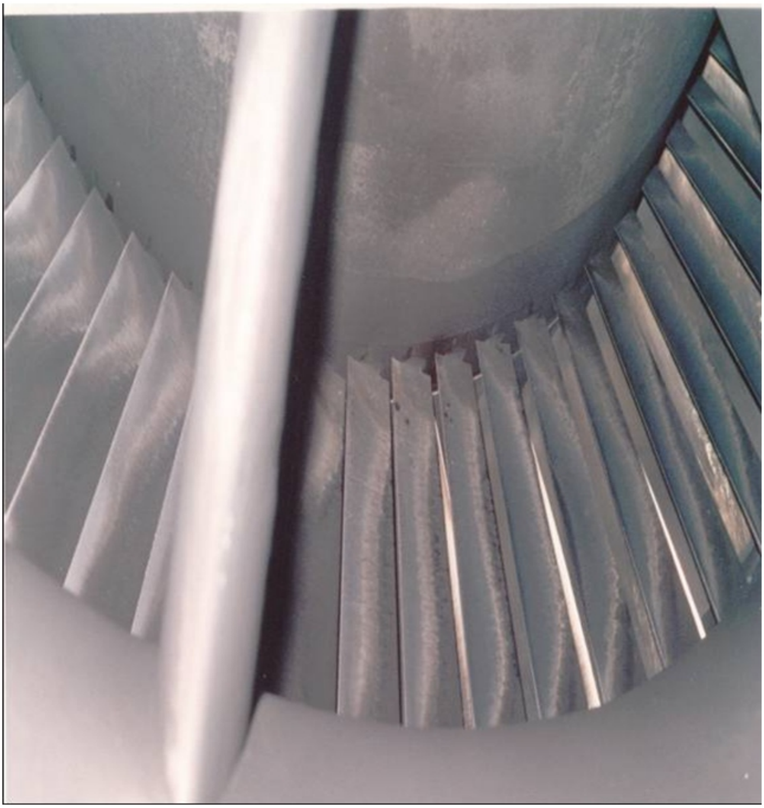


Plate 1: First Stage Compressor Blades of Ughelli GT 17

Plate 2:A Second View of Contaminated Compressor Blades

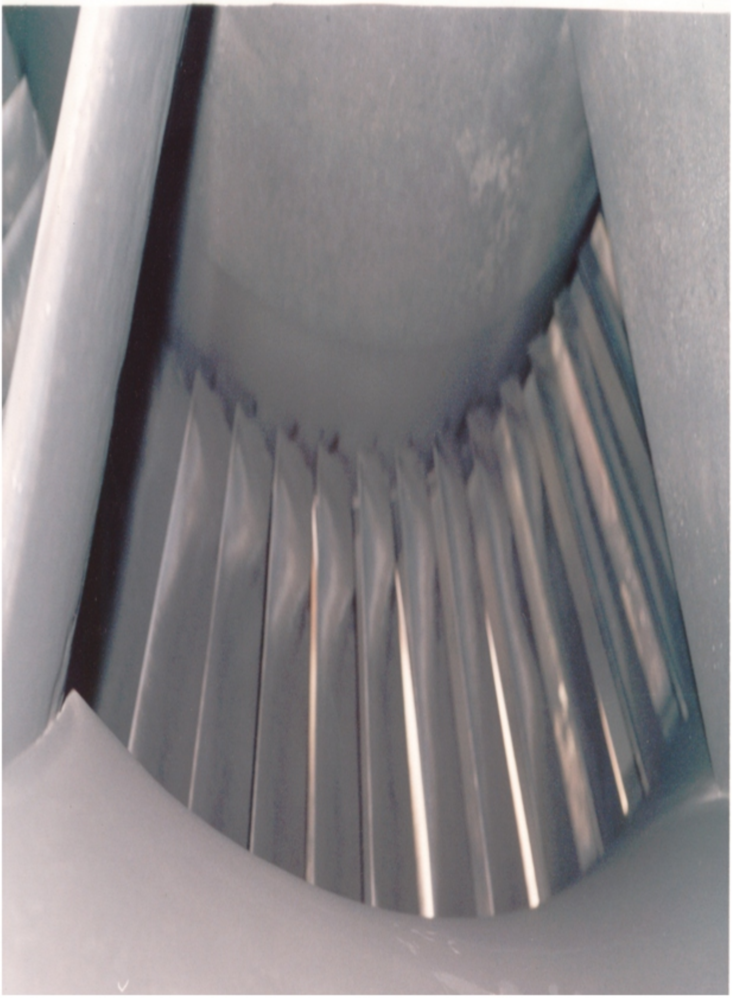


Plate 3: A Third View of Contaminated Compressor Blades at Ughelli G.T.17

3.2 Causes of rotor blade profile change

3.2.1 Fouling and Aging

Gas turbine engines consume large quantities of air. Due to its inherent design and the large volume of air consumed, they are very sensitive to air quality. Despite high quality filtration, some substances can still enter the compressor. Even talc-like sub – micronic particles can enter, adhere to the rotor blades by molecular attraction and build up to a significant extent. Fouling is caused by particles having diameters less than 5 microns, and dust is a major source of fouling. Compressor fouling is unavoidable in more or less all conditions. Its intensity is subject to ambient conditions for the particular area. In general, fouling due to ingesting of contaminated air may cause the deposition of contaminants which may cause blade profile deterioration. Fouling is a serious problem, particularly in the oil and gas industry where sticky hydrocarbon aerosols are universally present. The rate at which this fouling takes place is difficult to quantify because it depends not only on the types and quantities of materials ingested, but also on the peculiar properties of the substances that cause them to stick (Loud and Slaterpryce, 2000). These particles build up on the downstream side of the rotor blades, causing a reduction in the efficiency of the turbine system as well as increasing the turbine's heat rate.

Lebele-Alawa (2010a) investigated the metal concentrations in the dust particles causing fouling, corrosion and other forms of degradation on the first stage of compressor blades at Ughelli GT 17.The concentration of metal elements in the dust particles accompanying gas-turbine inlet air was investigated. Site parameters such as average wind speed and the relative humidity of the air were measured. The dust in the gas-turbine inlet air was trapped using a filter paper of 0.8micron installed in a gravimetric dust sampler. Experimental procedures of ashing, digestion, and analysis using atomic absorption spectroscopy (AAS) was done. The average results obtained showed the presence of dust and inherent metal elements on the rotor blades. The presence of these metal elements contribute to increased heat load on the system and reduction in the overall efficiency of the gas-turbine plant.

The work showed that despite the existing filtration processes on the gas turbine plants

operating in the Niger delta area of Nigeria, some dust and other particles still accompany the inlet air into the system. The dust contain proportions of metallic particles such as lead(Pb), Iron(Fe), Nickel(Ni), copper(Cu) and zinc(Zn). These bring about an increased heat load on the system. The contaminants also change the blade profile resulting in the reduction of air mass flow rate and pressure ratio culminating into a reduction in the effectiveness of the gas-turbine plant. It is therefore important that site locations and environmental conditions which dictate air-borne contaminants, their sizes, concentrations and composition, need to be considered in the selection of the appropriate air filtration process.

The deposits result in thickening of the boundary layer air stream. The thicker boundary layer results in a reduced mass flow through the engine. The most obvious sign of compressor fouling is a loss in the compressor discharge pressure. Aging causes surface roughness. This is capable of causing a drop in compressor efficiency.

3.2.2 Tip Rubbing

Tip rubbing always abrades materials from the contacting surface and results in increased radial clearance. In the case of heavy rubbing, thermally induced cracks can occur in the outer shroud of the stationary blades as well as in the moving blade tip. The braded material can also block the exits of the radial cooling holes which could result in overheating the blade material with serious consequences. There is also the possibility of low-cycle fatigue cracking in the root section which could also be caused by tip rubbing and consequential blade bending. If tip rubbing occurs, even without any mechanical damage to the parts, it has an effect on the thermodynamic performance of the plant.

3.2.3 Erosion of the blades.

Even though the inlet air of nearly all modern gas turbines is filtered by air filters and the Compressor blades themselves are coated, erosion will eventually occur. Erosion is understood to be abrasive wear of metal by fine solid particles in the working fluid. Erosion of the rotor blade is generally attributed to air borne particles with aerodynamic diameters exceeding 5 microns. Both the axial compressor and the hot gas parts can be affected by erosion from hard abrasive particles such as sand or mineral dusts. As these particles impact upon the compressor blades, they cut away a small amount of metal. The net rate of erosion, although not precisely quantifiable, depends on the kinetic energy change as the particles impinge, the number of particles impinging per unit time, the angle of impingement and on the mechanical properties of both the particles and the material being eroded. Erosion pits act as stress raisers and are therefore initiators of cracks which could ultimately cause blade failure. If this occurs in the first stage, blades in the following stage can be damaged by the ensuing debris. Simultaneously, change of profile can alter the resonance frequencies of the blades which can also lead to failure.

3.2.4 Thermal Fatigue Crack

Due to higher inertia of the thick sections, the thin sections of the inner shroud have to withstand straining in accordance with the temperature difference. Therefore, the thermal fatigue cracks will always start at the thin sections of a component. Thermal stresses are usually much higher than mechanically induced stresses and consequently can force the material to yield into its plastic range. Due to the high strain rates, thermal fatigue cracks are inter-granular and mostly split open.

3.2.5 Corrosion

This occurs where particles, chemicals or vapours in the air react with a compressor or turbine blade's metal properties and cause damage to the blades. Oxidation and all other reactions of metals with gaseous environments have long been recognized as severe limitations to the utilization of metal at high temperatures. Oxidation leads to premature failure. Grain boundary oxidation may produce a notch effect that also can limit life. All these are harmful in gas turbines.

One of the important and frequently encountered consequences of inadequate air filtration has to do with the ingestion of certain metals like iron, copper, zinc, etc, through dust. During the dry season in the Niger Delta area of Nigeria, there are considerable dust deposits of up to 100 mg/m^3. Between December and February which is referred to as harmattan period, dust concentrations may be as high as 20,000 mg/m^3 especially during dust storms. Dust sizes vary from fine to coarse and depending on the particular location. In desert climates the air is loaded with abrasive dust (Silica) and dust storms can be a particular challenge.

Salt containing atmospheres exert a deleterious effect on other materials besides steels and high temperature alloys. Hot corrosion is characterized by swelling together with splitting and flaking along the leading and trailing edges of the airfoil. For example, hot salt corrosion can be induced in titanium alloy kept in contact with salt during high-stress exposure at elevated temperatures. Hot corrosion of nickel – based and cobalt based alloys is accelerated oxidation caused by the presence of sodium chloride and sodium sulphate, usually from operating near sea water.

4.0 INFLUENCE OF ROTOR BLADE PROFILE CHANGE ON PERFORMANCE PARAMETERS

4.1 Pressure ratio across the compressor

The development trend of modern axial flow compressors is marked by higher pressure ratios. The compressor pressure ratio is the ratio of air total pressure exiting the compressor to the air pressure entering the compressor. A study was carried out on the effects of rotor blade profile change on the pressure ratio and the consequence on the compressor effectiveness (Lebele-Alawa *et al*, 2008)

$$W_c = H_2 - H_1 = C_{p\,avg} T_1 \left[(P_2/P_1)^{(\Upsilon-1)/\Upsilon} - 1 \right] \qquad (4)$$

It follows that anything that affects the pressure ratio will affect the compressor work required and the subsequent thermal power availability.

The decreasing pressure ratios resulting from blade profile change, as seen in this study, is therefore of significant negative consequence on the gas turbine plant as it imposes unfavourable force on the boundary layer and corresponding drop in the isentropic efficiency of the plant (Figure 9).

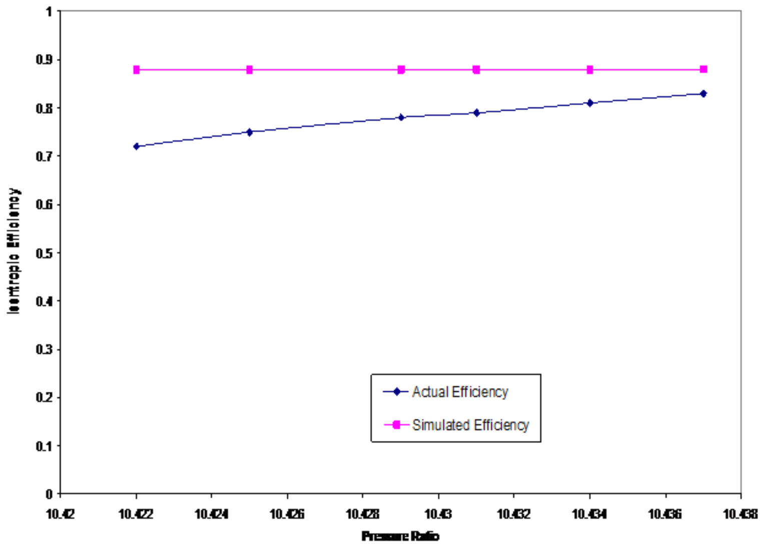


Figure 9: Variation of Compressor Isentropic Efficiency with Pressure Ratio.

4.2 The change in mass flow rate across the compressor

Fouling which is caused by the adherence of particles less than 10μm (such as salt, dust, ash, smoke, carbon, mist) etc to the airfoils leads to changes in the blade shape, surface roughness and blade inlet angles. Under normal conditions the inlet system has the capability to process the air by removing the contaminants to levels below those that are harmful to the compressor and turbine. Filtration systems, however, are not one hundred percent effective. Thus, some of these foreign particles enter and adhere to the compressor blades by molecular attraction and build up to a significant extent. These deposits build up on the downstream of the rotor-blades resulting in thickening of the boundary layer air system. The thicker boundary layer results in a reduced mass flow through the system. Thus, the rotor-blade fouling reduces the compressor mass flow rate, the pressure ratio and the cycle efficiency which subsequently results in the reduction of the available thermal power.

4.3 The axial thrust.

In a gas-turbine, atmospheric air is drawn in through an intake duct into the compressor and delivered at a higher pressure to the combustor. This is accomplished by the gas-turbine compressor, which consist of a cascade of several stages of blades, designed radially on a single axle. Maintenance of each blade's design-shape is fundamental to sustaining the compressor's high performance: distortions of each blade's profile lead to changes in the air incidence angle of the air flow on the blade.

Flow distortions can result in the vibration of the rotor blades. A particular concern is the manner in which the flow distortions propagate through each blade row. The distorted flow exiting one stage serves as the driving force for the subsequent blade row (Luedke, 2001). The cumulative effect of these distortions is a progressive increase of the axial thrust on the compressor, which can result in serious vibrations. These occur in the rotating blade rows when they move through a non-uniform flow field and when they are excited by alternating bending forces. Such non-uniformities can be generated by flow distortions and trailing-edge vortices from its struts in the compressor (Lamb, 2005). The most intensive vibration response occurs if the flow distortions are equally spaced over the rotating blades. This forms harmonic multiples of the rotational speed, especially if these harmonics coincide with the natural frequencies of the rotating blades. Blade-passing frequencies, combined with the rotation of blades, can lead to a condition known as high-cycle fatigue(HCF), which is a primary mechanism leading to blade failure caused by vibrations at levels exceeding the blade's material-endurance limit. The issues discussed above make the problem of compressor-blade profile-distortions a major source of concern, as it adversely affects the gas-turbine's thermodynamic performance, reliability and life duration.

Lebele-Alawa (2010b) The showed in his study that as the rotor-blade's inlet angle increases due to distortions: the axial thrust on the compressor also increases. Figure 10 shows the predictions of the variation of axial thrust on the compressor.

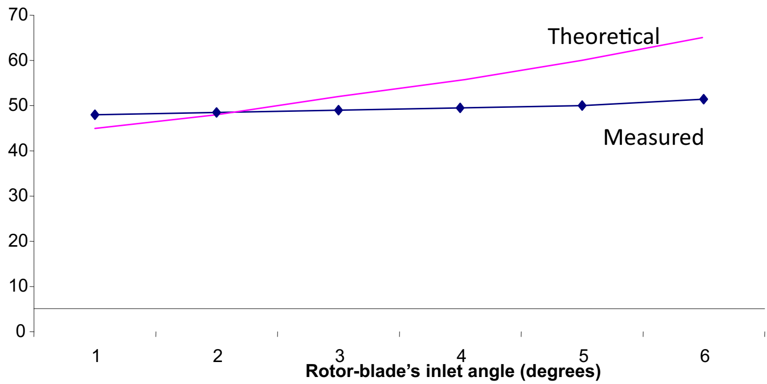


Figure 10: Variations of the Axial Thrust with Rotor-Blade's Inlet Angle

Both the trends of the measured values and the theoretical predictions are in qualitative agreement. Increases in rotor-blade's inlet angle due to blade-profile distortions result in increases of the axial thrust on the compressor, and can lead to vibration problems. There is a direct correlation between blade profile distortions and vibratory stresses resulting from axial thrusts. This corroborates the findings of Luedke (2001) and Vahdati, *et al* (2000). The cumulative effect of these is a drop in the gas-turbine's thermodynamic performance, reliability and operational life.

4.2 The compressor power requirement.

The power required in terms of blade angles is given by

$$\dot{W} = \dot{m} U C_a (\tan \beta_1 - \tan \beta_2) \quad (5)$$

$$C_a = C_1 Cos\alpha_1$$

For a stage C_a = *Constant*

The graphical presentation for the variation of compressor power with the rotor blade inlet angle (β_1), rotor blade outlet angle (β_2) and the rotor blade turning angle $(\beta_1 - \beta_2)$ are shown in figures 11, 12 and 13 (Lebele-Alawa, 2007)

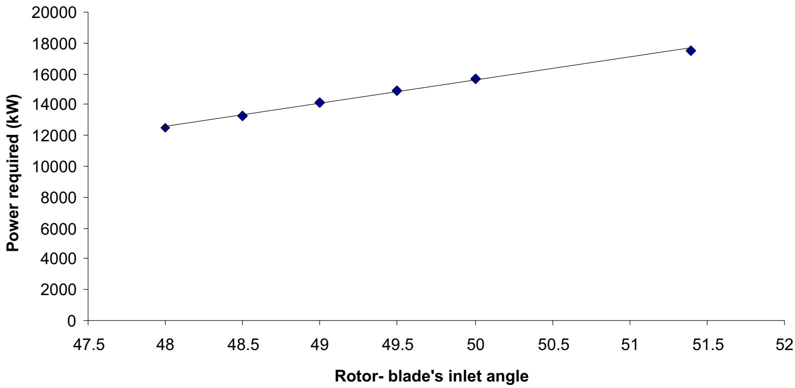


Figure 11: Measured Variation of Compressor Power with Changes in Rotor Inlet Blade Angle.

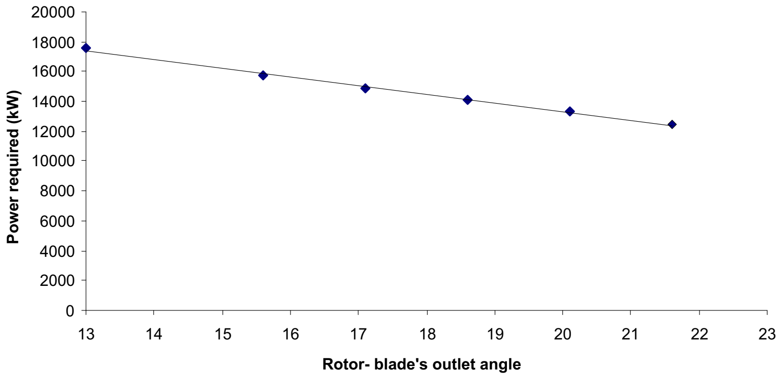


Figure 12: Measured Variation of Compressor Power with Changes in Rotor Blade Outlet Angle.

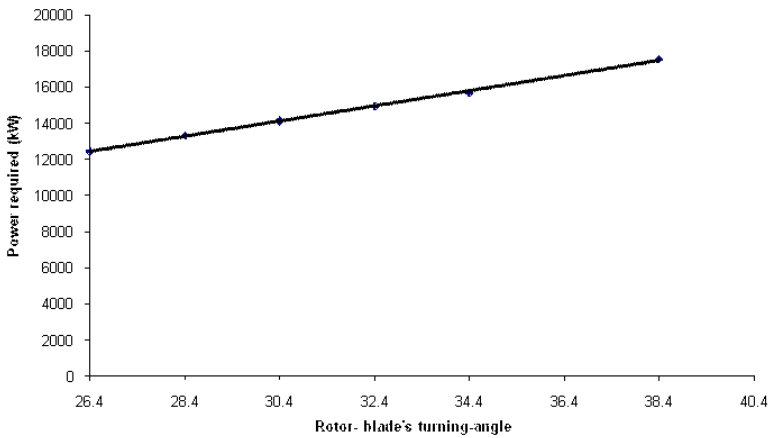


Figure 13: Measured Variation of Compressor Power with Changes in Rotor Blade Turning Angle

5.0 THERMAL POWER AVAILABILITY USING ENERGY AND EXERGY METHODS

Energy and exergy analysis are based on the first and second laws of thermodynamics.

5.1 First Law of Thermodynamics

This is also known as the principle of conservation of energy. It states that during any cycle, the cyclic integral of heat added to a system is proportional to the cyclic integral of work done by the system.

$$\oint \delta Q = \oint \delta W \cdot (Q_{in}\, J, W_{in}\, J) \qquad (6)$$

Considering a process, the first law can be represented as follows

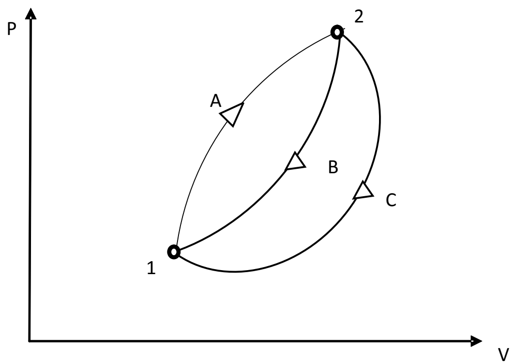


Figure 14: Sketch of P-V Diagram for various Combinations of Processes Forming Cyclic Integral

Through different paths:

- Cycle I: 1 to 2 on path A followed by 2 to 1 on path B.
- Cycle II: 1 to 2 on path A followed by 2 to 1 on path C.

Re-writing the first law, we have

$$\int_1^2 \delta Q_A + \int_2^1 \delta Q_B = \int_1^2 \delta W_A + \int_2^1 \delta W_B \quad Cylce\ I \qquad (7a)$$

$$\int_1^2 \delta Q_A + \int_2^1 \delta Q_C = \int_1^2 \delta W_A + \int_2^1 \delta W_C \quad Cylce\ II \qquad (7b)$$

From Equations (7a) and (7b), we have

$$\int_2^1 \delta Q_B - \int_2^1 \delta Q_C = \int_2^1 \delta W_B - \int_2^1 \delta W_C$$

Rearranging, we have

$$\int_2^1 (\delta Q - \delta W)_B = \int_2^1 (\delta Q - \delta W)_C (8)$$

Energy is a new extensive property of the system denoted by *E*. It is a theoretical construct suggested by the first law of thermodynamics as something to account for the difference between heat transfer and work in any process between the same start and end states.

Thus, $dE = \delta Q - \delta W$ (9)

And $\int_1^2 dE = \int_1^2 \delta Q - \int_1^2 \delta W$

This gives, $E_2 - E_1 = 1Q_2 - 1W_2$ (10)

First Law of Thermodynamics: For a system undergoing a process, the change in energy is equal to the heat added to the system minus the work done by the system.

Let us lump all the types of energy which are not potential or kinetic into a single term U (Internal energy).The internal energy (U) is that portion of the total energy E which is not kinetic or potential energy. It includes thermal, chemical, electric, magnetic and other forms of energy.

Note: Changes in U are associated with changes in the thermal energy of the system.

$$\underset{\text{Total Energy}}{E} = \underset{\text{Internal Energy}}{U} + \underset{\text{Kinetic Energy}}{KE} + \underset{\text{Potential Energy}}{PE}$$

Taking differentials, the first law can be written as

$$dU + d(KE) + d(PE) = \delta Q - \delta W \qquad (11)$$

It can be shown that

$$E_2 - E_1 = U_2 - U_1 + \frac{1}{2}m(V_2^2 - V_1^2) + mg(Z_2 - Z_1) \qquad (12)$$

And $U_2 - U_1 + \frac{1}{2}m(V_2^2 - V_1^2) + mg(Z_2 - Z_1) = 1Q_2 - 1W_2$ (13)

5.2 The Second Law of Thermodynamics

According to Clausius, it is impossible to construct a system which will operate in a cycle and transfer heat from a cooler to a hotter body without work being done on the system by the surrounding (figure 15).

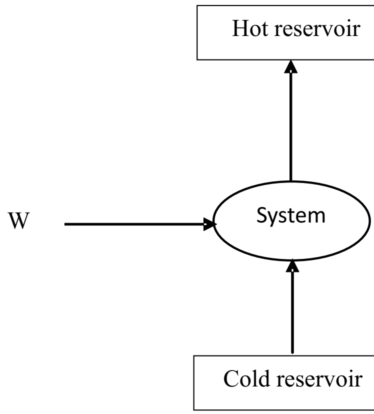


Figure 15: Sketch Showing a System with Two Thermal Reservoirs

Another statement of the second law as presented by Kelvin-Planck stated that it is impossible for any system to operate in a thermodynamic cycle and deliver a net amount of work to its surroundings while receiving an energy transfer by heat from a single thermal reservoir. In other words, it is impossible to construct a system which will operate in a cycle extract heat from a reservoir, and do an equivalent amount of work on the surroundings. It is important to note that all the statements of the second law are equivalent.

5.2.1 Second Law in terms of Entropy

Entropy is a thermodynamic property which provides a quantitative measure of the disorder of a given thermodynamic state.

From the statement of the second law

$$\oint \frac{\delta Q}{T} \leq 0 \qquad \textit{valid for reversible and irreversible heat transfer}$$

And a definition of entropy

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} \qquad \textit{provided the heat transfer is reversible}$$

Combining them, we can cast the second law in terms of entropy. Consider the cycle in the T-S diagram below.

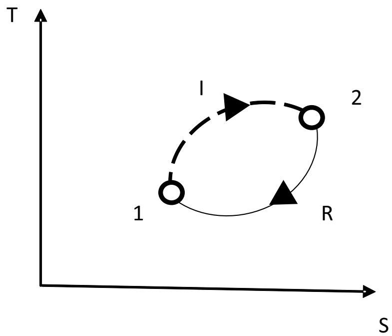


Figure 16 : T-S Diagram Showing Reversible and Irreversible Processes

Process 1 → 2 along path I represents an irreversible process.

Process 2 → 1 along path R represents an irreversible process.

The second law holds

$$\oint \frac{\delta Q}{T} \leq 0; \qquad 0 \geq \oint \frac{\delta Q}{T}$$

The equality implies all processes are reversible, the inequality implies some portion of the process is irreversible.
Now for a reversible process

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} \qquad (15a)$$

Since the process is reversible, we reverse it to get.

$$S_1 - S_2 = \int_2^1 \frac{\delta Q}{T} \qquad (15b)$$

Now applying the second law to the scenario of the figure:

$$0 \geq \left(\int_1^2 \frac{\delta Q}{T}\right)_I + \left(\int_2^1 \frac{\delta Q}{T}\right)_R$$

i.e. $0 \geq \left(\int_1^2 \frac{\delta Q}{T}\right)_I + S_1 - S_2$

or $S_2 - S_1 \geq \left(\int_1^2 \frac{\delta Q}{T}\right)_I$

More generally we can write the second law as

$$S_2 - S_1 \geq \int_1^2 \frac{\delta Q}{T} \qquad (16)$$

If $1 \rightarrow 2$ is reversible, the equality holds.

If $1 \rightarrow 2$ is irreversible, the inequality holds.

Lebele-Alawa and Asuo (2013) evaluated the performance of a 20MW Gas Turbine plant using energy and exergy methods. The energy and exergy analyses provide data for reduction in wastages and optimization of the power output of the plant. The study was conducted on a 20 MW gas turbine power plant at Kolo-Creek Power Station (at Imiringi Community, Ogbia Local Government Area of Bayelsa State). The Plant was the only power generating station in the locality before Bayelsa State was connected to the national grid in 2006.The Kolo-Creek Power Plant operated on Bryton Cycle. With the gas generator set in motion, the hydraulic starter was engaged for the starter drive to rotate the high pressure (HP) compressor and turbine assembly. The low pressure (LP) compressor was then rotated by induced air flow from the HP compressor. The air at increased pressure and temperature was passed into the combustor where a small portion passed through the primary holes into the head of each flame tube to mix with the fuel injected from the eight burners. The ignition system was switched on and the air/fuel mixture combusted by the two ignition plugs. Air was introduced through primary holes to dilute the gas, thereby reducing the gas temperature at the turbine entry. The diluent air confined the flame in the central part of the flame tube to prevent impingement on the walls. The hot gases from the flame tubes at high velocity entered the first stage of HP turbine through nozzle guide vanes for the turbine to rotate and drive the HP compressor. From HP turbine, the gases passed through to rotate the second stage LP turbine, and finally exhausted via an exhaust annulus. The exhaust gas was discharged into the inter-turbine duct, flowed in three-stages of the power turbine with final uptake by the exhaust volute. The drive output of the power turbine was taken by a transmission shaft to the alternating current (AC) generator mounted on a base plate.

Energy and Exergy analysis were conducted to evaluate the optimal performance of the 20 MW gas turbine powerplant. The energy analysis was based on First Law of Thermodynamics, while the exergy method used bothFirst and Second Laws of Thermodynamics. The locations and magnitude of losses which inhibited the performance of the power plant were identified by balance system equations. The internal losses associated with eachplant component were estimated for improvement to be made to such component for maximum power output.The energy efficiency was 20.73 %, while the exegetic efficiency was

16.39 %; but the exergy loss of 38.62 %in the combustor was the largest among the components of plant.

5.3 Energy Analysis

Energy analysis of operations and processes of a power plant is based on the first law of thermodynamics, which regards all forms of energy crossing a system boundary as equivalent, without differentiating between quality and quantity of the energy. Energy balance does not provide information about internal losses in a system, as energy analysis of an adiabatic system shows that the processes are without losses. Similarly, energy balance for an isolated system in a non-stable equilibrium state shows that the process the system undergoes exhibit no losses. This cannot be true since the causes of thermodynamic losses in thermal, chemical and mechanical processes (such as heat transfer, mixing, combustion and viscous flow), cannot be determined by energy balance. Such losses are not associated with quantity of energy, but with decrease in quality of the energy. However, energy wastage can be minimized by various methods in order to optimize the system by holistic evaluation using both energy and exergy methods (Hussein, *et al*, 2011).

5.4 Exergy Analysis

Exergy Analysis is a formalized way of applying availability theory to engineering installations such as power generating plants. Energy (first law) analysis keeps track of the heat and work transfer but does not indicate the source and magnitude of the irreversible entropy creation. Exergy (second law) analysis provides this information. Exergy of a system is the maximum useful work that can be obtained when the system proceeds to equilibrium with the environment. Exergy analysis provides a means to evaluate the degradation of energy, entropy, and loss opportunities to do work; and thus offers an alternative approach to improve operations of power plants. Exergy can be destroyed or lost due to irreversibility of a process and thus provides a measure of the thermodynamic losses in the system by locating and quantifying wasteful energy.

Exergy analysis can be used to identify the system component(s) responsible for irreversibility or lost work, and for efficiency improvements to be directed to such component(s). Exergy analysis clarifies the actual efficiency of a process and the method is very useful in finding the unit operations where efficiency improvements are the most needed (Lebele-Alawa and Asuo, 2013).

In the work carried out by Lebele-Alawa and Asuo (2013), the energy analysis showed that the combustor had the highest quantity of energy, but failed to show the highest loss and irreversibility in the component. The analysis also revealed that the smaller the irreversibility associated with a component, the greater the power that was produced or consumed by the component. The combustor with the highest amount of heat generated, gave the highest irreversibility in the plant with the maximum exergy loss of 38.62 %; implying that most of the energy generated in the combustor was not available to do work, but rather wasted during the process. The power turbine with the minimum exergy loss of 5.13 % was the most efficient of the plant components.

In all the components, energy analysis gave only the quantity of heat generated or work done in the components without the losses or wastes involved in the process. Hence the first law of thermodynamic analysis did not provide information about internal loss in a system and could not be used to find the unit operations where efficiency improvements are most needed.

With exergy analysis, the higher irreversibility in the combustor originated from the fluctuating gas pressure of the primary source of supply of natural gas and variations in system demands which led to the venting of excess gas into the atmosphere as wasted exergy; which could be contained in a reservoir by introducing additional control valves.

The general observation was that exergy analysis provided information for potential improvement and optimization of thermodynamic processes.

5.0 AVAILABLE AND UNAVAILABLE ENERGY

The energy content of a system can be divided into two parts.

The first part is the available energy, which under ideal conditions may be completely converted into work. The second part is the unavailable energy which is usually rejected as waste. Considering Q units of heat energy available at a temperature T. The available part of the energy can be obtained by assuming that the heat is supplied to a carnot engine. The work obtained from the carnot engine

$\left(\frac{T - T_o}{T}\right) Q$ *is the available part.*

The quantity $\left(\frac{T_0}{T}\right) Q$ is the unavailable part.

T_0 is the ambient temperature. Hence the available and unavailable part of energy content of a system depends on the ambient conditions also.

6.1 Availability

The maximum useful work that can be obtained from the system such that the system comes to a dead state, while exchanging heat only with the surroundings is known as availability of the system.Here the term dead state means a state where the system is in thermal and mechanical equilibrium with the surroundings. Therefore, for a closed system availability can be expressed as

$$\emptyset = (U - U_o) + P_o(V - V_o) - T_o(S - S_o) \quad (17)$$

Similarly, for an open system

$$\varphi = (H - H_o) - T_o(S - S_o) \quad (18)$$

In steady flow systems the exit conditions are assumed to be in equilibrium with the surroundings. The change in availability of a system when it moves from one state to another can be given as:

For a closed system

$$\emptyset_1 - \emptyset_2 = (U_1 - U_2) + P_o(V_1 - V_2) - T_o(S_1 - S_2) \quad (19)$$

For an open system

$$\varphi_1 - \varphi_2 = (H_1 - H_2) - T_o(S_1 - S_2) \quad (20)$$

$\boldsymbol{Note}: Availability\ (or\ Energy) = Maximum\ possible\ work - Irreversibility$

$$W_{max,useful} = W_{rev} - I$$

6.2 Irreversibility

Work obtained in an irreversible process will always be less than that of a reversible process. This difference is termed as irreversibility (i.e.) the difference between the reversible work and the actual work for a given change of state of a system is called irreversibility.

$I = W_{rev} - W_{act}$

Consider a stationary closed system receiving QKJ of heat and giving out

$W_{act}\ KJ\ of\ work.$

From 1st law of thermodynamics

$$Q - W_{act} = U_2 - U_1$$

$$W_{act} = U_1 - U_2 + Q$$

$$= (U_1 - U_2) - T_o(\Delta s)_{system} \qquad (21)$$

$$\therefore I = W_{rev} - W_{act}$$

$$= (U_1 - U_2) - T_o(\Delta s)_{system} - (U_1 - U_2) - Q$$

$$= T_o(\Delta s)_{system} - Q$$

$$= T_o(\Delta s)_{system} + T_o(\Delta s)_{surrounding} \qquad (22)$$

$= T_o(\Delta s)_{Universe}$ (A combination of the system and the surrounding is termed the universe).

Similarly, for a steady flow system

$$I = W_{rev} - W_{act}$$

Where

$$W_{rev} = \dot{m}(h_1 - h_2) - T_0(S_1 - S_2)$$

$$W_{act} = \dot{m}[(h_1 - h_2)] + Q_{system}$$

$$Q_{system} = Q_0 = T_0(\Delta s)_{surrounding}$$

Therefore

$$I = T_o(S_1 - S_2) + T_0(\Delta s)_{surrounding} \qquad (23)$$

$$= T_o\left(\Delta s_{system} + \Delta s_{surrounding}\right)$$

$$= T_o(\Delta s)_{Universe}$$

7.0 IMPROVED DESIGNS FOR THERMAL POWER AVAILABILITY

7.1 25MW Power Plant at Omoku

Lebele-Alawa and Le-ol (2015) carried out an improved design of a 25MW gas turbine power plant at Omoku in the Niger Delta area of Nigeria, using a combined cycle. It entailed retrofitting a steam bottoming plant to the existing 25MW gas turbine plant by incorporating a heat recovery steam generator. The focus is to improve performance as well as reduction in total emission to the environment. Direct data collection was obtained from the

HMI monitoring screen, log books and manufacturer's manual. Employing the application of MATLAB, the thermodynamics equations were modelled and appropriate parameters of the various components of the steam turbine power plant were determined. The results show that the combined cycle system had a total power output of 37.9MW, made up of 25.0MW from the gas turbine power plant and 12.9MW (an increase of about 51%) from the steam turbine plant, having an HRSG, condenser and feed pump capacities of 42.46MW, 29.61MW and 1.76MW respectively. The condenser cooling water parameters include a mass flow of 1180.42kg/s, inlet and outlet temperatures of 29.8°C and 35.8°C respectively. The cycle efficiency of the dry mode gas turbine was 26.6% whereas, after modification, the combined cycle power plant overall efficiency is 48.8% (about 84% increases). Hence, SIEMENS steam turbine product of MODEL: SST-150 was recommended as the steam bottoming plant. Also the work reveals that a heat flow of about 42.46MW which was otherwise being wasted in the exhaust gas of the 25MW gas turbine power plant could be converted to 12.9MW of electric power, thus reducing the total emission to the environment.

The Omoku gas turbine is located in the Niger Delta area of Nigeria. It has Six (6) units, each consisting of a gas turbine for electricity generation of 25MW and exhaust gas temperature of

487^0C. The manufacturer of the gas turbine system is General Electric (GE) and it is an MS 5001 (GE Frame 5), single shaft model having a double bus single breaker; 3 × 80MVA transformer and a 33kV to 132kV line switchyard system which is capable of evacuating 200MW of power (Service manual MS 5001).

It was observed that the hot exhaust gas from the Omoku gas plant is vented to the atmosphere, as common to all conventional open cycle gas turbines; hence, considerable amount of heat energy goes as waste with the exhaust of the gas turbine, thus, contributing to environmental pollution. Rather than flaring the hot exhaust gas from this Brayton cycle plant, this heat can be utilized for other useful purposes using heat recovery steam generator (HRSG). This reason prompted the need for the improved design of the Omoku gas turbine power plant to a combined-cycle system such that the otherwise wasted heat from the hot exhaust gas is captured and channelled into the HRSG and used to generate steam to drive another generator (steam generator) to produce more electricity, thus improving the performance (increased total output and efficiency) with less environmental pollution compared to the single cycle turbine plant. Hence, a combined cycle power plant (CCPP) usually consists of a Gas turbine plant, a HRSG and a Steam turbine plant. As common to all conventional CCPP, this modification requires a source of cooling water to be utilized in the condensing unit of the proposed steam turbine system. However, it is interesting to know that there is a flowing river close to the Omoku gas plant which enhances the availability of water source for this purpose.

7.1.1 Combined Gas and Steam

The combined-cycle unit combines the Rankine (steam turbine) and Brayton (gas turbine) thermodynamic cycles using HRSG. The HRSG comprising the economizer, the evaporator, and the superheater, trapped the energy in the exhaust gases of the turbine to generate steam for the steam turbine to generate more power. The schematic diagrams are shown in Figures 17 and 18.

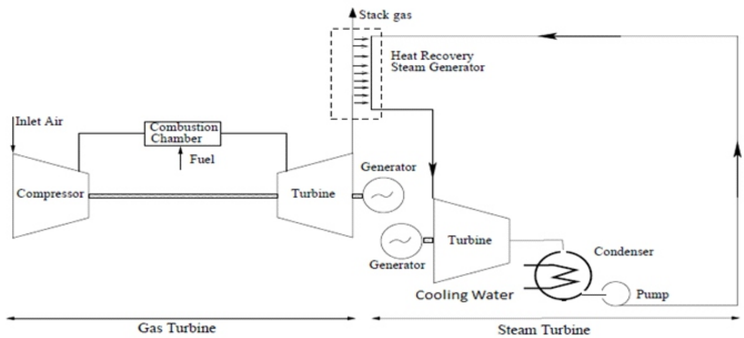


Figure 17:Schematic Diagram of Combined Gas and Steam Cycle

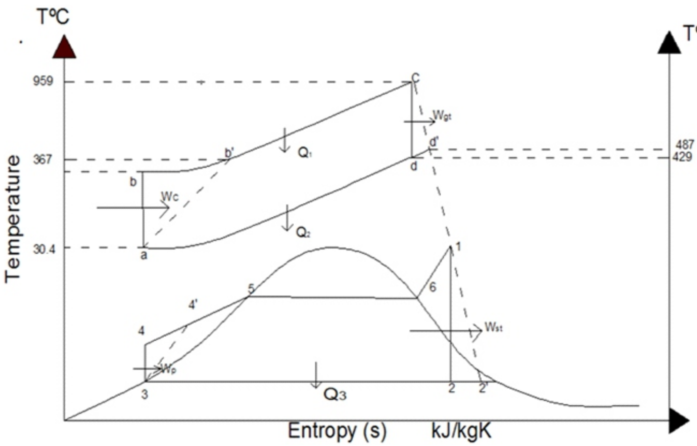


Figure 18: Combined Gas and Steam Cycle Power Plant on T-S Diagram

From Figure 18:

Omoku Gas turbine Cycle $\equiv$ a $\rightarrow$ b $\rightarrow$ c $\rightarrow$ d $\rightarrow$ a

Steam Turbine Cycle $\equiv$ 1 $\rightarrow$ 2 $\rightarrow$ 3 $\rightarrow$ 4 $\rightarrow$ 1

Process 4 $\rightarrow$ 1 is the heat transfer process from exhaust gas of gas turbine to the Heat Recovery Steam Generator of the Steam turbine part.

The combined-cycle overall efficiency and power output are given by:

$$\eta_{cc} = \eta_{ST} + \eta_{gT} - \eta_{ST}\eta_{gT} \quad (24)$$

$$P_{cc} = P_{ST} + P_{gT} \quad (25)$$

Where η_{cc} *and* P_{cc} are combined cycle overall efficiency and power output respectively.

P_{ST} = Steam turbine power output, P_{gT} = gas turbine power output

η_{ST} = Steam turbine thermal efficiency

η_{gT} = Gas turbine thermal efficiency

7.2 Performance Optimization with Heat Recovery Steam Generator(HRSG)

Adumene and Lebele-Alawa (2015) carried out a research on optimizing the power plant performance using a Heat Recovery Steam Generator (HRSG). HRSG is the standard term used for a steam generator producing steam by cooling hot gases. Heat recovery system is obviously a very desirable energy source, since the product is available almost operating cost –free and increase the efficiency of the cycle in which it is placed, either for steam generation or for incremental power generation. Heat recovery steam generator can regain energy from waste-gas streams, such as incinerator gases, furnace effluents or most commonly the exhaust of a gas turbine.

In modern operation of heat transfer equipment such as heat recovery steam generator (HRSG), the exit gas temperature determines the amount of energy extracted from the flue gas stream of the gas turbine. This is an indication of the HRSG performance. Therefore, efforts are often made to lower the stack temperature as much as possible taken into consideration cost effectiveness and low temperature corrosion. The modifications of a single pressure HRSG to multi-pressures have also improve the energy efficiency of the heat recovery steam generator unit. Heat recovery steam generators can be made up from a number of components, including evaporators, economizers, superheaters, reheater, integral deaerators and preheaters. Each of the heat transfer sections performs a specific task, and the one that is selected are generally dictated by the required steam conditions for process use or power generation, the type of power generation and/or the efficiency requirement, weighed against HRSG costs.

Heat recovery steam generator evaporator sections act to vaporize water and produce steam in one component. A bank of finned tubes is extended through the gas turbine's exhaust gas path from a steam drum (top) to a lower (mud) drum. The gas turbine is a very satisfactory means of producing mechanical power (Lebele-Alawa *et al*, 2008; Lebele-Alawa, 2010b). Feed water is carefully supplied at the appropriate pressure to the upper drum below the water level, and circulates from the upper to lower drum, back to the upper drum by convection within the finned tube.

The economizers are serpentine finned-tube gas-to water heat exchangers, and add sensible heat (preheat) to the feed water, prior to its entry into the steam drum of the evaporator. Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment (Lebele-Alawa and Egwanwo, 2012). In single pressure HRSG, the economizer will be located directly downstream (with respect to gas flow) of the evaporator section. In multi-pressure unit, the various economizer sections may be split, and be located in several locations both upstream and downstream of the evaporators. The superheater is a separate serpentine tube heat exchanger which is located upstream (with respect to gas flow) of the associated evaporator. This component adds sensible heat to the dry steam, superheating it beyond the saturation temperature.

In gas turbine heat recovery steam generator, its performance is dependent on the gas turbine exit temperature, inlet gas temperature, feed water temperature and steam pressure. The low exhaust gas temperature generates less steam on unit gas mass basis in the HRSG evaporator unit (Ganapathy, 2001).

The HRSG is widely used equipment in various industries to which include process, power generation, and petroleum industry. The development of HRSG as a component part of the combined power cycle and cogeneration has improved power production and enhanced costs effectiveness within the sector. Energy and materials saving consideration, as well as economical consideration have stimulated the high demand for high efficient HRSG.

The hierarchical strategy implied for optimization of the whole combine – cycle power plant is as follows optimization of the gas turbine cycle, optimization of the operation parameters of the HRSG, and detailed optimization of the single heat exchanger section in the HRSG. One of the suggested ways to reach theoretically maximum efficiency is by increasing the turbine inlet temperature (Ongiro *et al*, 1997). This requires a highly advanced cooling system to cool down the blades of the gas turbine. With existing technology levels, focus can be fixed on the HRSG, and its operating parameters to improve the efficiency of the combined- cycle plants. Optimization of the operating parameters of HRSG is the first step in the optimum design of the plants.

Models for the Heat transfer coefficients and Effectiveness

The overall heat transfer coefficient U by the total heat exchange area is calculated as follows (Ahmadi *et al*.,2011):

$$U_{A_T} = \frac{Q}{(f,\ LMTD)};\ A_T = A_{hrsg} = \sum_{SH} A_{SH} + \sum_{EVA} A_{EVA} + \sum_{EC} A_{EC} \qquad (26)$$

The logarithmic mean temperature difference is estimated from the following equation

$$LMTD = \left[\frac{(T_{gi} - T_{we}) - (T_{ge} - T_{wi})}{\ln\left(\frac{T_{gi} - T_{we}}{T_{ge} - T_{wi}}\right)}\right] \qquad (27)$$

$$\varepsilon_{HRSG_{LP\ Components}} = \frac{\text{Actual heat transfer rates}}{\text{Max. possible heat transfer rate}} = \frac{Q}{Q_{max}}$$

$$= \frac{m_s(h_{out} - h_{in})LP}{m_g C_{pg}(T_{g_{in}} - T_{g_{out}})} = \frac{m_s C_{pw}(T_{w_{out}} - T_{w_{in}})LP}{m_g C_{pg}(T_{g_{in}} - T_{g_{out}})} \qquad (28)$$

$$\varepsilon_{HRSG_{LP\ Components}} = \frac{\text{Actual heat transfer rates}}{\text{Max. possible heat transfer rate}} = \frac{Q}{Q_{max}}$$

$$= \frac{m_s(h_{out} - h_{in})LP}{m_g C_{pg}(T_{g_{in}} - T_{g_{out}})} = \frac{m_s C_{pw}(T_{w_{out}} - T_{w_{in}})HP}{m_g C_{pg}(T_{g_{in}} - T_{g_{out}})} \qquad (29)$$

The heat duty of the HRSG elements and the exhaust gas were modelled using appropriate equations.

The models gave the heat duty, log mean temperature difference, the heat transfer coefficient for the heating surfaces of the HRSG at exhaust gas temperature of 490^{0}C, 500^{0}C, 510^{0}C, 520^{0}C, and 526^{0}Crespectively.

Figure 19 gave the heat exchange trend within the HRSG system, while Figure 20show the plot of the total heat exchange at different exhaust gas temperature of the gas turbine. It provides a good improvement in the performance.

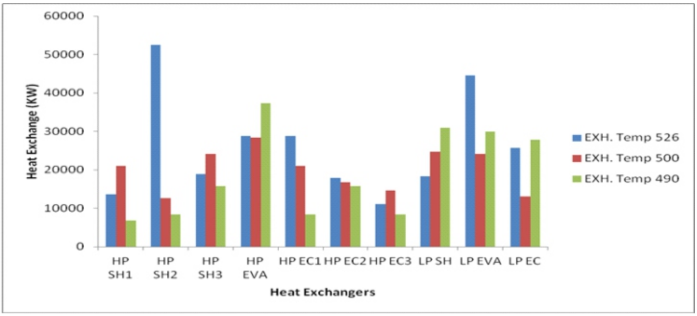


Figure 19 : Absorbed Heat Across the Heating Element of the HRSG

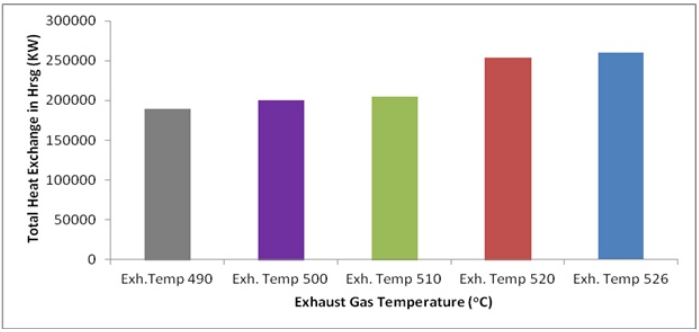


Figure 20: Total Heat Exchange in HRSG at Different Gas Turbine Exhaust Temperature.

8.0 THE EFFECTS OF THE EQUATORIAL RAIN FOREST ENVIRONMENT

Lebele-Alawa and Jo-Appah (2015) carried out a research on Thermodynamic Performance Analysis of a Gas Turbine in an

Equatorial Rain Forest Environment. The variation of operating conditions (ambient temperature, compressor discharge temperature, turbine inlet temperature, exhaust temperature and fuel mass flow rate) on the performance of gas turbine (thermal efficiency, net power output, heat rate, specific fuel consumption and compressor work) were investigated.

The results show that a degree rise in ambient temperature could be responsible for the following: 1.37% reduction in the net power output, 1.48% increase in power drop, 1.49% reduction in thermal efficiency, 2.16% increase in heat rate, 2.17% increase in specific fuel consumption and 0.3% increase in compressor work. Furthermore, the thermal efficiency decreases by 0.006% for 1kcal/kWh increase in heat rate and the heat transfer in the hot gas part was found to increase by 0.16% for a degree rise in ambient temperature. Also the work revealed that the gas turbine had a huge drop in power due to influence of site parameters in contrast to designed data.

The power produced by an expansion 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 economic materials and internal blade cooling technology and to operate the compressor with inlet air flow at a temperature as low as possible.

Overall efficiency of the gas turbine cycle depends primarily upon the pressure ratio of the compressor. It is important to realize that in the gas turbine the processes of compression, combustion and expansion do not occur in a single component as they occurred in a reciprocating engine. It is well known that the performance can be qualified with respect to its efficiency, power output, and specific fuel consumption as well as work ratio. There are several parameters that affect its performance including the compressor compression ratio, combustion inlet temperature and turbine inlet temperature (Rahman and Thamir, 2012).

The effect of surface roughness on thermodynamic performance parameter of axial flow compressor has also been studied (Huadong and Hong, 2013). This study respectively discussed the effect of surface roughness on performance parameter when surface roughness is constant and linearly distributed. The study chooses NASA rotor 37 as study object. Reverse design method was applied to reconstruct the fouled compressor by combining laser triangulation sensor with compressor fouling test rig and then reconstructed solid model is imported into ANSYS CFX to simulate flow field. Two thermodynamic properties – pressure and temperature – stood out as important, as they were measured at the entries and exits of the major components of the gas turbine; namely the compressor, combustion chamber and turbine.

In the treatment and collection of data, mean values of daily parameters were computed by the use of statistical method; followed by monthly average and the overall average for the research period. Some of the phenomena of the operation of the set could not be investigated directly by field measurements because the points were the measurements would have been made where inaccessible. One of such phenomena was the combustion temperature (T_3) of the hot gas as it passes through the combustion chamber. This could not be measured directly and as such the parameter was obtained by the use of thermodynamic relations and equations.

The Net Power Output (W_{net}) is the power generated by the generator and is given as:

$$W_{net} = W_t - W_c \tag{30}$$

Where W_t is the shaft work of the turbine and is given as:

$$W_t = m_p C_p (T_4 - T_3) \tag{31}$$

Where, m_p is mass of product (kg/s), C_p is specific heat capacity of product, $T_3 = TIT =$ turbine inlet temperature. Total heat supplied (Q_{add}), is calculated from the equation:

$$Q_{add} = \left(m_a + m_f\right) \times Cp_a \left(T_3 - T_2\right) = m_f \times CV \tag{32}$$

$$T_3 = \frac{m_f \times CV}{(m_a + m_f) \times Cp_a} + T_2 \tag{33}$$

Where m_a is mass flow of air, m_f is mass flow of fuel, Cp_a is specific heat capacity of air and CV is the calorific value of the fuel. To get $T_{3\ \text{for}}$ a different fixed amount of fuel supply at each T_1, it was therefore necessary to extrapolate.

For steady flow steady state condition, the extrapolation function is given as:

$$ET = CET + (ATTI - CET)(FFS/_{AFS}) \tag{34}$$

Where

ET = Extrapolated temperature, °K

CET = Compressor Exit Temperature, °K

$ATTI$ = Actual Temperature at Turbine Inlet for Actual Fuel Supply, °K

FFS = Fixed Fuel Supply, m^3/s

AFS = Actual Fuel Supply, m^3/s

Exhaust Temperature T_4 can be determined as:

$$T_4 = \frac{T_3}{\left(P_2/P_1\right)^{\frac{(\gamma-1)}{\gamma}}} \tag{35}$$

Where γ is isentropic index of compression of air. For the two isobaric processes, $P_2 = P_3$ and $P_4 = P_1$ Thus the turbine pressure ratio P_3/P_4 is equal to the compressor pressure ratio, $r = p_2/P_1$

Compressor Work (W_c) is given as:

$$W_c = m_a Cp_a (T_2 - T_1) = p_a V_a Cp_a (T_2 - T_1) \tag{36}$$

Where, p_a is density of air, V_a is volume of air aspirated by the compressor, T_1 is ambient temperature and T_2 is compressor discharge temperature.

Thermal Efficiency:

$$n_{th} = \frac{Network}{HeatSupplied} \quad (37)$$

Specific Fuel Consumption (SFC***):*** The ratio of fuel used by a machine to a certain force such as the amount of power the machine produces. And it can be determined by the equation:

$$SFC = \frac{3600 \times mf}{W_{net}} \quad (38)$$

Where $m_f =$ is fuel mass flow rate (kg/s).

The Heat Rate (HR) is a measure used to determine how efficiently a generator uses heat energy. It can be expressed as:

$$HR = \frac{HeatSupplies}{PowerGenerated} \quad (39)$$

Stoichiometric equation: The stoichiometric (or chemically correct) equation defines the exact proportions of fuel and oxidant that on completion of reaction leave no excess or deficiency in either fuel or oxidant.

$$C_m H_n + \left(m + \frac{n}{4}\right) O_2 + 3.76 \left(m + \frac{n}{4}\right) N_2$$

$$\rightarrow mCO_2 + \frac{n}{2} H_2O + 3.76 \left(m + \frac{n}{4}\right) N_2 \quad (40)$$

The effects on the operating conditions on the power output, thermal efficiency, heat rate, specific fuel consumption and compressor work were determined and plots are shown in figures 21 to 28. Table 1 shows the average temperature, flow rate and power output as measured. In table2 the percentage of design power output is gotten by comparing actual power output with the value of the design power output.

Table 1: Averages of Temperature, Flow Rate and Power Output (Measured)

Ambient Temperature	Compressor Exit Temperature	Fuel Gas Supply	Turbine Inlet Temperature	Exhaust Temperature	Power Output
°C (T_1)	°C (T_2)	/	°C (T_3)	°C (T_4)	MW
24	340	3	1217	532	33
25	341	3	1218	533	33
26	343	3	1220	534	32
27	345	3	1222	535	32
28	348	3	1225	536	32
29	349	3	1226	537	32
30	350	2.9	1199	522	30
31	353	2.8	1173	519	28
32	355	2.5	1088	462	26
33	357	2.4	1062	459	26
34	359	2.3	1036	434	24
35	360	2.3	1036	434	24

Table 2: Ambient Temperature and Turbine Inlet Temperature and Power Output

Ambient Temperature	Compressor Exit Temperature	Extrapolated Turbine Inlet Temperature	Exhaust Temperature	Actual Power Output	$\frac{\text{Actual Power Output}}{\text{Design Power Output}} \times 100$	Power Drop
°C (T1)	°C (T2)	°C (T3)	°C (T4)	MW	%	%
24	340	1217	532	33.14	89.94	10.06
25	341	1218	533	33.10	86.83	13.17
26	343	1220	534	32.18	84.42	15.58
27	345	1222	535	32.00	83.95	16.05
28	348	1225	536	32.00	83.95	16.05
29	349	1226	537	31.15	81.72	18.28
30	350	1228	538	31.00	81.32	18.68
31	353	1232	540	30.27	79.41	20.59
32	355	1235	542	30.16	79.12	20.88
33	357	1238	544	29.00	76.08	23.92
34	359	1241	545	28.13	73.79	26.21
35	360	1242	546	27.00	70.83	29.17

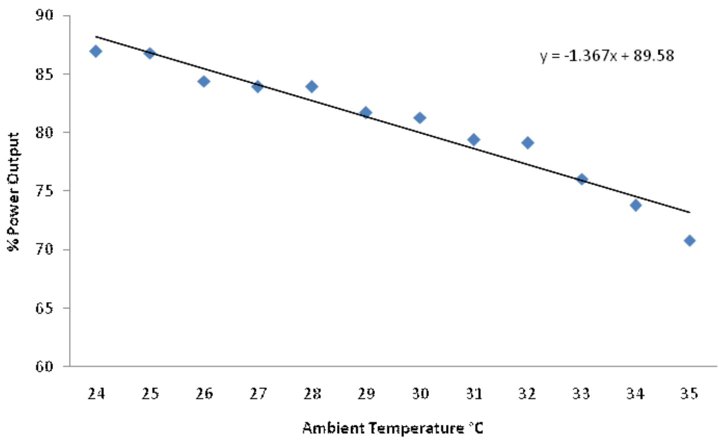


Figure 21: Effect of Ambient Temperature on Power Output

The governing equation is

$$y= -1.3676x + 89.586 \Rightarrow \frac{\delta y}{\delta x} = -1.3676$$

Based on Fig 21 and the resultant straight line equation above, it shows that there is a fall in the power output of about 1.37% for every 1°C rise in the ambient temperature. Translated in real terms it means if a gas turbine operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be a fall in power output of 20.55% or 7.83 Mw for a gas turbine design to generate 38.12 MW at 15°C.

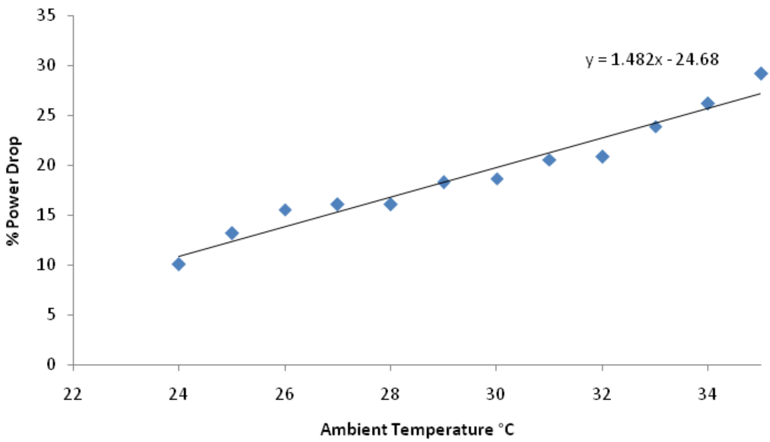


Figure 22: Effect of Ambient Temperature on Power Drop

Based on Figure 22 and the resultant straight line equation above, it shows that there is a rise in the drop of the power output of about 1.48% for every 1°C rise in the ambient temperature. Translated in real terms it means if a gas turbine operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be a rise in the power drop of 22.2% or 8.46 Mw for a gas turbine design to generate 38.12 MW at 15°C.

For this reason colossal losses are incurred in power generating industries in the tropical regions. To have an idea, let us consider some of the gas turbine power generating sets in the Niger Delta.

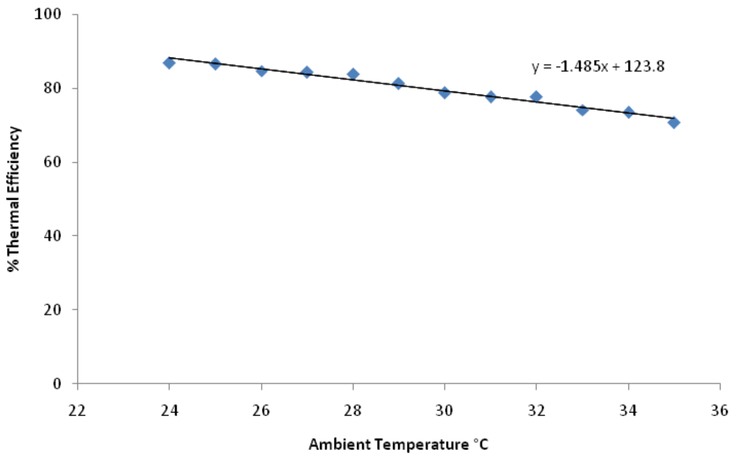


Figure 23: Effect of Ambient Temperature on Thermal Efficiency

Figure23 above shows that there is a fall in thermal efficiency of about 1.49% for every 1°C rise in the ambient temperature. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be a fall in thermal efficiency of 22.35% or thermal efficiency of 6.33% for a gas turbine designed to produce 28.3% η_{th} at 15°C.

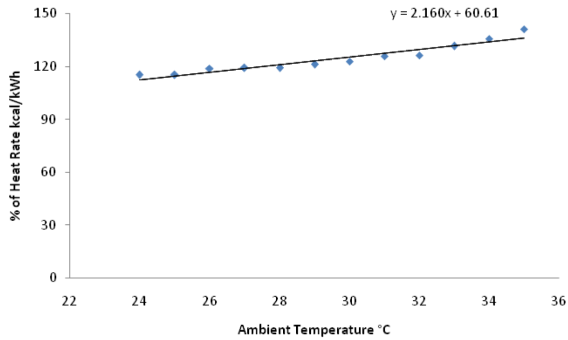


Figure 24: Effect of Ambient Temperature on Heat Rate

Figure24 shows that there is a rise in heat rate of about 2.16% for every 1°C rise in the ambient temperature. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be a rise in heat rate of 32.4% or 987.23 kcal/kWh for a gas turbine designed to generate 3047 kcal/kWh at 15°C

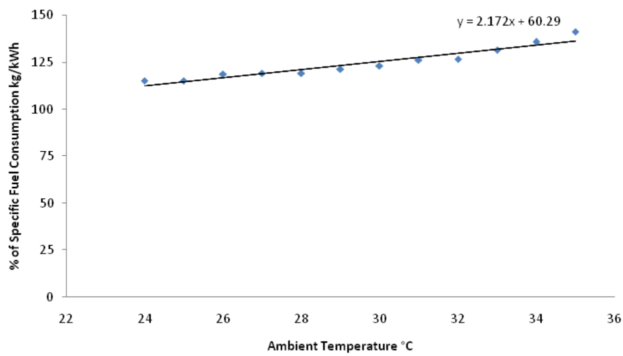


Figure 25: Effect of Ambient Temperature on Specific Fuel Consumption

Figure 25 shows that there is a rise in specific fuel consumption of about 2.17% for every 1°C rise in the ambient temperature. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C instead of 15°C used for the .design, there will be a rise in specific fuel consumption of 32.55% or 0.092 kg/kWh of fuel for a gas turbine designed to have a specific fuel consumption of 0.283kg/kWh at 15°C.

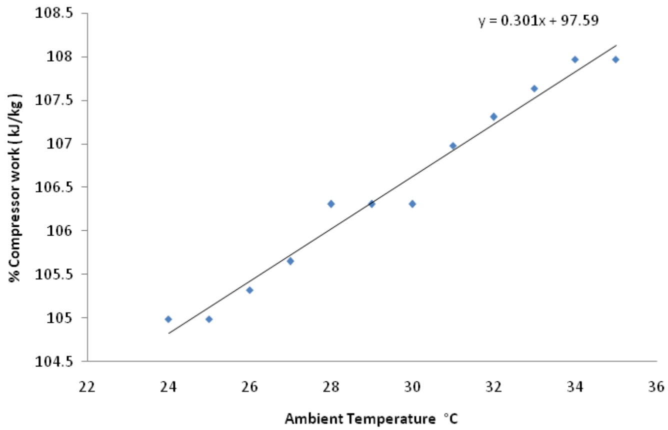


Figure 26: Effect of Ambient Temperature on Compressor Work

Figure 26 shows that there is an increase in compressor work of about 0.3% for every 1°C rise in the ambient temperature. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be an increase in compressor work of 4.5% or 2 MW for a gas turbine designed to have a compressor work of 45.28 MW at 15°C.

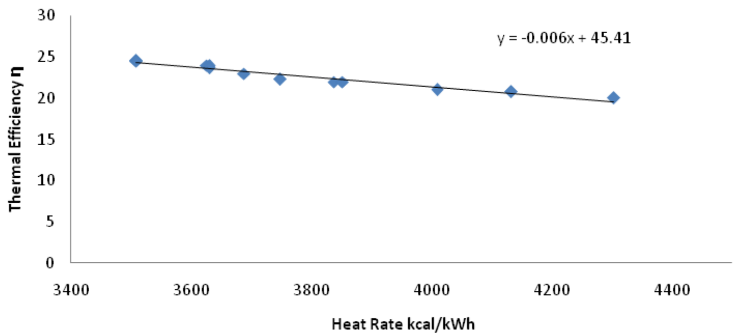


Figure 27: Effect of Heat Rate on Thermal Efficiency

Fig 27 shows that there is a decrease in thermal efficiency of about 0.006% for every rise in heat rate of about 1kcal/kWh. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C having heat rate of about 3748kca/kWh instead of 15°C as 3047kcal/kWh as design, there will be a decrease in thermal efficiency of 22.5%.

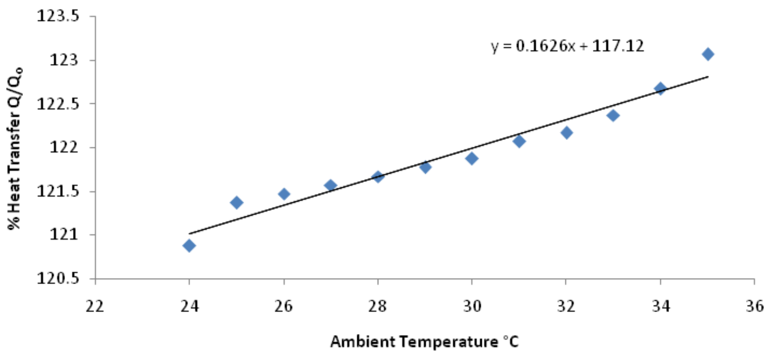


Figure 28: Effect of Ambient Temperature on Heat Transfer

Figure 28 shows that there is an increase in the heat transfer of about 0.16% for every rise in ambient temperature. Translated in real terms it means that if a turbine set operated at an average ambient temperature of 30°C instead of 15°C used for the design, there will be a rise in the heat transfer of about 2.4%. This in effect means higher temperature in the hot gas part. And where there is insufficient air mass flow in the inlet due to choking of inlet air filters as a result of high dust content in the Niger Delta area, the consequence will be undercooling in the hot gas part.

From the foregoing, it can be generally said that the climatic condition that is peculiar in the site that was not fully addressed at the time of installation of the gas turbine affected the operations and performance of the set. This particular condition of high ambient temperature of the area which has a mean daily value of about 30°C and which varies only slightly on both sides of this value as against the general design ambient temperature of 15°C. This high ambient temperature is a negative factor and it affects the thermodynamic process of compression, addition of heat and expansion. Apart from affecting the processes, the components in which these processes do occur namely the compressor, the combustion chamber and the turbine can also be physically affected. Petrochemical gas turbine was chosen as a case study for the analysis because it is in the Niger Delta area where all the known larger power generating gas turbines in the country are installed and it is expected that this will not change for some time to come because the fuel for the combustion is abundant in the area. Nevertheless, huge amount of loss in power is experienced due to high ambient temperature. Gas turbines are the power of industrial plants in which they are installed in terms of electricity generation within the plant. And as such in order to maintain them at high efficient level. A periodic performance evaluation of the equipment is required.

9.0 THE THERMO-ECONOMIC EFFECTS OF BLADE PROFILE DETERIORATION

As mentioned earlier, to provide the requisite aerodynamic flow patterns, compressor blades are correctly shaped and positioned at optimal angles of incidence, and any change in blade geometry would have an impact on the blade intake or exit velocity triangles, potentially resulting in major performance changes (Lebele-Alawa *et. al*., 2008). Changes in deviation at both rotor and stator blade rows and distortions in velocity diagrams at each compressor stage occur when the geometric dimensions of the air foils are changed under distorted conditions.

Using thermo-economic structural theory, Asuo, *et al* (2021), which was a PhD work supervised by me, developed models that establish a relationship between changes in compressor blade anglesand thermo-economic variables. The study also proposed models for entropy generation and capital cost estimation. The advantage of this is that axial compressors can be predicted in advance to help operators and energy investors make decisions regarding performance, sustainability, and economic viability. According to the findings, the losses, operating costs, and thermo-economic variables of axial compressors could be predicted during design stage. Exergy analysis of a gas turbine's axial flow compressor, estimation of quantities and costs of exergy destruction due to variation in compressor blade outlet angle, and analysis of the effects of variation in compressor blade outlet angle on non-exergy related compressor costs were established.

In the study, two levels were used to conduct an energy and thermo-economic analysis, with a focus on the compression process. An energy model was used to estimate the change in pressure ratio caused by variations in compressor blade angles. There are three steps to thermo-economic analysis. The first step was to perform an exergy analysis to determine the exergy flows and losses in the system. The economic analysis step evaluates the monetary costs of the system's installation, operation, and maintenance. Exergy costing was the third step, which was used to estimate the exergy cost of each flow.

Energy Model

We consider both the total temperature rise across the compressor and the temperature rise across a stage in the case of multistage compressors. The model developed are shown below.

$$r^{0}_{p,c} = \left[1 + N_S\left(\frac{\lambda\eta_{is}U_b\Delta V_w}{c_p T_{01}}\right)\right]^{\frac{\gamma}{\gamma-1}} \quad (41)$$

$$\therefore \quad r^{0}_{p,c} = \left[1 + N_S\left(\frac{\eta_{is}\Delta T_{st}}{T_{01}}\right)\right]^{\frac{\gamma}{\gamma-1}} \quad (42)$$

$r^{0}_{p,c}$:Pressure ratio with respect to blade angle ; N_S No of compressor stages

9.1 Thermo-economic model

9.1.1 Exergy model

Due to the limitations of energy analysis in thermal processes, exergy has been developed to account for energy losses due to irreversibilities within the system. The exergy component of fluid in a steady flow is given by the sum of the exergy's kinetic, potential, thermomechanical, and chemical components (Eke *et. al.*, 2018). Exergy analysis allows for the evaluation of energy degradation, entropy degradation, and the loss of opportunities to do work during a process, and thus provides an alternative approach to power plant improvement. Because the processes are fixed in composition, the effects of kinetic and potential energy were considered negligible in the study. In the steady state, the velocity difference between the inlet and output is negligible, so the kinetic energy effect is ignored. Similarly, in industrial equipment such as axial compressors, the elevation difference at inlet and exit is insignificant at steady state, therefore potential energy consequences were neglected. As a result, as illustrated in the models below, exergy wpas defined as the maximum work taken from the stream when it was brought to the reference state by physical exergy.

Exergy of products from compressor

$$E_2 = E_{p,c} = m_a \left[\left(h_2 - h_{ref}\right) - T_{ref} \left\{ \left(s_2^0 - s_{ref}^0\right) - R_a \ln\left(\frac{P_2}{P_{ref}}\right) \right\} \right] \tag{43}$$

Exergy of products from combustion chamber

$$E_3 = E_{p,cc} = m_a \left[\left(h_3 - h_{ref}\right) - T_{ref} \left\{ \left(s_3^0 - s_{ref}^0\right) - R_a \ln\left(\frac{P_3}{P_{ref}}\right) \right\} \right] \tag{44}$$

Exergy of products from turbine

$$E_4 = E_{p,t} = m_a \left[\left(h_4 - h_{ref}\right) - T_{ref} \left\{ \left(s_4^0 - s_{ref}^0\right) - R_a \ln\left(\frac{P_4}{P_{ref}}\right) \right\} \right] \tag{45}$$

Network output from turbine

$$E_5 = W_{NET} \tag{46}$$

Exergy of work input to compressor

$$E_6 = E_{w,c} = m_a \frac{\left[\left(h_2 - h_1\right)\right]}{\eta_{is}} \tag{47}$$

Exergy destruction rate in compressor

$$I_c^0 = m_a T_{ref}\left[\left(s_2^0 - s_{ref}^0\right) - R_a \ln\left(\frac{P_2}{P_{ref}}\right)\right] \qquad (48)$$

Exergetic efficiency of compressor: ε_c

$$\varepsilon_c = \frac{E_{p,c}}{E_{w,c}} \times 100 \qquad (49)$$

9.1.2 Economic model

In practice, every power plant has a useful life that must be met to provide reliable service. Since installation and commissioning, its equipment has steadily depreciated due to component wear and tear, lowering the plant's value. Thermo-economic analyses of the compressor, combustion chamber, and turbine show the investment, operation, and maintenance costs of the various components, and the annualization cost method proposed by Moran and Schuibba (1994) was used in the study.

The compressor equipment cost PEC_c^0 is expressed as

$$PEC_c^0 = \left[\frac{71.1 m_a}{0.9 - \eta_{is}}\right]\left[r_{p,c}^0\right]\ln\left[r_{p,c}^0\right] \qquad (50)$$

The annualized cost of a component is the cost that would result in the same net present cost as the actual cash flow sequence associated with that component if it occurred evenly in every year of the project's existence. The annualized cost (C_c^0) of the compressor was calculated, and all investment cost values in this study are expressed in dollars ($).

Annualization cost of compressor (C_c^0) was given by equation (51) and in the study all investment cost values are expressed in terms of the Dollar ($).

$$C_c^0 = PW_c^0 \times CRF \qquad (51)$$

Total capital investment, operation & maintenance cost of compressor, Z_c^0 is expressed according to equation (54) as follows:

$$Z_c^0 = \frac{\phi \times C_c^0}{3600 \times N} \qquad (52)$$

Figure 29 represents a graph of rotor blade angle versus equipment cost. The graph shows that as β_2 decreases, the cost of the equipment increases. The equipment cost increased by 12.44 percent from $20.0111 to $22.5106 as the β_2 decreased by 1^0 from 27^0 to 26^0. By decreasing β_2 the stage's loading capability and pressure ratio are increased. The pressure ratio rises as the β_2 decreases with a constant β_1 increasing equipment costs.

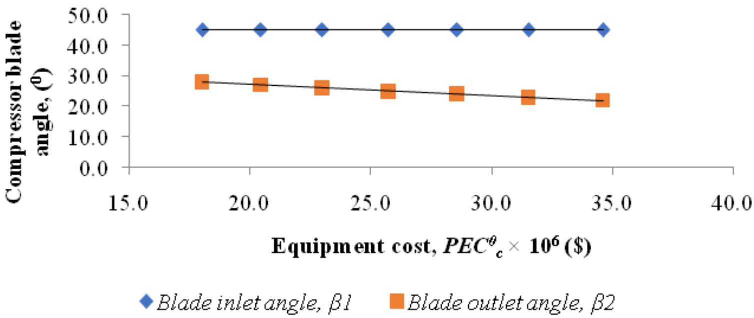


Figure 29: Graph of Compressor Blade Angle Vs Equipment Cost

The graph of rotor blade angles with annualized cost is shown in Figure 30. The annualized cost increased by 12.50 percent as β_2 decreased by 1^0. When compared to Figure 31, the total investment, operation and maintenance costs, pressure ratio, and mass flow rate all increase when β_2 decreases, resulting in an increase in the equipment's annualized cost.

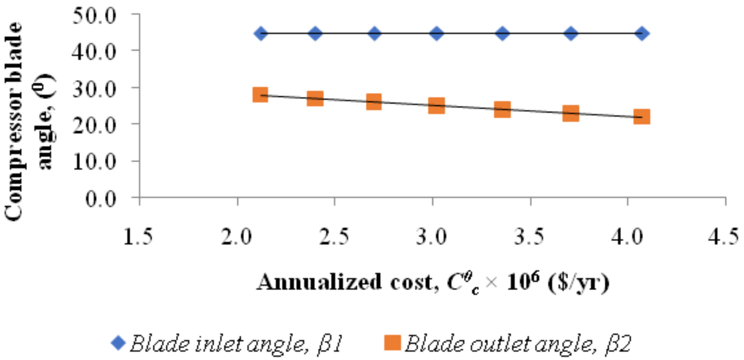


Figure 30. Graph of Compressor Blade Angle Vs Annualized Cost

Figure 31 represents a graph of rotor blade angle versus total investment, operation, and maintenance costs. As β_2 of a centrifugal compressor decreases, so does its head and efficiency. Like axial centrifugal compressors, the efficiency and pressure ratio of an air compressor increase as β_2 decreases because higher pressures require additional work input. According to the graph, as β_2 decreases, the overall investment, operating, and maintenance costs rise. As β_2 decreases by 1^0, the total capital investment, operating, and maintenance costs rise by 12.56 percent.

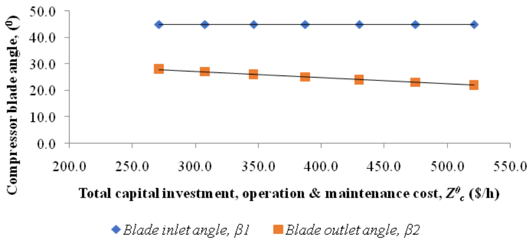


Figure 31: Graph of Compressor Blade Angle Vs Total Capital Investment,

The rate of exergy destruction determines the cost of exergy destruction (Oyedepo *et. al*., 2015). The graph of rotor blade angle against cost of exergy destruction is shown in Figure 32. The cost of exergy destruction rises as β_2 decreases. The rate of entropy generation increases as β_2 is reduced, resulting in an increase in exergy destruction and pressure ratio. As β_2 decreased by 1^0, the cost of exergy destruction increased by 0.72 percent.

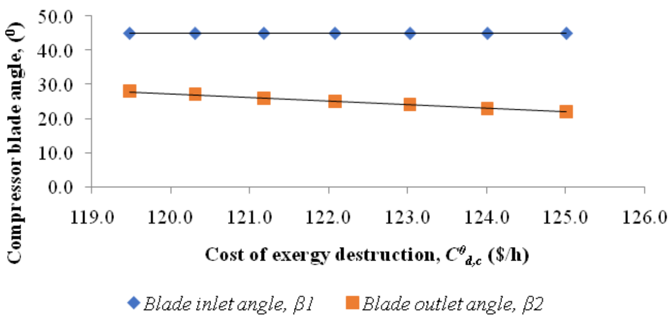


Figure 32: Graph of Compressor Blade Angle Vs Cost of Exergy Destruction

As the pressure ratio rises due to a decrease in β_2, the average cost per unit of exergy input rises. The average cost of exergy input to the system, as well as the cost of additional work, tends to rise as the pressure ratio rises. The average cost per unit of exergy input vs. the rotor blade angle is shown in Figure 33. When β_2 was reduced by 1^0, the average cost per unit exergy input increased by 0.64 percent.

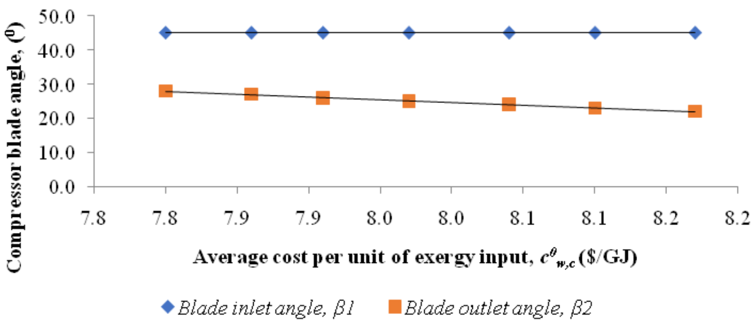


Figure 33: Graph of Compressor Blade Angle Vs Average Cost Per Unit of Exergy Input

A graph of rotor blade angle against average cost per unit exergy output is shown in Figure 34. The average cost per unit of exergy output rises as β_2 decreases. The cost of the unit product rises with pressure ratio due to higher investment costs and exergy destruction at higher turbine inlet temperatures, according to Mondal and Ghosh (2018). Figure 34 shows that as β_2 decreased by 1^0, the average cost of unit exergy output increased by 1.32 percent.

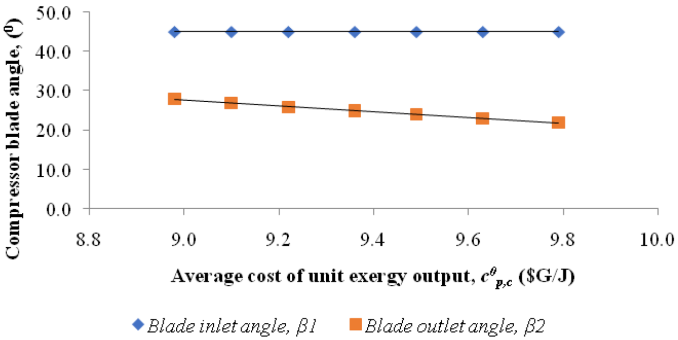


Figure 34: Graph of Compressor Blade Angles Vs Average Cost per Unit of Exergy Output

A graph of rotor blade angle against relative cost difference is shown in Figure 35. Because of the pressure ratio and exergy destruction, the relative cost difference increases as β_2 decrease. The comparison with related data is comparable with that published by Ding *et. al.* (2019), who found that as β_2 decreases, the pressure near the tongue of an impeller increases. Figure 35 shows that as β_2 reduced by 1^0, the relative cost difference increased by 4.89 percent.

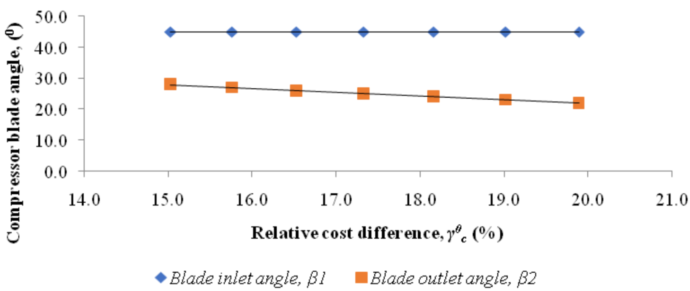


Figure 35: Graph of Compressor Blade Angle Vs Relative Cost Difference

A graph of rotor blade angle vs exergo-economic factor is shown in Figure 36. As β_2 drops, the exergo-economic factor rises due to higher total capital investment, operating and maintenance cost, pressure ratio, and cost of exergy destruction. Exergo-economic factor was shown to be influenced by the overall cost of investment and the cost of exergy destruction, according to Aliu and Ochornma (2018). As β_2 decreased by 1^0, the exergo-economic factor increased by 3.059 percent, as illustrated in Figure 36.

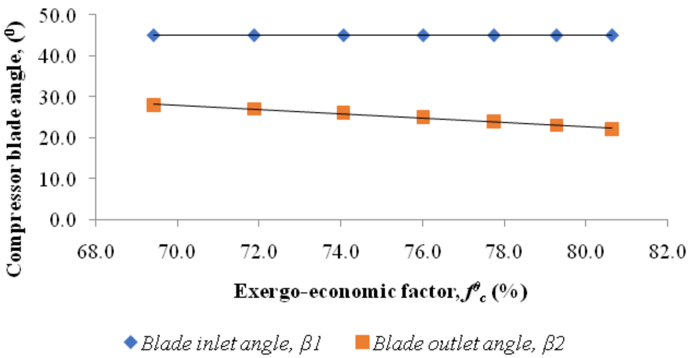


Figure 36: Graph of Compressor Blade Angle Vs Exergo-economic Factor

Summarily, the effects of blade angle changes on the thermo-economic performance of a gas turbine's axial compressor were very profound. The pressure ratio increases as the compressor β_2 decreases. Equipment cost, annualize cost, total investment, operation, and maintenance costs all increase as β_2 decreases, according to the models used in this study. The reason for this is that as β_2 decreases, the pressure ratio rises. Increase in β_2 lower the cost of exergy destruction. More work is required when the pressure ratio increases during compression, resulting in an increase in irreversibilities, entropy generation, and exergy destruction, as well as an increase in the cost of exergy destruction. As β_2 drops, the average cost per unit of exergy input and output rises. The compressor β_2 has a strong influence on thermo-economic performance because it determines the inlet to the next stage, according to the values of the relative cost difference and the exergo-economic factor. A high relative cost difference indicates a high rate of exergy destruction, which can be improved by increasing β_2.

10.0 MY CONTRIBUTIONS

In the course of this lecture the content have stated clearly some of my modest contributions through studies and researches and the outcomes as they relate to thermal power plant effectiveness and improvements in energy availability which are well documented and published. Some organizations and agencies have communicated their acknowledgements of the relevance of these researches to the development of their energy sector as well as manufacturing of power plant and aircraft propulsion equipment. Contributions have also been made to extending the frontiers of knowledge concerning the phenomena of how compressor blade profile changes due to so many outlined factors affect the technical functionality and availability of gas turbines as well as the economic implications on their operations (Lebele-Alawa and Hart, 2001; Lebele-Alawa, 2007; Lebele-Alawa *et al*, 2008; Lebele-Alawa, 2009a; Lebele-Alawa, 2009b; Lebele-Alawa, 2010a; Lebele-Alawa, 2010b; Lebele-Alawa and Asuo, 2011a; Lebele-Alawa and Asuo 2011b; Johnson and Lebele-Alawa, 2011; Lebele-Alawa, 2012; Lebele-Alawa and Egwanwo, 2012; Lebele-Alawa and Ohia, 2013; Lebele-Alawa and Asuo, 2013a; Lebele-Alawa and Asuo, 2013b; Okeke and Lebele-Alawa, 2013; Lebele-Alawa and Ohia, 2014; Aziaka *et al*, 2014; Ochiche and Lebele-Alawa, 2014; Lebele-Alawa and Jo-Appah, 2015; Adumene and Lebele-Alawa, 2015; Lebele-Alawa and Le-Ol, 2015; Nkoi and Lebele-Alawa, 2015; Adumene *et al*, 2016; Asuo *et al*, 2021a; Asuo *et al*, 2021b).

11.0 CONCLUSION AND RECOMMENDATIONS

In the course of this lecture the indispensability of Thermal power and Energy in our day to day life and the technological development of any nation has been established. Key to the effective performance of the heat engine is the compressor which is comprised of cascades of blades. A lot of researches and novel contributions on the rotor-blade's profile as it affects the air mass flow, the pressure ratios, the power requirements, the axial thrust and the overall reliability and availability of the gas turbine has also been presented. The thermo-economic models, costs and other economic implications arising from changes in blade shapes are also very clearly stated. It can therefore be concluded that rotor-blade profile has profound influence on thermal power and energy.

I wish to therefore make some recommendations to improve energy and power availability as well as to mitigate some of the problems identified:

i. **Power and energy hybridization:** The issue of efficient power supply to the Nigerian populace has been a major challenge which has adversely affected the economy and increased the pressure on the government in terms of energy demand. Despite the country's abundance in fossil fuels and renewable energy sources like solar, wind, and natural gas, the Nigerian situation remains that of low electricity access particularly in rural areas. For a nation to develop technologically, reliable energy is needed. The conventional form of power generation (thermal, diesel, etc) can be complemented using a hybrid system to generate more and steady electrical power. Hybridization of energy systems is an important way of increasing the energetic and exergetic efficiencies of energy conversion and utilization systems. Hybrid power solutions combine renewable energy sources, thermal power generation and energy storage systems in a hybrid power plant These hybrid systems can be standalone or grid connected. Hybrid power plants are more cost effective, lower emitting and more resilient than conventional systems such as diesel generators. Hybridization also encourages energy storage for future usage.

ii. **Appropriate inlet air filtration:** Considering the importance of the quality of inlet air to the performance of the compressor blade as well as the entire life of the gas turbine, appropriate filtration of the inlet air should never be compromised. Strict compliance to the maintenance schedules as well as adjustments based on on-site conditions is recommended.

iii. **Offline and online regular blade washing:** Since gas turbines lose efficiency due to the build-up of foulants, it is imperative to carry out regular gas turbine water washing in order to maintain performance. If such build-up persists, airflow through the compressor is disrupted. This in turn affects the efficiency with symptoms such as reduced output, a drop in compressor discharge pressure, increase in operating temperature, greater fuel consumption, and eventual shut down. Financially it can lead to loss of millions of Naira. In gas turbine offline washing, the turbine is stopped and a thorough cleaning process is done to remove all the contaminants. This is done in stages of applying cleaning solutions, series of rinse cycles, dissolving the remaining salts in solutions and a final water rinsing. In online washing, atomised cleaning fluid is injected into the system while the turbine is still running.

iv. Cooling the Turbine inlet air: Most gas turbines used in Nigeria are designed for inlet temperatures far below our ambient conditions. This presents some challenges and implications as the inlet air enters at a temperature far above the design specifications. The implications are quite negative on the performance of the gas turbine. Inlet air cooling is therefore recommended to mitigate this situation. This will result in significant increase in power production and improved power plant efficiency.

v. Re-blading: Re-blading is recommended in cases where there are obvious signs of deterioration of the blades.

12.0 ACKNOWLEDGEMENT

God has been my refuge and strength, a very present and well-proved help in my journey of life. He deserves public acknowledgement, praises and honour. I therefore give Him the first acknowledgement as the one who has made all things possible in my life and made me an Engineering Professor at the appropriate time.

I wish to specially thank our Vice Chancellor Professor Nlerum S. Okogbule for his attractive leadership style and for the opportunity to deliver this inaugural lecturer. I also appreciate the Chairman and members of the Governing Council of this University. I appreciate the other principal officers of the University: The Deputy Vice Chancellor (Academic), Professor V.B. Omubo-Pepple; the Registrar Dr S.C. Enyinda; the University Librarian Prof. Mrs J.N. Igwela; the Acting Bursar Mr J. Ebere, for your tireless services to the University community. The provost, College of Medical Sciences Professor C.G. Orluwene is highly appreciated as well as the Chairman Senate lectures committee Prof N.H. Ukoima and the members of the Senate lectures committee.

Worthy of appreciation are all former Vice Chancellors of this great University who laid a good foundation and also built upon them to produce what we have today as a great University in our

nation. They include Professors T.T.Isoun, J.G. Chinwa (Ag), E.I.O. Banigo, Emeritus Professors A.I. Ahiauzu, S. Odi-Owei, and S.C. Achinewhu. Others are Professors V. Omuaru (Ag), B.B. Fakae, B.C. Didia and O.B. Owei. Your works shall be rewarded. I also appreciate former Deputy Vice Chancellors and Registrars of this great University who in one way or the other contributed to my progress in the University.

I am very grateful to Mr Gioachinno Fundarotto(late), the then Managing Director of W.W.Whyte Int Ltd Trans Amadi Port Harcourt who offered me my first ever letter of employment after my national service. I thank the then head of Engineering of the company, Engr Gabriel Asein and all other staff of the company for their cooperation while we were together. I cannot forget the then SPDC project Engineers of ENGE (C. Essien, C. Obakpolor, T. Opanobi and others) that broadened my understanding of the rudiments pipeline construction and non-destructive testing (NDT) while on their projects.

I appreciate my teachers in primary and secondary schools who contributed greatly to what I am today. Worthy of special mention are two of my secondary school principals at the Government Secondary School Borikiri Port Harcourt, namely Mrs Esther Blue-Jack and Mr Ndah Aguma (of blessed memory) who inculcated discipline in me and showed me the way to greatness. Mr Aguma saw something in me at a young age and appointed me to leadership position at that level. I will never forget my Pakistani physics teacher Mr S.S. Rahman who kept advising us (his students) to 'suffer now and enjoy later' by giving full attention to our books.

I acknowledge with gratitude the former Vice Chancellor of Nigeria Maritime University (NMU) Engr. Prof (Mrs) O.M.O Etebu, who made me the Deputy Vice Chancellor (Administration) of the University. I thank the former Pro-Chancellor and Chainman of Governing Council of NMU Chief Timipre Sylva, other members of Council Prof (Mrs) Flora Oluwafemi (DVC Academic), Hon Gandi Umar Mohammed, Hon.Barr (Mrs) Ijeoma Igboanusi, Hon Farouk Mustapha, Mr Aliyu Saliu, Prof James Asalor, Prof Godwin Ovuworie, late Prof Tosan Otokunefo. It was a pleasant time working together.

My gratitude goes to the Dean, Faculty of Engineering Professor J.G. Akpa and all former Deans of the Faculty. Worthy of special mention are Professors E.N.Wami, A.J. Akor and M.J.Ayotamuno who were my bosses while I served as Acting Head of Department of Mechanical Engineering Department for a long stretch of time. I appreciate all the professors of our great faculty especially Professors K.D.H Bob-Manuel, C.O.Ahiakwo, F.T. Ademiluyi, I. Fubara-Manuel, M.E. Ephraim, D.C. Idoniboyeobu, A.A. Ujile, S.A. Amadi, K.K. Dagde, S.O. Nkakini, C.P. Ukpaka, A.H. Igoni, S.B. Akpila, F.E.

Ezeilo, I. Itolima, E.O. Ekwulo, and E.A. Igwe. Drs I.E. Douglas, E.Ehirim, G. Mmegbu, Engr K. Theophilus-Johnson also deserve special commendation. The present and all former Faculty officers and non-academic staff are equally appreciated. My fathers and colleagues in my Department of Mechanical Engineering are very highly appreciated. On this occasion I wish to pay a very special tribute to late Professor Howells I. Hart who contributed ipmmensely to what I am today. Firstly he was my undergraduate teacher, later my M.Tech and Ph.D main supervisor. Above all, he was the one who asked me to come over to the University and recommended my application for employment in his capacity as the then Head of Department. Professor F.J.K. Ideriah'scontributions to my becoming a professor is immeasurable. May God reward you Sir. I'm also very grateful to my revered academic father and mentor Emeritus Professor S. Odi-owei whose leadership and encouragements impacted my life positively. Late Prof M.T. Lilly helped me at various times in the Department. I also extend my thanks to other staff of the Department: Professor J.I. Sodiki, Drs B. Nkoi, T.K. Jinyemiema, k. Morris, O.E. Isaac, S.L. Bani, R.K. Amadi, D.M. Amafabia, Engrs M.V. Ndor, A.K. Le-Ol, F. Oparadike, F. Worgu, E.C. Eke, K.J. Christopher, I.E. Chuku, G.C. Akaninwo and others. Also the workshop manager Engr Godam Salem and all staff of the workshop as well as the

administrative staff of the Department, including Mrs Ibiye Fubara, Morris Nnokam, Ann Yaakor and Mr Echendu Weli are appreciated.

My parents Mr Pius Lebele-Alawa (of blessed memory) and Mrs Helen Beage Alawa played the foundational role in my life upon which other things rested. They gave me their best, ensuring strict discipline and made sure I became somebody in life. May God continually remember them for good. I thank all my siblings for always being there for me- Mrs Juliana Ezuma, Mrs Christiana Boboh, Mrs Anna Vareba, Mrs Barinem Edward, and Mrs Barivule Gokana. Some members of the Alawa's family of Bodo-City are also specially acknowledged on this occasion –Prof John P. Alawa (late), Sir Joseph Alawa, Prof. Clement B. Alawa, Prof. Rev Fr Peter Alawa, Mr Louis Alawa. I also appreciate all my in-laws for their love and support.

My spiritual father and General Overseer of Charismatic Renewal Ministries Worldwide Dr Cosmas Ilechukwu is specially appreciated on this occasion. You provided the needed leadership for me to grow spiritually and gave me the platform to serve God in various capacities which has attracted divine favour upon my life and family. Thank you sir. I also thank my Regional overseer, other State overseers, the State core group of CRM Rivers state (1&2), the Pastors and the entire brethren of the Charismatic Renewal Ministries for your great support and

steadfastness. We shall have revival cost what it may.
The Chairman of Nigerian Society of Engineers Port Harcourt branch Engr Dr H. Amadi, all past Chairmen of the branch and all professional colleagues, including the Chairman of NIMechE is greatly appreciated.
I have great pleasure expressing my deep appreciation to my dear Wife Mrs Mercy Donubari Lebele-Alawa and my beloved children Kilsi, Tonubari, Baribor, Kabari and Dunebari. Not forgetting my granddaughter (Eden). They gave me so much joy and unquantifiable encouragement. May God continually bless them.
I sincerely appreciate everyone who made out time today to attend this inaugural lecture. May God bless you all.
To God be all the glory for the great things He has done.

REFERENCES

Abdulrahman, A., Pericles, P. & Nawaf, A. (2016). Exergetic, Exergo-economic andExergo-environmentalAnalysis of Intercooled GasTurbine Engine. Paper presented at the 52nd propulsion energy Conference, Salt Lake City, Utah, USA. 11-25.

Adumene,S., **Lebele-Alawa,B.T.** (2015). Performance Optimization of Dual Pressure Heat Recovery System Generator (HRSG) in the Tropical Rainforest. *Engineering* **7**, 347-364.

Adumene,S., Le-ol,A.K., **Lebele-Alawa,B.T.** (2016). Performance and Economic Analysis of Gas Turbine Sub-System Generation in the Niger Delta. *International. Journal of Engineering and Technologies,* 9, 29-41.

Agboola, J., **Lebele-Alawa,B. T** & Nkoi,B. (2018). A Study of Combustion Temperature Distribution in The Cylinder of Compression Ignition Engine.*American Journal of Engineering Research,* 7(11), 304-312.

Ahaotu, J. O., Nkoi, B. &**Lebele-Alawa,B.T.** (2018). Thermodynamic Design and Economic Analysis of Ocean Thermal Energy Conversion for Coastal Nigeria. *American Journal of Engineering Research,* 7(12), 120-128.

Ahmadi, P., Hajadollahi, H. & Dincer, I. (2011). An Exergy-Based Multi-Objective Optimization of a Heat Recovery Steam Generator (HRSG) in a Combined Cycle Power Plant (CCPP) using Evolutionary Algorithm. *International Journal of Green Energy*, 8, 44-64

Aliu, S.A., and Ochornma, P.I. (2018). Exergoeconomic

Analysis of Ihovbor Gas Power Plant. *Nigerian Journal of Technology*, 37(4), 927-935.

Asuo,J. M.,**Lebele-Alawa,B.T**. & Nkoi,B. (2021). Exergo-Economic Analysis of 180MW Gas Turbine in the Niger Delta. *International Journal ofEngineering Research and Technology*, 10(11), 340-347

Asuo, J.M.,**Lebele-Alawa,B.T**. Sodiki, J. I. & Nkoi,B. (2021). Effects of CompressorBlade Profile Change on Thermo-Economic Performance of a Gas Turbine.*International Journal of Scientific and Engineering Research,* 12(11), 1165-1176

Ayodele, T. R. (2014). Feasibility Study of Stand-Alone Hybrid Energy System for Rural Electrification in Nigeria: The case study of Ala-Ajagbusi Community. *International Journal of Renewable Energy Resources*, *4*(1), 1-12.

Aziaka, D.S; Osigwe,E. O. &**Lebele-Alawa,B.T.** (2014). Structural and Conceptual Design Analysis of an Axial Compressor for 100MW Industrial Gas Turbine (IND 100). *World Journal of Mechanics***4**: 332-347.

Bejan, A; Tsatsaronis G; and Moran M. (1996). Thermal Design and Optimization,JohnWiley & Sons, Inc. USA.

Boyce, M.P (2002). Gas Turbine Handbook 2nd Edition. Gulf Professional Publishing, Oxford.

Cohen, H; Rogers, G.F.C; Saravanamutto, H.I.H (1998) *"Gas Turbine Theory"* 4th Edition. Addison Wesley Longman limited, England.

Egwanwo, V., **Lebele-Alawa, B.T.** (2012). Prediction of The Temperatue Distribution in a Shell and Tube Heat Exchanger Using Finite Element Model. *Canadian Journal of Mechanical Sciences and Engineering*,*3* (3):72-82.

Eke, M.N; Onyejekwe, D.C; Iloeje. O.C; Ezekwe, C.I; and

Akpan, P.U. (2018).Energy and Exergy Analysis of a 220MW Power Plant. *Nigerian Journal of Technology,* 37(1), 115-123.

Faango,K.K., **Lebele-Alawa,B.T.,** Nkoi,B. (2018). Adaptation of Ejector Refrigeration System to Gas Turbine Power Plant for PerformanceImprovement. *Global Scientific Journals,* 6(10), 10-17.

Gorji-Bandpy, M; Goodarzian, H; and Biglari, M. (2010). The Cost-Effective Analysis of a Gas Turbine Power Plant. *Energy Sources Part B: Economic, Planning and Policy,*

Güllen, S.C. (2019). *Gas Turbines for Electric Power Generation*. New York: Cambridge University Press.

Ganapathy. V (2001), Optimize Energy Efficiency of the Heat Recovery Steam Generator. Power Engineering Article, 105 (8).

Huadong,Y; Hong,X. (2013). The Effect of Surface Roughness on Thermodynamic

Performance Parameters of Axial Flow Compressor. *Journal of Applied Science, Engineering and Technology, 5(18), 4458-4463.*

Hussein I.B; Yusoff M.Z.B; and Boosroh M.H. (2011) Exergy analysis of a 120 MW Thermal Power Plant, in Proc. BSME-ASME Intern. Conf. on Thermal Engineering, Dhaka, Bangladesh,157-166

Igoma,E.N; **Lebele-Alawa,B.T;** Sodiki,J.I. (2016). Evaluation of the Influence of Ambient Temperature on the Performance of Trans-Amadi Gas TurbinePlant. *Journal of Power and Energy Engineering,* 4, 19-31.

Johnson, P.O;**Lebele-Alawa; B.T.** (2011). Performance Optimization of Oilfield Decanting Centrifuges. *International Research Journal inEngineering, Science and Technology,* 8(1), 28-32.

Lamb, C. M. (2005) Probabilistic Performance Based Geometric Tolerancing of Compressor Blades. M. Sc Thesis, Massachusetts Institute of Technology.

Lazzaretto, A., & Tsatsaronis, G. (2006). SPECO: A Systematic and General Methodology for Calculating Efficiencies and Costs in Thermal Systems. *Energy*, 32, 1257-1289.

Lebele-Alawa, B.T(2007). Effects of Compressor Blade Profile Change on Gas Turbine Performance. Ph.D. Thesis, Rivers State University of Science and Technology, Port Harcourt.

Lebele-Alawa, **B.T.,** Hart, H.I., Ogaji, S.O.T., Probert, S.D. (2008). Rotor-Blade's Profile Influence on a Gas-Turbine's Compressor Effectiveness. *Applied Energy*, *85*(6), 494-505.

Lebele-Alawa, B.T. (2009).On Rotor-Blade Deterioration and Pressure Losses in a Gas Turbine Plant. *Discovery and Innovation, 21*(1&2), 80-84.

Lebele-Alawa, B.T. (2010a). An Experimental Investigation of the Metal Concentrations of Dust Particles in Gas-Turbine Inlet Air. *Discovery and Innovation, 21,*(3&4), 74-77.

Lebele-Alawa, B.T. (2010b).Axial Thrust Responses to Gas Turbine's Rotor- Blade Distortions. *Journal of Engineering* Physics *and Thermophysics*, *83*(5), 991-994.

Lebele-Alawa, **B. T.** & Oparadike, F.E. (2011). Analysis of The Effects of ValvePropagated Pressure Surge On Pipe Flow. *Engineering SCIRP Journal,3*(11), 1098-1101.

Lebele-Alawa, **B. T.** (2001). The Development of Compressed Natural Gas as an Automotive Fuel in Nigeria. Proceedings of the International Conference of the Society of Petroleum Engineers (SPE), Abuja. 114-116

Lebele-Alawa, B.T., Asuo, J.M. (2011). Exergy Analysis of

Kolo-Creek Gas-Turbine Plant. *Canadian Journal of Mechanical Sciences and Engineering.2*(8),172-184.

Lebele-Alawa, **B.T.,** Asuo, J.M. (2011). Optimization of the System Performance of a Gas-Turbine Plant. *International Journal of Applied Science and Technology, 1*(6), 250-255.

Lebele-Alawa, **B.T.,** Egwanwo, V. (2012). Numerical Analysis of the Heat Transfers in Heat Exchangers.*International Journal of Applied Science and Technology*, *2* (4), 60-64.

Lebele-Alawa, **B. T.** & Ohia, I.O. (2013). Performance Evaluation of Heat Exchangers in a Polyethylene Plant. *International Journal ofEngineering and Technology Innovation,3*(1), 49-57.

Lebele-Alawa, **B.T.** & Asuo, J.M. (2013). Influence of the Variation of Power Turbine Inlet Temperature on Overall Turbine Efficiency. *International Journal of Engineering and Innovative Technology*, *2*(7), 226-229.

Lebele-Alawa, **B. T.** & Asuo, J.M. (2013). Performance analysis of a 20MW Gas Turbine Power Plant Using Energy and Exergy Methods' *Journal of Applied Science and Technology, 1*(1&2), 20-26.

Lebele-Alawa,B. T. &Ohia,I.O. (2014). Influence of Fouling on Heat Exchanger Effectiveness in a Polyethylene Plant. *Energy and Power,* **4**(2); 29-34.

Lebele-Alawa,B. T. &Jo-Appah, V. (2015). Thermodynamic Performance Analysis of a Gas Turbine in an Equatorial Rain Forest Environment. *Journal of Power and Energy Engineering,* 3(1):11-23.

Lebele-Alawa,B. T. & F.E.Oparadike (2015). Pressure Surge Dependence on Valve Operations in a Pipeline Loading System. *Engineering* **7**, 322-330.

Lebele-Alawa,B. T. &Le-ol,A.K. (2015). Improved Design of a25MW GasTurbine Plant Using Combined Cycle Application. *Journal of Power andEnergy Engineering,*3(8), 1-14.

Lebele-Alawa, B.T.&Hart,H.I. (2001). Design for Improvement of Flow Station Flare System Performance. *Nigeria Society of Engineers Technical Transactions*, *36*(2), 29-40.

Lebele-Alawa, B.T. (2001).On the Combustion of Associated Gas Effluents.*International Journal of Research in Science and Education, 1*(1), 108-115.

Lebele-Alawa, B.T. (2007). Effect of Compressor Blade Profile Change on Gas-Turbine Isentropic Efficiency. *International Research Journal in Engineering, Science and Technology,4* (2), 96-105.

Lebele-Alawa, B.T. (2009). Simulation of the Air Mass Flow Rate in a Fouled Axial Compressor. *International Research Journal in Engineering, Science and Technology, 6(1), 18-26.*

Lebele-Alawa. B.T. (2012). On Photovoltaic Systems as an Alternative Energy Source in Nigeria. *International research Journal inEngineering, Science and Technology, 9* (1), 62-66.

Loud, R.L, Slaterpryce, A. A. (2000), "Gas turbine inlet air treatment" GER 34 19 A Schenettady, New York.

Luedke, J. G. (2001) "Use of nonlinear voltera theory in predicting the propagation of non-uniform flow through an axial compressor". M.S thesis Virginia polytechnic institute and state university Blacksburg, V.A.

Massardo, A.F. & Scialo, M (2000). Thermoeconomic Analysis of Gas Turbine Based Cycles. *Journal of Engineering Gas Turbines and Power,* 122(4), 664-671.

Mondal, P. & Ghosh S. (2018). Integrated Biomass Gasification

Combined Cycle Plant for Small Scale Generation: Part B – Exergetic and Exergo-economic Analysis. *Journal of Material Science and Engineering,* 377(1), 1-6.

Moran, M. J. & Schiubba, E. (1994) Exergy Analysis: Principles and Practice.

Journal of Engineering Gas Turbines Power, 116(2), 285-290.

Nkoi,B. &**Lebele-Alawa,B.T.** (2015). Comparative Assessment of Combined-Heat-and-Power Performance of Small-Scale Aero-Derivative Gas Turbine Cycles. *Journal of Power and Energy Engineering,* 3,20-32.

Nkoi,B., **Lebele-Alawa,B. T.** & Odobeatu,B. (2018). Design and fabrication of a Modified Portable Biogas Digester for Renewable Cooking Gas Production. *European Journal of Engineering Research and Science*, 3(3), 21-29.

Nwabanne, J.T, Okoye, A.C., **Lebele-Alawa, B.T.** (2011). Packed Bed Column Studies for the Removal of Lead(II) Using Oil Palm Empty Fruit Bunch. *European Journal of Scientific Research*,63(2), 296-305.

Nwakpang,I. A.,**Lebele-Alawa,B.T**, &Nkoi,B. (2019). Performance Assessment of a Two-stage Reciprocating Air Compressor. *European Journal of Engineering Research and Science*, 4(4), 74-82.

Ochiche, O. S. & **Lebele-Alawa,B.T.** (2014). Improving CompressorPerformance to Reduce Loss Production Opportunities in Oil and GasProduction. *Journal of Mechanical Engineering and Automation*, 4(3):101-109.

Okeke, E. U. &**Lebele-Alawa, B.T** (2013). Modification Design of Petrol Engine *for Alternative Fueling Using Compressed Natural Gas (CNG). International journal of Engineering and Technology Innovation3*(2):86-94.

Ongiro, A; Ugursal, V.I; Al Taweel, A.M; and Walker, J. D.

(1997). Modelog of Heat Recovery Steam Generator Performance. *Applied Thermal Engineering*. 16 (5), 427 – 444.

Oyedepo, S.O., Richard, O.F., Samuel, S.A & Mahbub, A. (2015).Thermo-Economicand Thermo-Environomic Modelling and Analysis of Selected Gas Turbine Power Plants in Nigeria. *Journal of Science and Engineering,* 3(5), 423-442.

Rahman,M. M., Thamir,K. I. &Ahmed,N.A. (2012) Thermodynamic Performance Analysis of Gas Turbines Power Plant, *Journal of the Physical Sciences*, 6 (14)3539-3550.

Song, B. (2003).Experimental and Numerical Investigations of Optimized High-Turning Supercritical Compressor Blades. Ph.D. Thesis Virginia Polytechnic Institute and State University, Blacksburg, VA.

Ufot,E., **Lebele-Alawa**, **B.T.**, Douglas, I. E. & Bob-Manuel, K.D.H. (2010). A Non-Dimensional Consideration in Combustor Axial Stress Computations. *Engineering Scirp Journal,* 2(9), 733-739.

Ufot, E., **Lebele-Alawa**, **B. T.** & Douglas, I.E. (2011). On the Adiabatic Flame Temperature in Gas-Turbine Combustors. *European Journal of Scientific Research,64*(3), 387-393.

Ufot, E., **Lebele-Alawa, B. T.** & Bob-Manuel, K.D.H. (2011). Influence of Convection Heat Transfer Coefficient On Heat Transfers and Wall Temperatures of Gas-Turbine Combustors. *International Journal of Applied Science and Technology*, 1 (6), 210-218.

Ufot, E., **Lebele-Alawa**, **B. T.** & Douglas, I.E. (2012). Temperature and Thermal Stress Dependence on Fourier and Biot Numbers During Transient Heating of a Hollow

Cylinder. *Journal of Applied Science and Technology*, *17*(1&2), 102-106

Vahdati, M., Breard, C., Sayma, A, & Imregim, M,(2000). An Integrated Time-Domain Aeroelasticity Model for the Prediction of Rotor Forced Response Due to Inlet Distortion. Proceedings of ASME Turbo Expo 2000, Munich, Germany.

Valero, A., Lozano, M.A., Serra, L. & Torres, C. (1994). The CGAM Problem. *Energy*, 19(3), 365-381.