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Development of Model Equations for a Gas Turbine Rotor System for Fault Detection

Angelyna Amos[1] E Kotingo Kelvin[2]

Department of Mechanical Engineering

Niger Delta University, Wilberforce Island, Bayelsa State Nigeria

Abstract: *The rotor shaft is a major mechanical component of the gas turbine engine responsible for power generation and power. Failure of the rotor shaft system is catastrophic with long down times and high maintenance cost as the consequences. Therefore, the aim of this research is to develop a model-based software for Torsional Vibration analyses of gas turbine rotor shaft system for early fault detection including artificial intelligence. The purpose is to predict and detect early faults that can cause catastrophic failures and stoppage of work. This paper, therefore, presents a computer based analytical solution method for Torsional vibration analysis of the gas turbine engine. In arriving at the method, a single degree of freedom system was modeled and relevant equations were derived. These equations were programmed using Java programming technique and results were compared with the data of an operational existing plant. From the results it is seen that the value for the highest possible fault is 0.909 on the y axis. In this process, it was discovered that input parameters such as air flow rates of 21.9Kg/s, fuel flow rate of 0.324Kg/s, and heat value of fuel at 11750kJ/kW-hr would yield thermal efficiency as high as 51% and an optimum operating point of 298K. The software developed will be useful for training in tertiary institutions and easy for both technical and non-technical personnel on both thermodynamic and Torsional Vibration analysis of Gas Turbine rotor shaft.*

Keywords: Gas Turbine, Rotor Shaft, Torsional Vibration Analysis, Catastrophic Failures, Thermodynamic, Compressor

INTRODUCTION

The rotor shaft is a major mechanical component of the gas turbine engine responsible for power generation and power transformation. According to Amos (2007) reliability and performance of the system is mostly determined by the efficiency of the rotor system. Failure of the rotor shaft system is catastrophic with long down times and high maintenance cost as the consequence. The rotor of gas turbine engine is the assembly of blades, discs and shafts. The compressors and turbines of the GT are periodically affected by the action of vibration (Ogbonnaya, et al 2013a). The rotor is very important and responsible for the absorption of pressure, energy transformation into mechanical energy and further transformation to electrical energy. The problems of imbalance, misalignment, eccentricity and looseness are common to gas turbine (GT) rotor and are capable of causing catastrophic failures and down times. The signs, symptoms and failure of these faults can only be identified, detected and determined by thermodynamic and vibration analysis techniques. Vibration monitoring involves measurements, recording, analysis, alarming and predicting the state of machinery, component and their subcomponent. A vibration is characterized and assessed by three parameters, namely; Amplitude, Frequency and Phase. Amplitude is the maximum displacement from the central position; frequency is the reciprocal of the period which is the time for one cycle of vibration and is expressed in cycles/s while phase is a measure of the instance at which a vibration passes through the central position.

The operating life of a rotor shaft, breakdown prevention, productivity accuracy and performance of GT plants depends on the level of rotor vibrations and pressure values between the rotor and bearing. Maintenance is the process or art of restoring and prolonging the life and operation of a component or system. Over the years, maintenance programme such as routine checks, preventive maintenance and complete overhaul (corrective) of systems at breakdown were applied and benefits achieved. Thereafter it was discovered that certain expenditures can be eliminated or reduced by predicting failures (condition monitoring) and scheduling a maintenance programme at the right time (proactive maintenance). Proper implementation of preventive maintenance and condition monitoring programmes has also reduced capital expenditures by 25% (Ogbonnaya, et al 2013a). The power plant industry is therefore adopting proactive maintenance policies that change the reactive maintenance culture to proactive programme. It includes vibration monitoring activities and root cause analyses of major systems and components.. Due to the complexity of GTE, it is essential to model the processes inside the GT to detect faults. Models were required to predict performance of the Joule-Brayton cycle and the performance of individual components that takes place in the GT, This paper is intended to employ thermodynamic, vibration monitoring techniques and come up with a proactive model for fault detection of a GT plants used in Nigeria.

OBJECTIVES

i)To develop models capable of detecting early signs and symptoms that could generate catastrophic failures.
ii) Detection of early fault that can lead to catastrophic failures.

iii) Extend life of plant.

MATERIALS AND METHODS

The gas turbine engine (GTE used) for this work consists of compressor rotor assembly, turbine rotor assembly and the generator with a power output of 125 MW. It is a single shaft system connected together by couplings, rotating at a constant speed of 3000rpm. It is designed to trip at 3300rpm and at velocity amplitude of 18mm/s. Three proximitors of Bently Nevada make are mounted at 45^0 on the bearing casings to measure bearing vibration and six seismic vibration sensors for measuring shaft deflection. Schematic of bearing/sensors is shown in figure 3.1 while particulars of the plant used for the analysis is shown in table 3.1. The operating parameters considered are the vibration amplitude, turbine speed and load. These factors are important to know the effect of vibration on the speed and load which in turn give the condition or status of the plant.

Table 3.1 Particulars of Niger Delta Power Holdings Company of Nigeria (NDPHCN) Gbarain GT2

Particulars	Specifications
Name of equipment	Turbine 2
Manufacturer	GE frame 9
Capacity	125MW
Year of manufacture	2006
Year of installation	2016
Number of turbine stages	3
Number of compressor stages	17
Length of turbo-compressor rotor	5.692m
Mass of turbo- compressor rotor	49168 kg
Nominal Diameter of rotor	1.32m
Spring stiffness constant K	20GN/mm
Damping coefficient D	3502 N s/m
Nominal Speed	3000 rpm
Mass moment of Inertia	22071.5 kg m2

Data is collected from GT2 NDPHCN Gbarain, Bayelsa State, a power plant for electric power generation. The data obtained from the station are average values taken hourly; data was extracted for 10 days shown in tables Appendix A. For easy analysis, the turbo-compressor rotor assembly was chosen as SDF system for the modeling process.

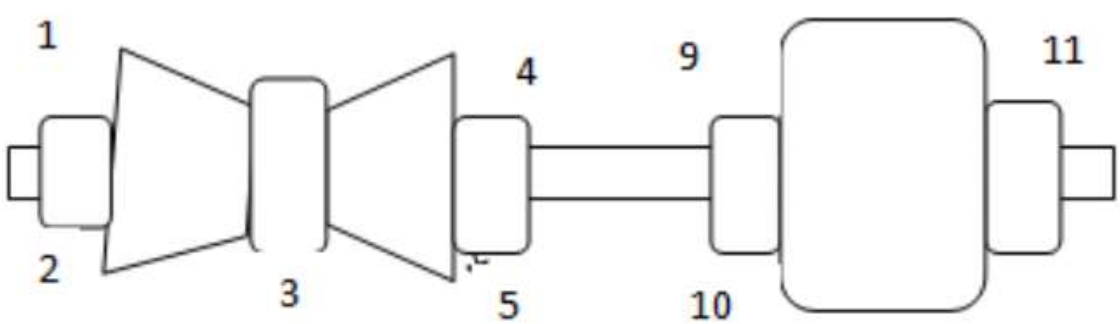

Fig. 3.1: Schematic of bearings/sensors location on GTsystem

3.2 Development of model equations

According to Kameswara 2006, TV analysis of GT rotor shaft involves four structured procedures as follows (i) Modeling of the system (ii) Derivation of equation (iii) Solution of equation (iv) Comparison and validation of solution. In this work the four steps are presented using existing power plants with particulars shown in table 3.1.

3.2.1 Rotor system monitoring

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Modeling a system is the representation of the system dynamics mathematically and/or graphically. For this work, the turbo compressor rotor of a GE frame 9 GT, a single shaft system rotating at 3000 revolutions per minute (rpm) in NDPHCN is used for the analysis.

The following assumptions were adopted for modeling process.

- The system is a single degree of freedom (SDF) unit, that is, only one independent
- coordinate is enough to locate the position of the system.
- It is also assumed that the system is linear. The weight of the turbo compressor is a lumped mass held by bearings at both ends of the assembly.
- Damping of system is assumed viscous only.
- System is running at a constant speedω.

For the rotor system considered, the parameters involved are the mass moment of inertia, damping capability and the elastic/stiffness nature of the rotor shaft system. These are the system's properties that can affect the responds to vibration of the system. Therefore, the forces acting on the assembly are the inertial force due to the masses on the system, elastic force and springiness of the shaft, the damping force due to fluid viscosity in contact at the bearings. Its ability to dissipate heat energy and the frictional effects on the GT bearings. These forces act in opposite direction to restore the system to equilibrium. The system is also disturbed by an external torque *T*. In vibration analysis, the position of mass at any time *t* is very important so the mass is modeled as an element proportional to the acceleration, damping element is proportional to velocity and the stiffness element is proportional to the position of the mass (Ogbonnaya and Koumako, 2020).

Applying Beam theory (Newton's second law of motion)

$$\sum F = ma$$

$$I\ddot{\theta} + D_T\dot{\theta} + K_T\theta = \mathrm{T} \qquad 3.1$$

Consider the rotor system having two disc representing the compressor and turbine assemblies the GTE with the compressor stages lumped as a single mass system of inertia and the turbine stages lumped as another mass or inertia system is connected together by an elastic shaft of spring constant K_T. These rotor shaft bearings have a damping coefficient D_T, rotating with an angular speed ω and moving in the vertical direction as shown in figure 3.2, presented equations 3.2 and 3.3

$$I_1(\theta_1 - \theta_2) + D(\theta_1 - \theta_2) + K\theta_1 - \theta_2 = T \qquad 3.2$$

$$I_2(\theta_2 - \theta_1) + D(\theta_2 - \theta_1) + K\theta_2 - \theta_1 = T_2 \qquad 3.3$$

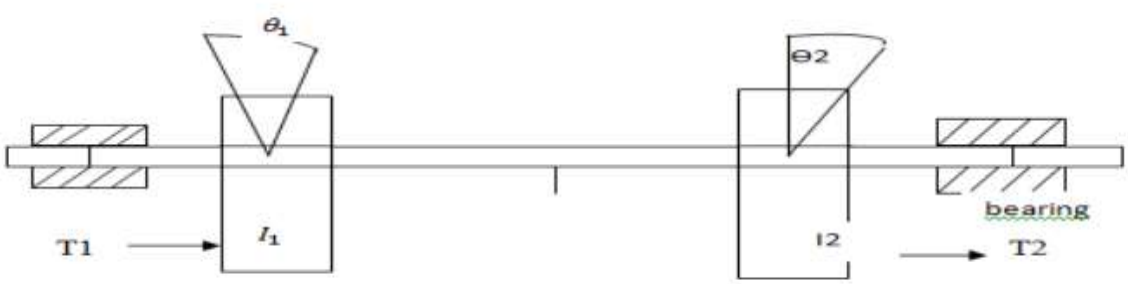

Fig.3.2: Two rotor shaft system (Ogbonnaya and Koumako, 2020)

Each mass system is being acted upon by an external torque $T_1$and T_2 and the relative angular displacement of each inertia is $\theta_1 and \theta_2$ .The parameters considered in the modeling of this system are the inertia force, the damping force, and elastic force of the shaft. Each rotor mass system behaves as a SDF system

$$I_1 = I_2, \; T_2 - T_1 = T, \; \theta_2 - \theta_1 = \phi$$

For the SDF system, applying the assumptions stated above and substituting these values into equations 3.2 and 3.3, the equation of motion of system is reduced to equation 3.4.

$$I\ddot{\varphi} + D\dot{\varphi} + K\varphi = T \qquad 3.4$$

Where,

$I = Mr^2$

M=Mass moment of inertia kg-m^2

D=damping coefficient (Ns/m),

K_T = spring constant $\left(\frac{\text{N}}{\text{m}}\right)$

T = exciting torque =mrω^2 (Nm/s)

Newton's second law of motion expressed for an undamped system.

$$I_1\ddot{\theta}_1 = K_1(\theta_2 - \theta_1) \quad 3.5$$

$$I_2\ddot{\theta}_2 = -K_1(\theta_2 - \theta_1) + K_2(\theta_3 - \theta_2) \quad 3.6$$

$$I_3\ddot{\theta}_3 = -K_2(\theta_3 - \theta_2) \quad 3.7$$

For

$$\theta_1 = A_1 sin\omega_n t$$

$$\theta_2 = A_2\ sin\omega_n t$$

$$\theta_3 = A_3\ sin\omega_n t$$

Taking $\theta_2 - \theta_1$, and $\theta_3 - \theta_2 = \emptyset_1$ and $\emptyset_2$

$$\dot{\theta} = \frac{d\theta}{dt} = \omega_n cos\omega_n t$$

$$.\ \ddot{\theta} = \frac{d\dot{\theta}}{dt} = -\omega_n^2 \sin\omega_n t$$

Substituting values of θ into equation 3.6 to 3.7 we have

$$-I_1\omega_n^2 A_1 = K_1\left(A_2 - A_1\right) \quad 3.8$$

$$-I_2\omega_n^2 A_2 = K_1\left(A_2 - A_1\right) + K_2\left(A_3 - A_2\right) \quad 3.9$$

$$I_3\omega_n^2 A_3 = K_2\left(A_3 - A_2\right) \quad 3.10$$

$$A_1 = \frac{K_1 A_2}{K_1 - I_1\omega_n^2} \quad 3.11$$

$$A_3 = \frac{K_2 A_2}{K_2 - I_3\omega_n^2} \quad 3.12$$

Substituting values of A_1, and A_3into equation 3.9

We get

$$-I_2\omega_n^2 A_2 = -K_1 A_2\left(1 - \frac{K_1}{K_1 - I_2\omega_n^2}\right) + K_2 A_2\left(\frac{K_2}{K_2 - I_3\omega_n^2} - 1\right) \quad 3.13$$

$$\omega_n^4 - K_1\left(\frac{1}{I_1} + \frac{1}{I_2}\right) + K_2\left(\frac{1}{I_2} + \frac{1}{I_3}\right)\omega_n^2 + \frac{K_1K_2}{I_1 I_2 I_3},\left(I_1 + I_2 + I_3\right) = 0 \quad 3.14$$

For a free undamped TV system

$$let\lambda = \omega_n^2$$

$$\therefore \lambda^2 - k1\left(\frac{1}{I_1} + \frac{1}{I_2}\right)\lambda\frac{K_1K_2}{I_1 I_2 I_3},\left(I_1 + I_2 + I_3\right) = 0 \quad 3.15$$

$$\lambda = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

3.3 Derivation of TV response equation

For a simple harmonic motion, the position vector signal is given by *Asinω*t, differentiating twice we get the velocity and acceleration signals as thus

$\emptyset = A\ sin\ \omega t$ Angular displacement amplitude(mm)

$\dot{\emptyset} = \omega A \cos \omega t$ Angular velocity amplitude(deg/s)

$\ddot{\emptyset} = -\omega^2 A \sin \omega t$ Angular acceleration amplitude (deg/s^2)

Substituting these values into equation 3.3 with some algebraic manipulations we get

$$\Phi = \frac{T}{\sqrt{((K - I\omega^2)^2 + (D\omega)\text{^}2)}} \quad 3.16$$

$$\Phi = \frac{T}{\sqrt{((K - I\omega^2)^2 + (D\omega)\text{^}2)}}$$

$$\theta = tan^{-1}\frac{D\omega}{K - I\omega^2} \quad 3.17$$

$$\dot{\varphi} = \omega\varphi \quad 3.18$$

$$\ddot{\varphi} = -\omega^2\varphi \quad 3.19$$

Equation 3.16 is used for the computer programming. Before the computer program was developed an algorithm on how the device works was drawn. All the important plant parameters, plant particulars and data are used for the developing the program is presented in table 3.1 and its vector force diagram is shown in figure 3.3. It has provision for alert conditions for shut down when the set point

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is exceeded and repairs if necessary. The system uses 8mm/s and the resonance ωn/critical speed for the safety alarm alert and shut down measures according to ISO10816 2003 for the operation of GT engines.

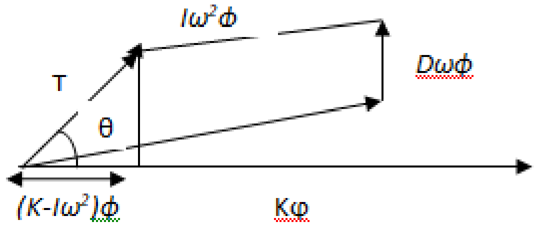

Fig.3.3: Vector force diagram (Ogbonnaya 2009)

The equation of forces acting on the rotor shaft system is written thus:

$$I\ddot{\vartheta} + D_T\dot{\vartheta} + K\vartheta = Tosin\omega t \qquad 3.20$$

The solution to equation 3.9 consists of two parts; the complimentary part ϑc and the particular integral ϑp. The solution can be written as

$$\theta = \theta_C + \theta_P \qquad 3.21$$

The complimentary function is obtained by setting equation 3.9 to zero

$$I\ddot{\theta} + D_T\dot{\theta} + K_T\theta = 0 \qquad 3.22$$

thus, the complimentary solution becomes

$$\theta c = Be^{-\alpha t} + \sin(\omega nt - \phi) \qquad 3.23$$

$$\theta p = A_1 Sin\omega t + A_2 cos\omega t \qquad 3.24$$

$$\dot{\theta} = \omega A_1 cos\omega t - \omega A_2 sin\omega t \qquad 3.25$$

$$\ddot{\theta} = \omega^2 A_1 sin\omega t - \omega^2 A_2 \cos\omega t \qquad 3.26$$

Substituting equations 3.24, 3.25 and 3.26 into equation 3.20 and rearranging gives $(K_T - I\omega^2)(A_1 sin\omega t + A_2 cos\omega t) + D_T(A_1 cos\omega t - A_2 sin\omega t) = Tosin\omega t$ng coefficients of *sinωt* and *cosωt* on the right and the left hand side separately yields $(K_T - I\omega^2)A_1 - D_T\omega A_2 = To$ 3.28

$$(K_T - I\omega^2)A_2 + D_T\omega A_1 = 0 \qquad 3.29$$

From equation 3.29, making A_2 the subject of the formula and substituting into equation 3.28 we get

$$(K_T - I\omega^2)^2 A_1 + (D_T\omega)A_1 = To(K_T - I\omega^2)^2 \qquad 3.30$$

$$A_1 = To\frac{(K_T - I\omega^2)}{\{(K_T - I\omega^2)^2 + (D_T\omega)^2\}} \qquad 3.31$$

$$A_2 = \frac{To(-D_T\omega)}{} \{(K_T - I\omega^2)^{2A_1} + (D_T\omega)^2\} \qquad 3.32$$

The particular integral of the differential equation (3.25) is

$$To(K_T - I\omega\text{^}2)sin\omega t/\{(K_T - I\omega^2)\text{^}2A_1 + (D_T\omega)\text{^}2\} +$$
$$To(-D_T\omega)\, cos\omega t/\{(K_T - (I\omega^2)\text{^}2A_1 + (D_T\omega)\text{^}2\} \qquad 3.33$$

$$\theta p = To\{(K_T - I\omega^2)(\sin\omega t - \cos\omega t)\}/[(K_T - I\omega^2)^2 + (D_T\omega)^2 \qquad 3.34$$

For

$$D_T\omega = \theta sin\phi and\ (K_T - I\omega^2) = \theta cos\phi \text{fore,}$$

$$\theta = To \sqrt{}\{(K_T - I\omega^2)^2 + (D_T\omega)^2] \qquad 3.35$$

$$tan\phi = D_T\omega - (K_T - I\omega^2) \qquad 3.36$$

Hence, the particular integral solution becomes:

$$\Theta p = To * \theta \sin(\omega t + \phi)\{(K_T - I\omega^2)^2 + (D_T\omega)^2\} \qquad 3.37$$

Conversely, the complete solution of equation 3.2 becomes

$$\Theta = Be^{-\alpha t} + sin(\omega nt - \phi) + (To*\theta sin(\omega t + \phi))/\{(K_T - I\omega^2)\text{^}2 + (D_T\omega)\text{^}2 \qquad 3.38$$

This implies that the complimentary function is small compared to the particular integral. Therefore, the angular displacement ϴ at any point is given by the particular integral, ϴp only. Hence, the amplitude of forced TV is given as:

$$\theta = \frac{To}{\sqrt{\{(K_T - I\omega^2)^2 + (D_T\omega)^2\}}} \qquad 3.39$$

Also the exciting torque causing the forced TVcan be calculated as

$$To = \theta\sqrt{[(K_T - I\omega^2)^2 + (D_T\omega)^2]} \qquad 3.40$$

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Equation 3.39 is used for the Java computer program to calculate the TV angular displacement amplitude, angular velocity *(Θω)* and angular acceleration *(Θω²)* amplitudes respectively. This is shown in Appendix B.

3.4 Development of thermodynamic mathematical model for GTE

The system behavioral performance are detected with the use of mathematical models. These mathematical relations in combination with certain constitutive relations are used for modeling the GT system (Ailer et al, 2001). The mathematical model of the GT system for this study is developed in accordance with thermodynamic principles for the system, the continuity and energy balanced equations.

3.4.1 Modeling equations and assumption

For the purpose of this study the following assumptions were made

- An ideal gas is used
- Adiabatic compression takes place.
- The turbine inlet and outlet are at the same altitude.
- Throughout process 1-2 and 4-1, the specific heat capacity C_p is constant.

3.4.2 Modeling the compressor

The compressor is a major component of the GTE, it sucks in air and compresses it, the action of compressing the air heats it up and hot, high pressure air moves to burner (combustion chamber), mixed with fuel and ignited. Centrifugal and axial compressor are the two types of compressors.

The axial compressor based on its working principles continuously pressurizes gases with its air foil component in which the working fluid flow to the axis of rotation in parallel while

the centrifugal compressor comprises of a radial component in which the working fluid flows through. Our model utilizes the axial compressor type. From the static energy balance equation, the compressor work is given as the change in enthalpy of fluid within the compressor.

$$W_c = h_2 - h_1 \qquad 3.41$$

Where wc is the specific work in $\frac{\mathrm{J}}{\mathrm{Kg}}$ provided to the compressor, h_1 is the fluid specific enthalpy at the inlet of the compressor, h_2 is the fluid specific enthalpy at the outlet

of the compressor.

The rise in specific enthalpy is calculated as:

$$h_2 - \mathrm{h}_1 = C_P.(T_2 - T_1) \qquad 3.42$$

Therefore

$$W_{12} = C_P.(T_2 - T_1) = C_P.T_1.\left(\frac{T_2}{T_1} - 1\right) \qquad 3.43$$

(Rahman et al, 2011)

In a real cycle where we take into consideration the isentropic efficeincy of actual compressor work is calculated using Equation 3.44

$$W_{Ca} = C_P.(T_{2a} - T_1) \qquad 3.44$$

From Isentropic temperature and pressure relation is as shown in equation 3.45

$$\frac{T_{2i}}{T_1} = \left(\frac{P_1}{P_2}\right)^{\frac{\Gamma-1}{\Gamma}} = (rp)^{\frac{\Gamma-1}{\Gamma}} \qquad 3.45$$

The expression for the isentropic efficiency of the compressor is stated in equation 3.46

$$\eta_{Ci} = \frac{W_{Ca}}{W_{Ci}} = \frac{T_{2i}-T_1}{T_{2a}-T_1} \qquad 3.46$$

Therefore, the actual temperature at the exit of the compressor is given as

$$T_{2a} = T_1 + \frac{T_{2i}-T_1}{\eta_{Ci}} \qquad 3.47$$

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RESULTS AND DISCUSSIONS

The plots are developed from the performance results generated from the program shown in Appendix A. The parameter influence in terms of air to fuel, TIT, compression ratio, and ambient temperature on the performance of GT cycle power plant are discussed in this chapter.

4.1 Effect of thermal efficiency and pressure ratio on FDIA

The thermal efficiency is a prime factor in GT performance, it is given as the ratio of work produced by the engine to the energy or heat supplied by the fuel. figure 4.3 presents a relation between the GT cycle thermal efficiency versus pressure ratios, it is observed in figure 4.1that at higher pressure ratios of 18 and above the overall thermal efficiency of the system increases up to 55% this causes an increment in the turbine inlet temperatures which might be harmful to the turbine blades.

Figure 4.2 shows the effect of pressure ratio on the GT cycle thermal efficiency with increasing ambient temperatures. The thermal efficiency increases with compression ratio at increased ambient temperature, the dash line indicates an ambient temperature of 288K at this temperature the thermal efficiency is relatively low. At low pressure ratios of 10-15 the variation in thermal efficiency at different temperature is not as significant as obtained at higher pressure ratios. Change in ambient temperature does not significantly that affect the thermal efficiency of the cycle as much as the rise in compression ratio which will increase the rate of GTE breakdown.

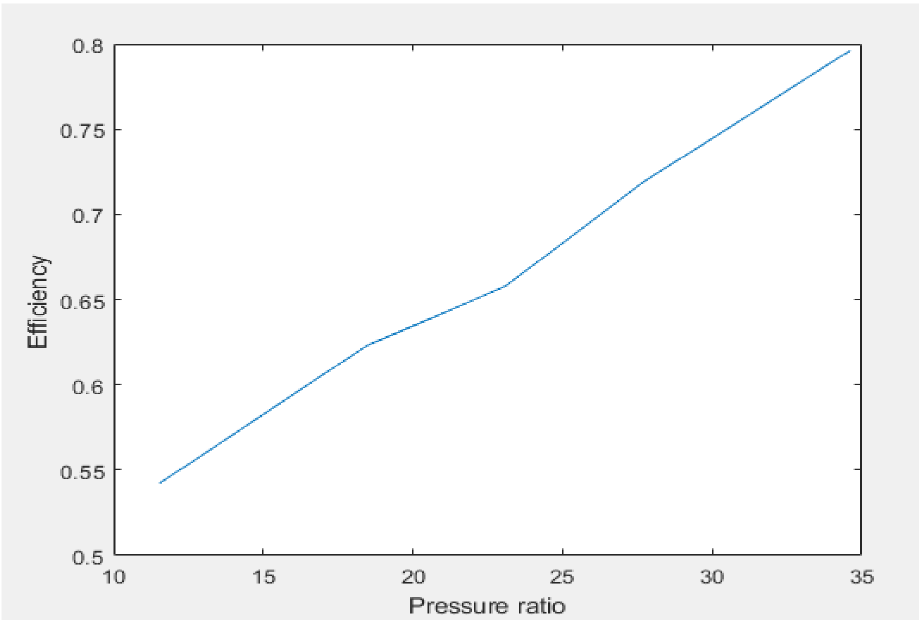

Fig. 4.1: Pressure ratio versus thermal efficiency

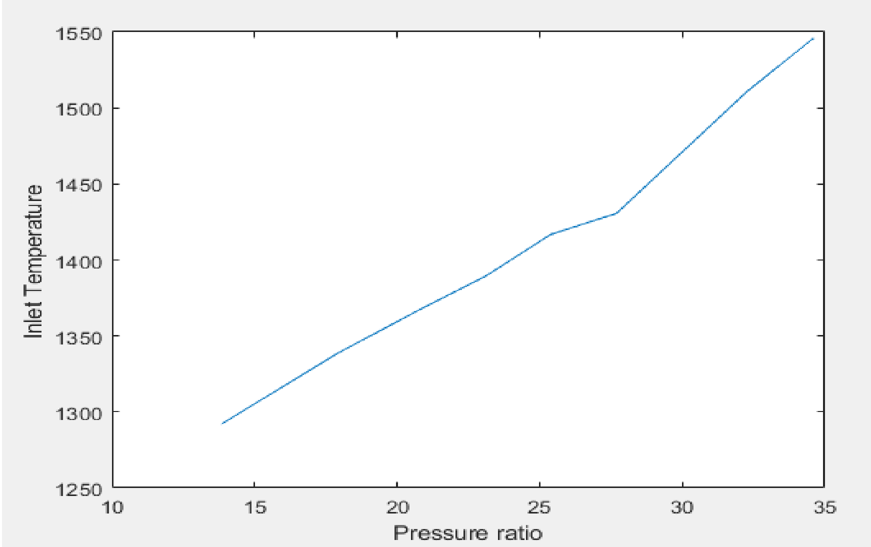

Fig. 4.2:Pressure ratio versus turbine inlet temperature

CONCLUSION

The present research work adopted the statistical standard deviation (σ) method by comparing the values obtained using Javacomputation technique and the data from the plant under consideration. Inlet temperatures, efficiency and pressure ratio results were shown while torsional displacement, velocity and acceleration were obtained for a speed range of 3000 and 3050rpm. Considering the assumptions taken in the modeling process, operational factors and environmental factors, the difference is within the set values and acceptable. The software is useful for TVA of all rotors dynamic systems. The graphs and procedure of analysis were compared with the work presented by Ameen (2014), for the construction of graphical user interface using Holzer and Java technique and were similar and acceptable.
Every mechanical component or assembly vibrates. The amplitude of vibration is related to the frequency or speed. It is the reason why the amplitude, resonance or natural frequency are used as control to the software, for instance using equation 3.28 at very low frequencies well below resonance the TVθ, will approximately be in phase with the forcing torque and at zero frequency the amplitude becomes a steady state twist equal to T/K_T
At resonance $\omega = \omega_n$, the amplitude of vibration will peak sharply because of low damping stiffness and the phase of response will lag the phase of the forcing torque by about 90^o at this point shear stresses are built up on the torsional load stresses and a combination of these stresses can lead to fatigue crack. Therefore, this region is avoided completely.

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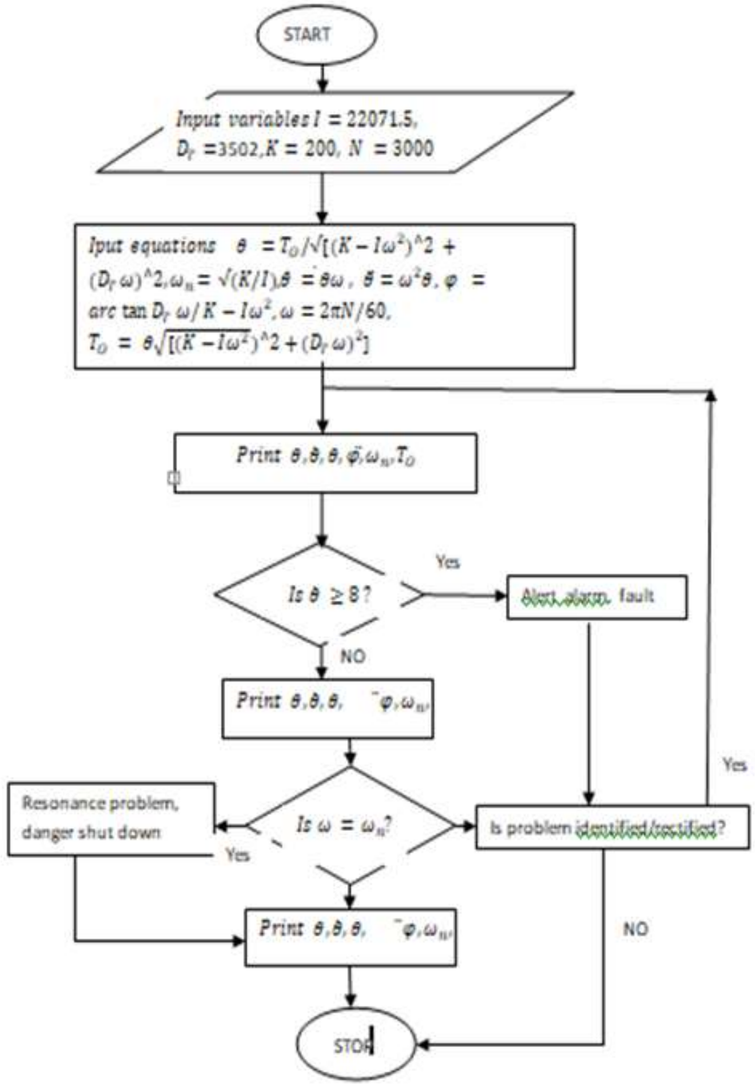
APPENDIX A1



Flow chart to compute vibration amplitude of model GT

APPENDIX A2
JAVA CODE FOR RESULTS IMPLIMENTATION

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```
/*
 * To change this license header, choose License Headers in Project Properties.
 * To change this template file, choose Tools | Templates
 * and open the template in the editor.
 */
package solver;

/**
 *
 * @author Chinedu
 */
public class Cw extends javax.swing.JFrame {

    /**
     * Creates new form
     */
    public Cw() {
        initComponents();
    }

    /**
     * This method is called from within the constructor to initialize the form.
     * WARNING: Do NOT modify this code. The content of this method is always
     * regenerated by the Form Editor.
     */
    @SuppressWarnings("unchecked")
    // <editor-fold defaultstate="collapsed" desc="Generated Code">//GEN-BEGIN:initComponents
    private void initComponents() {

        jLabel1 = new javax.swing.JLabel();
        jLabel2 = new javax.swing.JLabel();
        jLabel3 = new javax.swing.JLabel();
        jLabel4 = new javax.swing.JLabel();
        jLabel5 = new javax.swing.JLabel();
        jButton1 = new javax.swing.JButton();
        jButton2 = new javax.swing.JButton();
        ma = new javax.swing.JTextField();
        cit = new javax.swing.JTextField();
        pr = new javax.swing.JTextField();
        ga = new javax.swing.JTextField();
        jPanel1 = new javax.swing.JPanel();
        jLabel6 = new javax.swing.JLabel();
        answer1 = new javax.swing.JLabel();
        answer2 = new javax.swing.JLabel();
        jPanel2 = new javax.swing.JPanel();
        jButton3 = new javax.swing.JButton();
```