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Impact of Modeling in Fault Detection and Identification Analysis (FDIA) of a Gas Turbine

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Abstract*: The impact of modeling in fault detection has been studied in this paper. Failure of the rotor shaft in a gas turbine plant could be problematic due to downtimes and repair cost. Therefore, the development of a model based software for torsional vibration analysis of a gas turbine system to identify impending faults and artificial intelligence can't be over emphasized. Furthermore, a degree of freedom system was modeled; the derived equations were subjected to java programming methods and results compared to an operational plant data. The results showed that the value for possible highest fault is 0.909 on the y axis. 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 of 51% and a maximum operating point of 298K.*

Keywords: Rotor Shaft, Torsional Vibration Analysis, Catastrophic Failures,

INTRODUCTION

Every component part or structure of mechanical system vibrates due to loads and pressure. Vibration can be defined as the oscillatory motion of mechanical systems about a central position. Vibration can be rectilinear or torsional. Torsional Vibration (TV) is the periodic motion of angular elastic shafts with rigid attachment circular rotors same as gas turbine (GT) rotors which is basically an assembly of blades, shafts, disc etc. The function of the rotor is to absorb pressure and energy, in other words transform mechanical energy into electrical energy as well as transformation of other technological processes. Failures and break downtimes occur frequently due to factors such as poor maintenance, misalignment, which are commonly embedded in GT rotor systems. The signs, symptoms that causes these faults can only be identified through and vibration and thermodynamic analysis techniques. The importance of vibration analysis is to schedule operations and maintenance departments to design and implement effective programmes that will solve system problems. It is on this note that vibro activity is built into the design, operation and maintenance programme of the gas turbine machinery which involves recording, measuring, system analysis and failure rate prediction.

OBJECTIVES

The increase in failure and long down times of GT engines due to vibration operating in Nigeria is a serious concern. It is on this ground that the rotor shaft system is taken as a research work for vibration and thermodynamic analysis. The followings are the objectives of this research work.

i. To develop models that can identify early signs of failures and downtimes.

ii. The use of thermodynamic and TV monitoring as means of fault detection.

iii. The model can also be used as proactive measure for power plant operation. Integrating these models into power plant operation shall increase reliability, availability and productivity.

iv. To extend life of plant.

METHODOLOGY

Modeling is required to create a clear and concise performance behavior of the system before construction. In engineering applications, the representation of objects with equations is referred to as mathematical modeling. System modeling is useful to understand, monitor, predict and control system which is the reason for this research work. This chapter, therefore, deals with the development of proactive models for early fault detection in the gas turbine engine. This involves modeling of the major components whose failure is catastrophic using torsional vibration and thermodynamic principles. The solution method is computer based and the results are compared with that of an operational existing GT. Table 3.1. presents Particulars of Niger Delta Power Holdings Company of Nigeria (NDPHCN) Gbarain GT2 used in the analysis.

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 m^2

3.1 Modeling the combustion chamber.

The model utilizes an annular type of combustion chamber, asides its weight and cost advantage the annular combustion chamber is preferred over the multiple combustion chamber and the turbo-annular combustion chamber types due to its elimination of combustion propagation problem from chamber to chamber. An electric spark is needed. Thereafter the ignited flame must sustain continuously, the air leaving the compressor at T_2 and P_2 enters the combustion chamber, mixes with fuel and is ignited, not all the air is burnt, a fraction of air is used for cooling the flame before it reaches the nozzle guide vanes. The heat added in the combustion chamber is given as:

$$Q_s = \dot{M}_f * H_f = m_a C_p (T_3 - T_{2a}) \qquad 3.1$$

3.2 Modeling the turbine.

According to Saravanamuttoo, et al (2013) the hot gases leaving the combustor are passed to the turbine, which acts like a series of windmill with the nozzle guide vanes directing the hot gases from the combustor into rows of rotating turbine blades. These blades are attached to large discs which are directly connected to the compressor. The turbine in this model is a three stage turbine it utilizes the combustion system energy to drive the compressor, accessories and the alternator for power generation. As the hot gases exit the combustion chamber, it hits on the turbine blade, the turbine inlet temperature is given as:

$$T_3 = \frac{\dot{M}_F * H_F}{\dot{M}_A * C_{Pa}} + T_{2a} \qquad 3.2$$

Also from isentropic P-T Relation

$$\frac{T_3}{T_{4i}} = \frac{P_3}{P_4}^{\frac{\Gamma - 1}{\Gamma}} \qquad 3.3$$

$$T_{4i} = T_3 . R_P^{\frac{1-\gamma}{\Gamma}} \qquad 3.4$$

$$T_{4a} = T_3 - \eta_{Ti}(T_3 - T_{4i}) \qquad 3.5$$

The isentropic and actual work done by the turbine are expressed in Newton’s second law of motion expressed for an undamped system

$$W_{Ti} = (h_3 - h_{4i}) = c_P . (T_3 - T_{4i}) \qquad 3.6$$

$$W_{Ta} = (h_3 - h_{4a}) = c_P . (T_3 - T_{4a}) \qquad 3.7$$

The Network generated by the system is given as

$$w_{net} = w_{ta} - w_{ca} \qquad 3.8$$

The thermal efficiency of the system is expressed as

$$\eta_{\text{tha}} = \frac{w_{net}}{Q_{in}} = \frac{w_{ta} - w_{ca}}{Q_{in}} \qquad 3.9$$

The Powered delivered to the system is

$$P_d = w_{net} * \dot{M}_a \qquad 3.10$$

(Rahman et al, 2011), Where;

$H_{1=}$ Enthalpy at the inlet compressor (kj/kg)
$H_{2=}$ Enthalpy of theat the outlet compressor (kj/kg)
C_p= Specific heat of air at constant temperature (kj/kg/k)
T_1= Ambient air temperature entering the compressor (^{o}k)
$T_{2=}$ Fluid temperature leaving the compressor (^{o}k)

T_{2a} = Actual fluid temperature leaving the compressor (^{o}k)
M_a = Air flow rate (kg/s)
M_f = Fuel flow rate (kg/s)
H_f= Heat value of fuel (kj/kg.k)
Ts= Inlet turbine temperature and maximum temperature of the system (k)
T_{4i}= Isentropic temperature of fluid leaving the turbine (^{o}k)
T_{2a}= Actual temperature of fluid leaving the turbine (^{o}k)
W_c = W_{ci} = isentropic compressor work (kW)
W_{ca} = actual compressor work (kW)
W_{ti}= isentropic turbine work (kW)
W_{ta} = actual turbine work (kW)

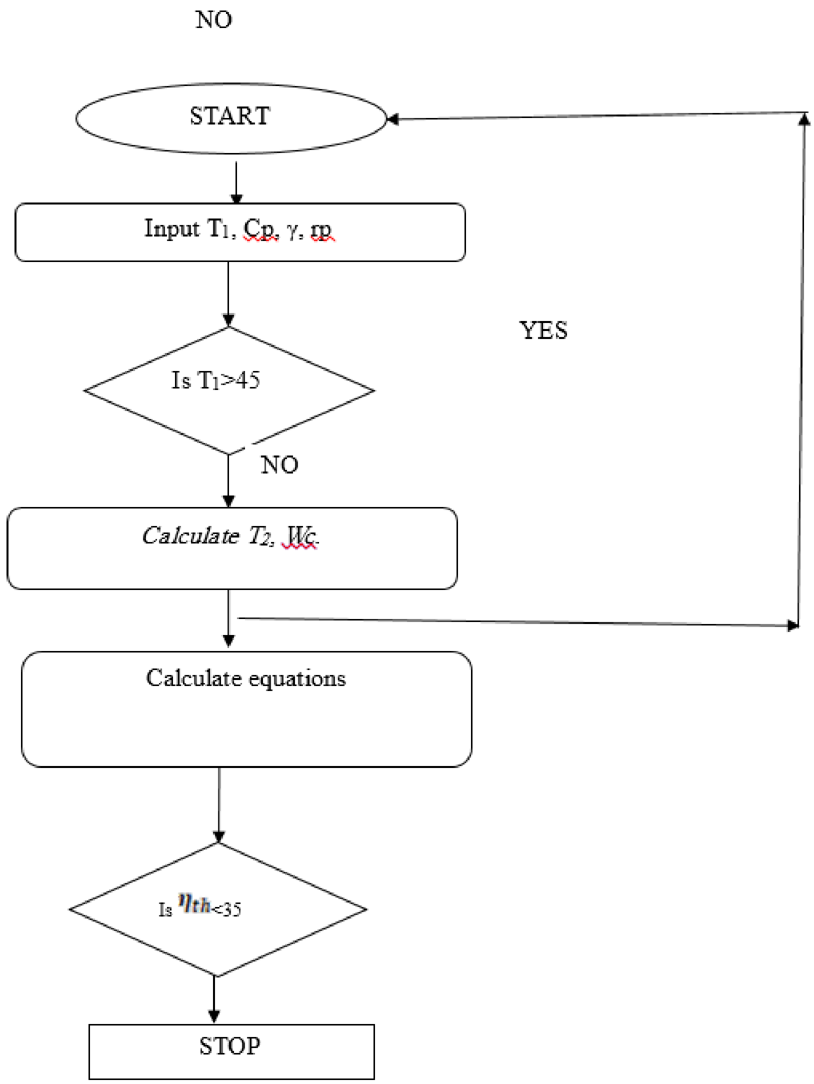

Fig 3.1 Flow chart for implementing GT simulation program

3.3 Vibration Measurement Instruments

Measurement of vibration signal is done using all the portable vibration censors or devices mounted on machine for permanent monitoring. These devices ranges from transducers connected to the instrument analyzer used to analyze the signal or spectrum in frequency or time domain (Ogbonnaya, 2004).

Analyzers include vibration analyzers, octave, and real time, Time series, percentage bandwidth, Narrow bandwidth modal plotters. These instruments are used to analyze complex vibration wave-forms at different frequency range depending on the frequency; these analyzers record the amplitude and frequency of components with complex sound, (octave) online measurement, real time analyzers and weakness in structure due to fatigue (modal plotters). There are more and new sophisticated analyzers with programmable microcomputers, installed for conversion from time to frequency domain using fast Fourier transformation (FFT) as shown in figure 3.8.

$$F_{(j\omega)} = \int_{-\infty}^{+\infty} f(t)\, e^{-} = (j\omega) dt \qquad 3.11$$

Where

$F_{(j\omega)}$ is the spectral density function in frequency domain,

$f(t)$ is spectral density in time domain

$$f(t) = \int_{-\infty}^{+\infty} \big(F(j\omega)\big)\, e^{(j\omega t)} dt \qquad 3.12$$

This can be made suitable for digital computation by making the time variable (t) discrete ie

$$T = nt$$

Where

$n =$ integer

$$F(k, j, \omega k) \ = \sum_{N=0}^{N-1} f(nT)\, e^{-\left(\frac{j\omega kT}{N}\right)} \qquad 3.13$$

Where

$k = 0,1 --- , N - 1$

This is further simplified, with T normalized to unity and the number of time samples N equals the number of frequency sample k, is as follows.

$$F(k) = \sum_{N=0}^{N-1} f(nT)\, e^{-\left(\frac{j\omega kT}{N}\right)} \qquad 3.14$$

Where

$E^{(j\omega/N)}$ is replaced with ω_n and the discrete fourier transform takes the form.

$F(k) = \sum\_(N = 0)\hat{}(N - 1)[f(n)]\ (\omega n - nk)$

$$f(n) \ = \sum_{N=0}^{N-1} (F(k)\, \omega n - nk) \qquad 3.15$$

The block diagram of FFT algorithm is shown in figure 3.7

3.4 Defuzzification

Defuzzification is defined as the method of generating a quantifiable result in fuzzy logic system with a given fuzzy set and corresponding degrees. It is a process that maps out a fuzzy set into corresponding membrship degees in fuzzy sets as shown in figure 3.2. The defuzzification technique used in this system is called centroid and is usually applied in fuzzy control systems. The technique is given by the formulae;

$$COG = \frac{\sum_{x}^{b} \mu_A(x) x}{\sum_{x}^{b} \mu_A(x)} \qquad 3.16$$

Where, $\mu_A(x)$ membership degree of x in a set A.

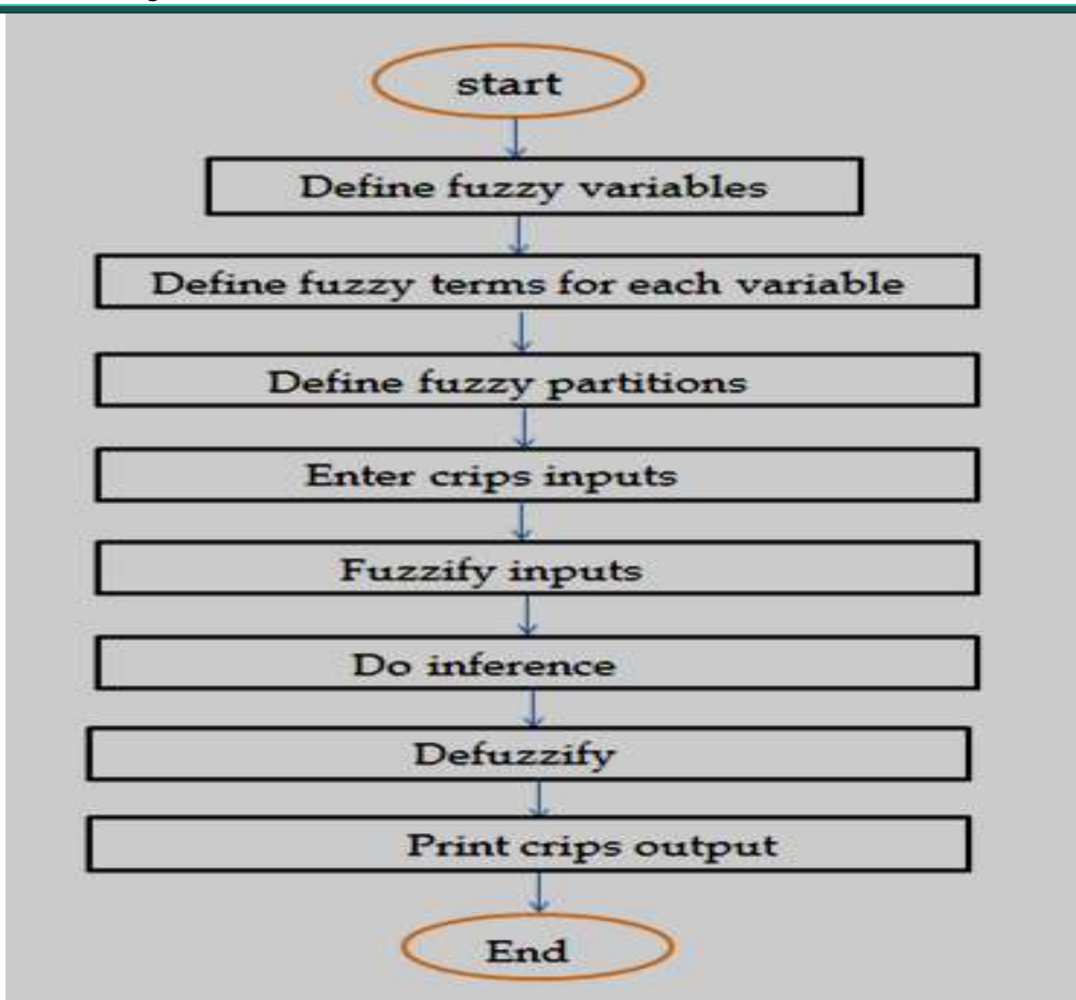

Figure 3.2: Typical flow chart on faulty logic used for this work

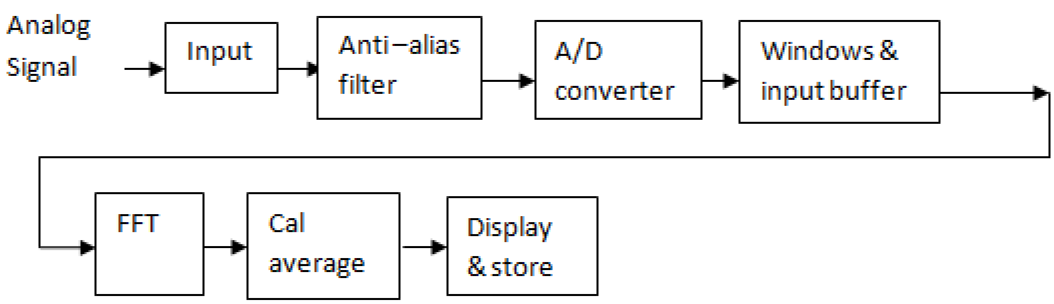

Fig 3.3: FFT block diagram (Ogbonnaya 2004)

3.5 Sensors used for fault detection

i. Temperature sensors

Temperature sensing equipment used is the radiation pyrometer. It is in a number of places within the GT. A radiation pyrometer is divided into three categories: engine casing head mounted on engine which receives infrared radiation (*IR*) given off by the rotating turbine signal into two electrical signals which are temperature compensated, amplified and fed directly into the digital engine control unit.

ii. Pressure sensors

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The gas turbine uses the differential pressure type transducer for pressure sensing. It measures the filtered load by continuously monitoring the pressure on both sides of the filter and then reports the difference as the pressure loss.

iii. Speed Sensor

The sensor used in measuring the shaft speed of the GT is the tachometer which is connected to the turbine shaft. It generates a direct current signal having a magnitude proportional to the actual speed of the turbine.

3.6 Calculation and program implementation

The combined formulas expressed above for the development of the GTE are implemented using Java Source Code provided at the Appendix A. The programme algorithm is graphically represented as a flowchart while the model design parameters which are the input values to the programme are presented in table 3.1. These are initialized and used in the programme to a flowchart that led to the Java program source code and are shown in figures 3.3 and Appendix A1.

For the development of the flow chart equations $(K_T - I\omega^2)A_1 - D_T\omega A_2 = To$, is used for the evaluation of the amplitudes of angular displacement θ, angular velocity $\dot{\theta}$ angular acceleration $\ddot{\theta}$, also equation $\theta c = Be^{-\alpha t} + \sin(\omega nt - \phi)$is used for evaluation of the natural frequency of the system under consideration while equation $A_1 = To\frac{(K_T - I\omega^2)}{\{(K_T - I\omega^2)^2 + (D_T\omega)^2\}}$is used to evaluate the phase angle. Flow chart is necessary to direct how the equations developed are used as devices to measure and control vibration activities in the model developed for GT early fault detection. All variables relevant to the torsional vibration response of the system are included in the flow chart in Appendix A1.

RESULTS AND DISCUSSIONS

Speed, velocity and GT load relationship on FDIA

The flow chart was formulated from equations $(K_T - I\omega^2)A_1 - D_T\omega A_2 = To$ and $A_1 = To\frac{(K_T - I\omega^2)}{\{(K_T - I\omega^2)^2 + (D_T\omega)^2\}}$. The flow chart uses an allowable velocity amplitude of 8mm/s and a Natural frequency ωn (resonance) when $\omega = \omega$n as alert and danger warning condition for an impending fault. These according to ISO 10816-4 are safety gadgets used to protect the plant from catastrophic failure..

The flow chart is used in the programming to calculate the TV response of the system. From the equation, all other parameters are constant except the speed. The angular velocity amplitude is plotted against speed and load for the calculated values and measured values from the plant as shown in figures 4.3. The plot shows the variation of amplitude with speed / frequency. An increase in the amplitude with speed or frequency is an indication of a fault or an impending failure.

From the computational analysis the graphs of angular velocity amplitude versus speed and angular velocity amplitude versus load are drawn as shown in figures 4.1 and 4.2. Also graphs of velocity amplitude versus speed and velocity amplitude versus load of data from plant (mv) was obtained and shown as figures 4.3 and 4.4. Similarity from both graphs and the range of deviation shows that a combination of Java programming technique and the dynamic response equation can be used for TV analysis of GT turbine rotor shaft system.

High amplitude of vibration is an indication of fault or an impending failure. An increase in vibration with changes in the speedω or frequency and phase angle shows the cause and probability of impending faults. Therefore, the graphs of angular velocity amplitude versus speed/frequency and velocity amplitude versus load simulated values and plant measured data depict presence of vibration. This shows the power of computer aided design, its application and relevance in TVA of GT.

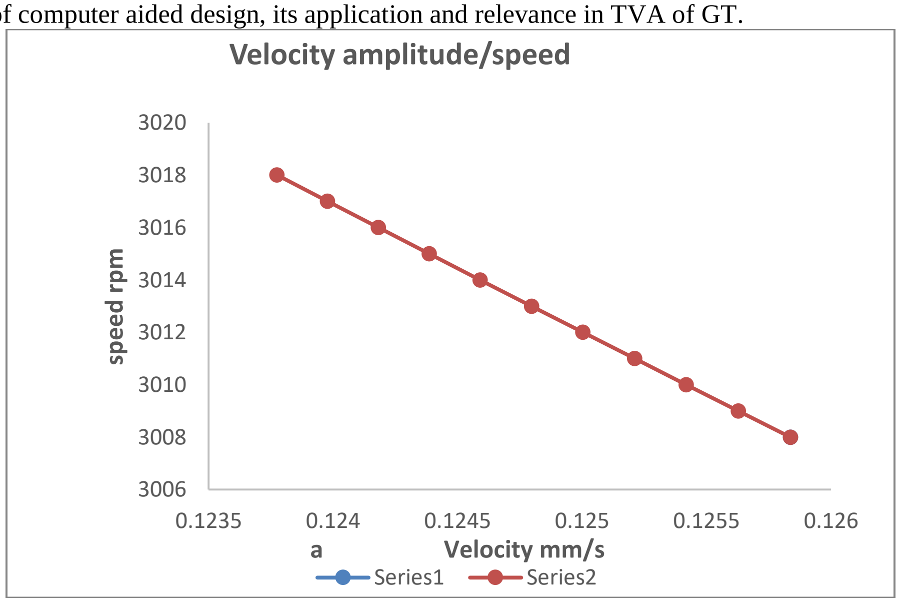

Fig. 4.1: Velocity amplitude / speed of simulated values on FDIA

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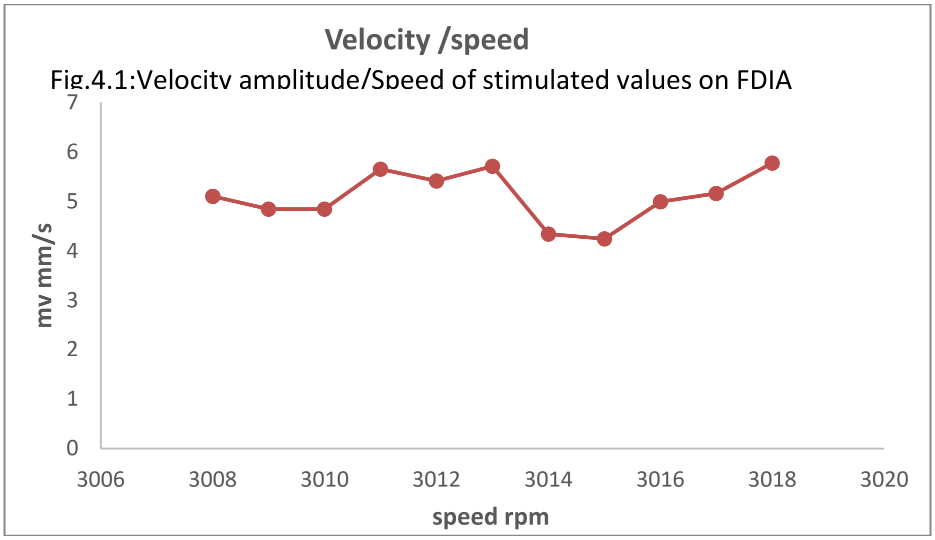

Fig. 4.2: Plant data velocity /speed on FDIA

Higher loads causes low inlet temperature of air and a reduction of engine mass flow rate due to pressure. However, during operation at varying velocity, the gas turbine engine is more difficult to control and could develop a faulty engine. Figure 4.2 presents plant data velocity and fault detection identification analysis plot.

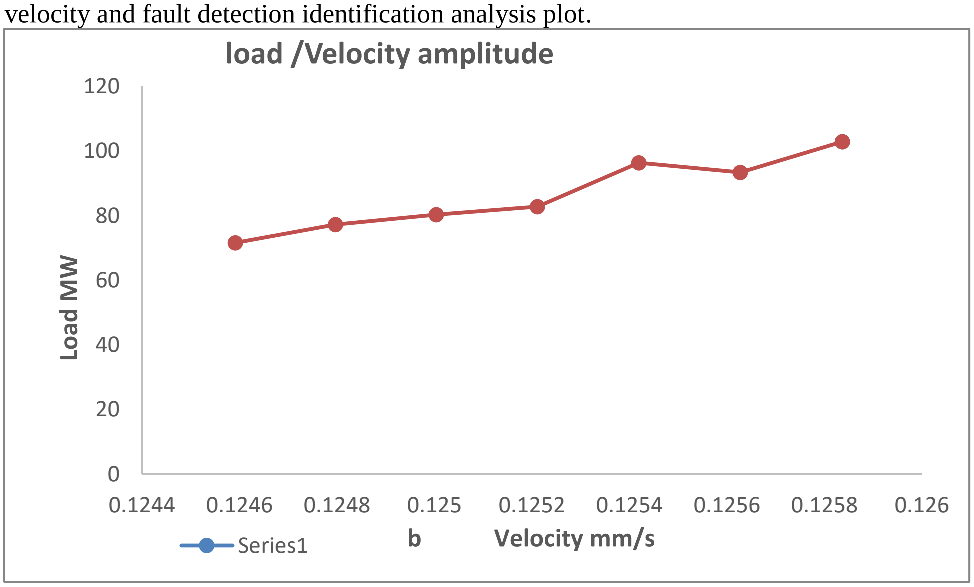

Fig. 4.3: Plant load/calculated velocity on FDIA

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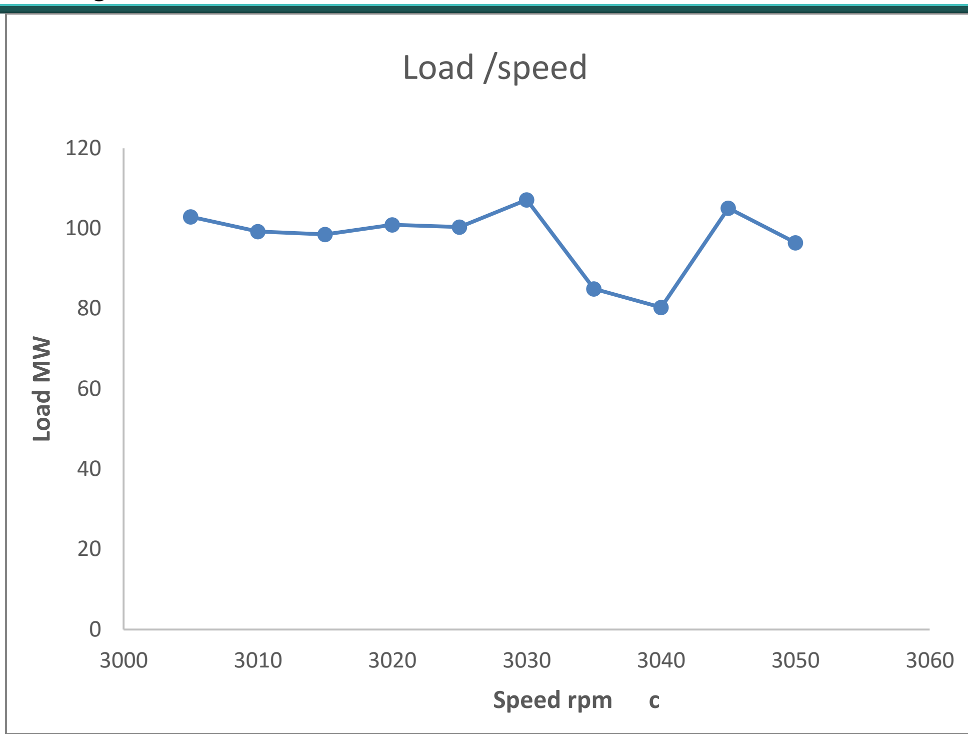

Fig 4.4: Plant load/speed on FDIA

Figures 4.3 and 4.4 gives a description of shaft over-speed scenario and its prediction to mechanical shaft failure rate. A broken causes an instant disassembly between the turbine and its compressor. This however occurs without any change of load, and therefore, the expected consequences of this unbalance activity will be noticed by the rapid deceleration of the compressor not powered by the turbine, and the rapid acceleration of the turbine due to sudden loss of load. During disc rotation, centrifugal forces acting on the structure are resisted by the stiffness of the material of the disc. The greater the rotational speed, the greater the stress. When the velocity exceeds 0.1254 the probability of fault occurring increases and this can cause a permanent deformation of the disc. If the velocity increases the ultimate strength of the disc may fail. Therefore, excessive speed could burn the disc and release debris due to high-energy from the engine. Since the casing is not designed to contain these high energy fragments. It is therefore significant that the speed of a turbine disc under any condition does not exceed its burst speed.

4.1 Torsional response on FDIA

The shaft is associated with vibration and as shown in figure 4.7. From the figures shown the model GT is a healthy one. An increase in TV response with speed is an indicator of an impending fault. An increasing Θ at start up constant speed and stop are symptoms of failure. Also with an increasing Θ at different frequencies is pointing at faults. If the torsional displacement is increased it is also pointing at GTE fault due to misalignment (Marticorena et al, 2019).

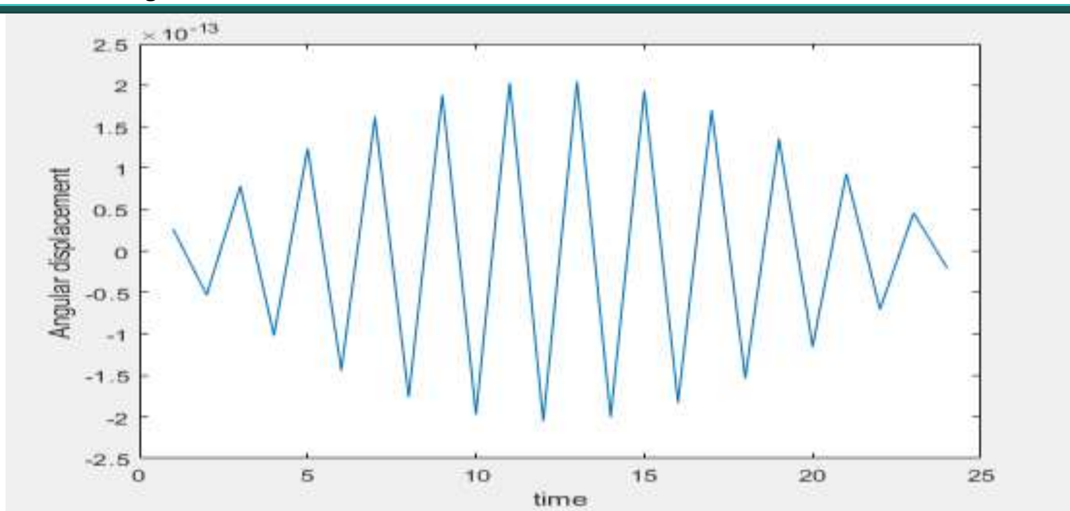

Fig. 4.5:Time versus angular displacement

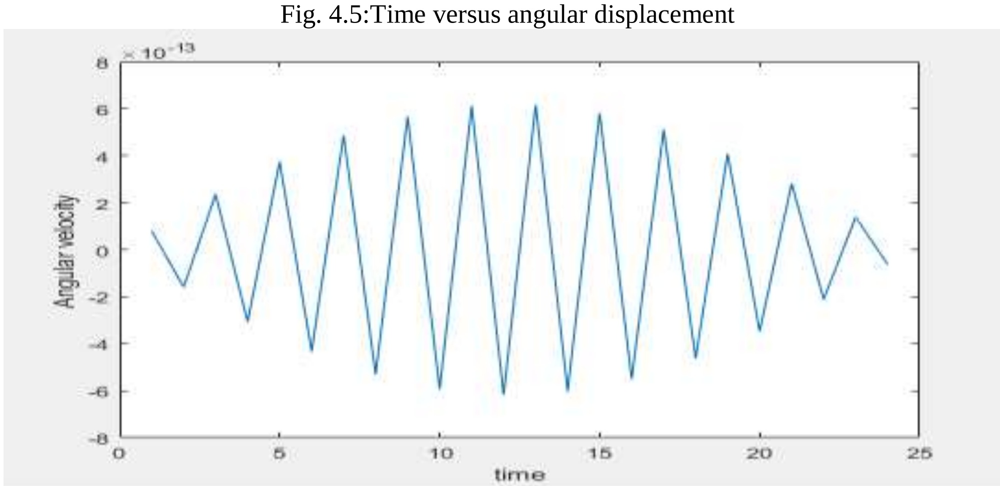

Fig. 4.6: Time versus angular velocity

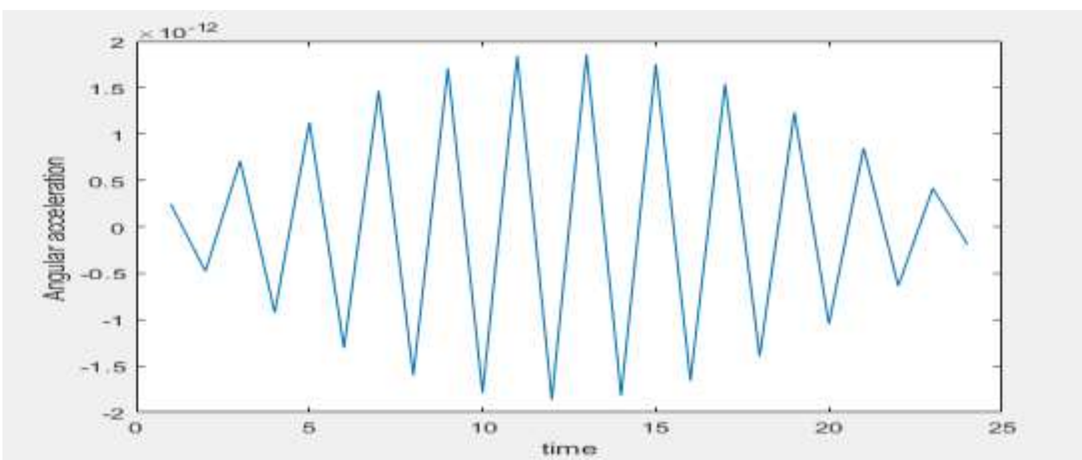

Fig. 4.7:Time versus angular acceleration

The torsional response analysis showed that the vibration was minute and was really detected

A total of eleven troughs and ten crests are shown in the vibration response. $2.18 * 10^{-13}$ is the peak angular displacement, $6.08*10^{-13}$is the peak angular velocity and$1.84*10^{-12}$ is the angular acceleration. The GT engine torsional response is quite low and it signifies that the engine is healthy and faults will really occur in this turbine.

CONCLUSION

The gas turbine engine (GTE) transforms fluid energy to mechanical energy therefore, the combination of thermodynamic and TVA for fault detection is a proactive solution required to reduce all unscheduled break down of the gas turbine (GT) rotor shaft system. The adoption of Java program and AI for result simulation and control in the software developed makes it superior and sensitive for GT monitoring and performance improvement.
Finally, a device has been developed for early fault detection, identification and prediction analysis for the power plant industry from the research work which if well implemented will reduce catastrophic failure of the GTE. Consequently, most of the numerous unscheduled breakdown will be reduced including improved plant reliability and availability.

RECOMMENDATIONS
The software presented by this research work is a proactive solution for failure analysis of the gas turbine engine (GTE). The recommendations are as follows:
(i) The developed software should be interfaced for use in the power plant industry.
(ii) The developed software should be introduced in tertiary institutions as training aids.
(iii) The system should be modified to include environmental factors and operational factors.

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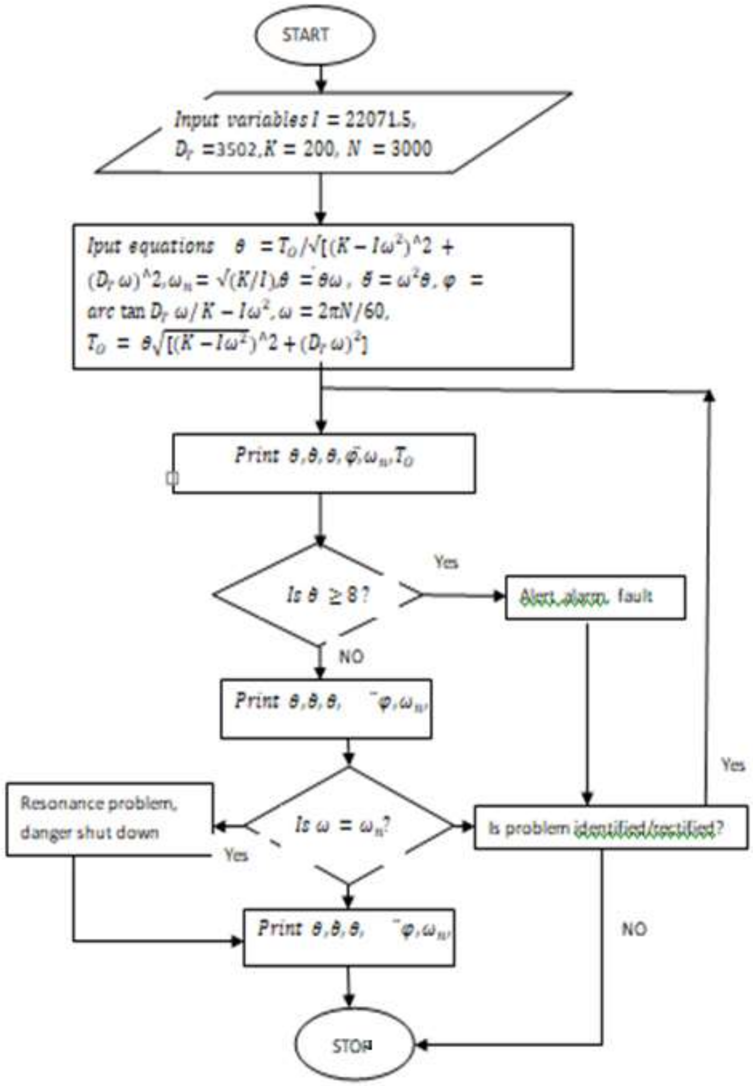
APPENDIX A1



Flow chart to compute vibration amplitude of model GT