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Optimizing Gas Turbine Rotor Shaft Fault Detection, Identification and Analysis for Effective Condition Monitoring

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Abstract

Vibration analysis of rotating machinery can give warning of potential faults such as imbalance, bent shaft, shaft crack, misalignment, looseness and other malfunctions. This work considered the origin, causes and the effect of vibrations in rotor shaft and how they can be optimized to avert catastrophic downtime. The approach used in the research was vibration signature analysis. Data were taken from a gas turbine called GT18, GE Frame 9 on industrial duty for electricity generation in Ughelli Power station Delta State, Nigeria. The data collected after collation was used in a MATLAB computer program. The result shows that the engine should not be operated beyond 4.8 mm vibration displacement amplitude. A speed of 7860 rpm should also be avoided in order to prevent resonance. Furthermore, it enabled early detection of the various effects of vibration signature such as imbalance, misalignment, and crack that may lead to downtime.

Keywords: *rotor shaft, resonance, vibration signature, shaft defects, fault identification, and downtime.*

NOMENCLATURE:

$\ddot{X}$ = Acceleration vibration amplitude, mm/s^2
$\dot{X}$ = Velocity vibration amplitude, mm/s
C = Bearing damping coefficient, kNs/M
X = Displacement vibration amplitude, mm
F_o = External force vector, kgm/s^2
K = Rotor shaft stiffness
m = Mass of shaft, kg
ω = Frequency of vibration, Hz
C_r = Critical damping, Nm/rad

INTRODUCTION

Gas Turbines (GTs) are the power generation source of choice in many applications both for mechanical drive of machinery and electrical power generation. Many operational GT power plants use aircraft derived jet engines as a gas generation, adding a power turbine and transmission to complete the plant (International Electric commission, 2007). In Nigeria, GTs are used mainly for power generation. The oil industries utilized this form of power generation mostly. The GTs are described thermodynamically by the Brayton Cycle, in which air is compressed isentropically, combustion occurs at constant pressure, and expansion over the turbine occurs isentropically back to the starting pressure (Ogbonnaya, 2004). When fault takes place, the GT parameters are subjected to change. The changes depend on the degree of faults and interaction with other parameters. Effective condition monitoring can be carried out when the GT is in operation, which is known as on-line, or when it is off-line that is when it is down and not in operation. It is here that optimized model-based monitoring is important by using signature analysis. Vibration signature analysis is intended to monitor the health of the GT rotor shaft by recording systematic signals derived from the form of mechanical vibrations, relative displacement and so on(Ogbonnaya, et al,2010). GT rotor fault detection, identification and analysis (FDIA) is associated with rotational motion. This depends upon the turning speed of the rotor and the dynamic interaction between the shaft and other items in contact with it. Vibration measurement can be used to optimize and diagnose problems with machinery that affect output, quality and production downtime. It can also solve issues related to employee health and well being when vibration creates unacceptable levels of noise. Furthermore, it can be used for the followings:

(a) Mechanical forces detection
(b) Producing active damping to reduce bridge vibration.
(c) Electrospining technology.

In Ogbonnaya (2004), it is shown that the rotor shaft is one of the most important parts of a GT system. It has several other components attached to it whose health conditions are monitored by vibration signature analysis.

MATERIALS AND METHODS

Modern condition monitoring techniques encompass many different themes; one of the most important and informative is that of vibration analysis of the machine. Constantly monitored and detailed analysis may be made concerning the health of the "engine" and any faults, which may arise or have already arisen can be diagnosed and prognozed (Jayaswal, etal, 2008). In Ogbonnaya (2004), vibration analysis is coined to be the processing of the sensed vibration signal in displacement, velocity or acceleration amplitude. Sensing involves the processing of vibration signal using an accelerometer or a velocity transducer. The word signature is used to designate signal patterns, characterize the state or condition of a system from which they are acquired. It is very important to consider the type and range of transducers or pickup for capturing vibration signal. With the development of soft computing techniques such as artificial neural network and fuzzy logic, there is a growing interest in applying the approach to the different areas of engineering. In Bob-Manuel etal (2004), pattern recognition with back propagation architecture were adopted. It considered the training neuron of the network on the basis of pattern recognition technique valuable for large amount of data worked with. Also Aborisade et al (2004), focused on the approaches to short-term load forecasting and software tools for training and forecasting processes in power system. They also considered planning such as load forecasting based on mathematical modes to provide only for specific situations of power system under respective assumptions. According to Zuo and Fyfe (2004), these gained popularity over the other methods as they are modeled free estimators capable of synthesizing nonlinear and noisy systems. The fuzzy is developed as a means for representing, manipulating, and utilizing uncertain information. Furthermore, it is enuciated that development in micro-technology and artificial intelligence have driven the trends toward more extensive on- board diagnostic.

Vibration Signature Analysis

According to Dass and Dutt (2008), vibration in a rotating machine cannot be completely eliminated, but can be controlled within an acceptable limit. In general, the vibration condition of an engine system is assessed by consideration of the shaft vibration and the associated structural component. The effect of any excitations present is that, through the speed range, stress peaks occur as the frequency of each vibration mode matches the excitation frequency. (Cohen and Rogers, 2000). Mitchell et al (2000), describes a prototype system called the time interval measurement system. Here a digital tachometer circuit is able to record the passing times and then converted it to angular shaft velocity. Various investigations have addressed a number of issues and proposed enhancements to this approach. In Lovejoy (2008), a bowed rotor presents both machinery damage and plant safety risks. As a bowed rotor is accelerated above about 600 rpm, the centre of mass offset to the centre of rotation will increase the bow due to centripetal force. Continuing acceleration to between 1000 rpm and 1250 rpm on most large rotors brings the rotor into the first critical speed, or the first mode resonant frequency of the shaft.

Resonance

Resonance is a phenomenon that occurs when a physical system is periodically disturbed at the same period of one of its natural frequencies. At forced vibration the system will tend to vibrate with the frequency forcing function, and also its natural frequency. When this occurs, the amplitude of vibration will increase without bound. Hence giving rise to failure of components. When the GT shaft is turning, eccentricity causes a centrifugal force deflection which is resisted by the shaft's flexural rigidity. Simulation and modeling used by Shigley and Charles (2006), proved that at certain speeds the shaft is unstable, with deflections increasing without upper bound. It is fortunate that although the dynamic deflection shape is unknown, using a static reflection curve gives an excellent estimate of the critical speed. The shaft, due to its own mass, has a critical speed. The ensemble of attachments to a shaft likewise has a critical speed which is much lower than the shaft's intrinsic critical speed. Estimating these critical speeds (and harmonics) is a task of designers in order to control/prevent catastrophy which may emanate from the shaft component. Hence the purpose of this work is to enunciate means to prevent failure to rotor shafts of GTs dut to vibration. In Ramesh (2002), it was stated that due to the nature of manufacturing of the rotor and its components, unbalance always exists, although in small quantities. The amount of unbalance can be reduced to a tolerable level by different unbalance techniques. Nevertheless, the residual unbalance that is always present in the rotor system acts as a forcing function on the rotor and tends to pull the rotor away from the undeformed rotor centerline during operation. Assume an unbalance of mass, *m,* located at a radius,r and the rotor spinning at ω.The unbalanced force acting on the rotor can be given by

$$F=m\omega^2 r \quad (1)$$

This displacement of the rotor is the response of the rotor to the unbalance as shown in fig.1

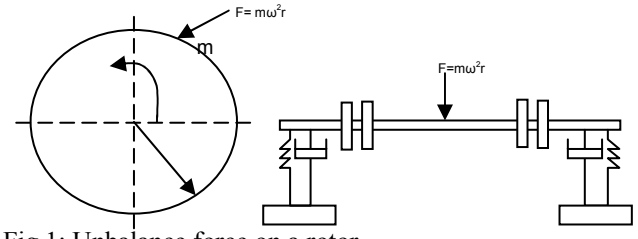

Fig 1: Unbalance force on a rotor
(**Source**: Rao, 2006)

This present work would contribute analytical solution to the unexpected downtime of a gas turbine rotor shaft. Since the shaft transmits motion, the geometry of the shaft is generally that of a stepped cylinder. Gears, bearings, and pulleys must always be accurately positioned and provision made to accept thrust loads. The use of shaft shoulders is an excellent means of axially locating the shaft elements; these shoulders can be used to preload rolling bearings and provide the necessary thrust reactions to the rotating elements. It is common practice in rotor/magnetic bearing systems to incorporate auxiliary bearings that prevent damage to the rotor. The data collected from the test engine was fed into the existing equations. Figure 2 show the forced vibration of damped systems. By applying Newton's second law {fig 2 (a)} to the free body diagram shown in fig. 2(b) and utilizing the equilibrium condition $mg = k\Delta$ gives equation (2).

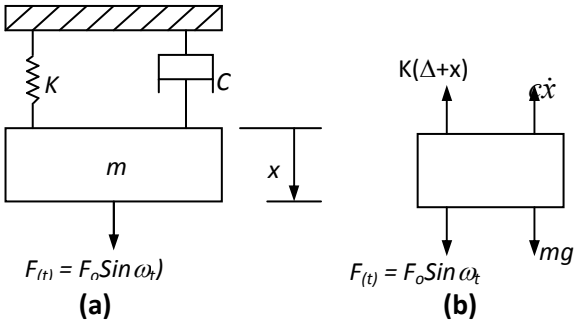

Fig. 2: Forced vibration damped systems
(**Source:** Ogbonnaya, 2004)

$$m\ddot{x} + c\dot{x} + kx = F_o Sin\ \omega_t \quad (2)$$

$$\ddot{x} + \frac{c}{m}\dot{x} + \frac{k}{m}x = \left(\frac{F_o}{m}\right) Sin\ \omega_t \quad (3)$$

Equation (2) is an inhomogeneous function. The particular solution or the steady state solution x_p can be assumed in the form

$$x_p = A_1 \sin\ \omega_t + A_2 \cos\ \omega_t \quad (4)$$

which gives the following equations:
For velocity and acceleration

$$\dot{x}_p = \omega A_1 \cos\ \omega_t - \omega A_2 \sin\ \omega_t \quad (5)$$

$$\ddot{x}_p = -\omega_{A_1}^2 \sin \omega_t - \omega_{A_2}^2 \cos \omega_t = -\omega^2 x_p \quad (6)$$

Substituting equation (4), (5) and (6) into equation (2) and rearranging the terms gives

$$(k - \omega_m^2)(A_1 \sin\ \omega_t + A_2 \cos\ \omega_t) + C\omega(A_1 \cos\ \omega_t - A_2 \sin\ \omega_t) = F_o \sin\ \omega_t$$

or

$$\left[(k - \omega_m^2) A_1 \cos\ A_2\right]\sin\ \omega_t + \left[C\omega A_1 + (k - \omega_m^2) A_2\right]\cos\ \omega_t = F_o \sin\ \omega_t \quad (7)$$

Equation (7) yields the following two algebraic equations in A_1 and A_2 $(k - \omega_m^2) A_1 c\omega A_2 = F_o$ (8)

$$C\omega A_1 + (k - \omega_m^2) A_2 = 0 \quad (9)$$

Dividing equations (8) and (9) by the stiffness coefficient K gives

$$(1 - r^2)A_1 - 2r\xi A_2 = x_o \quad (10)$$

$$2r\xi A_1 + (1 - r^2)A_2 = 0 \quad (11)$$

where $r = \omega/\omega_n$ (12)

$$\xi = \frac{C}{C_C} = \frac{C}{2m\omega_n} \text{ and } x_o = F_o/K \quad (13)$$

in which $C_C = 2m\omega_n$ is the critical damping coefficient.

Fig. 3 shows the vector relationship for forced vibration with damping.

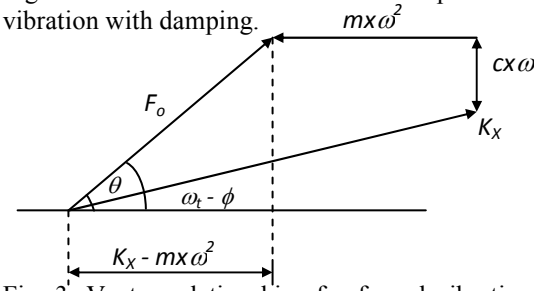

Fig. 3: Vector relationships for forced vibration with damping
(**Source:** Rao,2006)

From the diagram, forced vibration is obtained thus

$$F_o^2 = (cx\omega)^2 + (Kx - mx\omega^2)^2 \quad (14)$$

$$F_o = \sqrt{(cx\omega)^2 + (Kx - mx\omega^2)^2} \quad (15)$$

The vibration amplitude can be obtained from equation (15) as

$$x^2 = \frac{F_o^2}{(c\omega)^2 + (K - m\omega^2)^2}$$

$$x = \frac{F_o}{\left[(c\omega)^2 + (K - m\omega^2)^2\right]^{1/2}} \quad (16)$$

Dividing the numerator and denominator of equation (16) by K gives

$$x = \frac{F_o/K}{\left[\left(\frac{c\omega}{K}\right)^2 + \left(1 - \frac{m\omega^2}{K}\right)^2\right]^{1/2}} \quad (17)$$

$$Tan\ \phi = c\omega / K - m\omega^2 = \frac{c\omega / K}{1 - \frac{m\omega^2}{K}} \quad (18)$$

$$\therefore F_o = Kx\left[\left(\frac{c\omega}{K}\right)^2 + \left(1 - \frac{m\omega^2}{K}\right)^2\right]^{1/2} \quad (19)$$

Fig. 4 shows the flowchart used to write computer program in MATLAB language for obtaining the vibration amplitude of the experimental data from Delta (Ughelli) thermal station for GT unit 18 in operation.Equation (19) is used to write the program and thus compute the vibration displacement amplitude(x) at various speeds (ω).

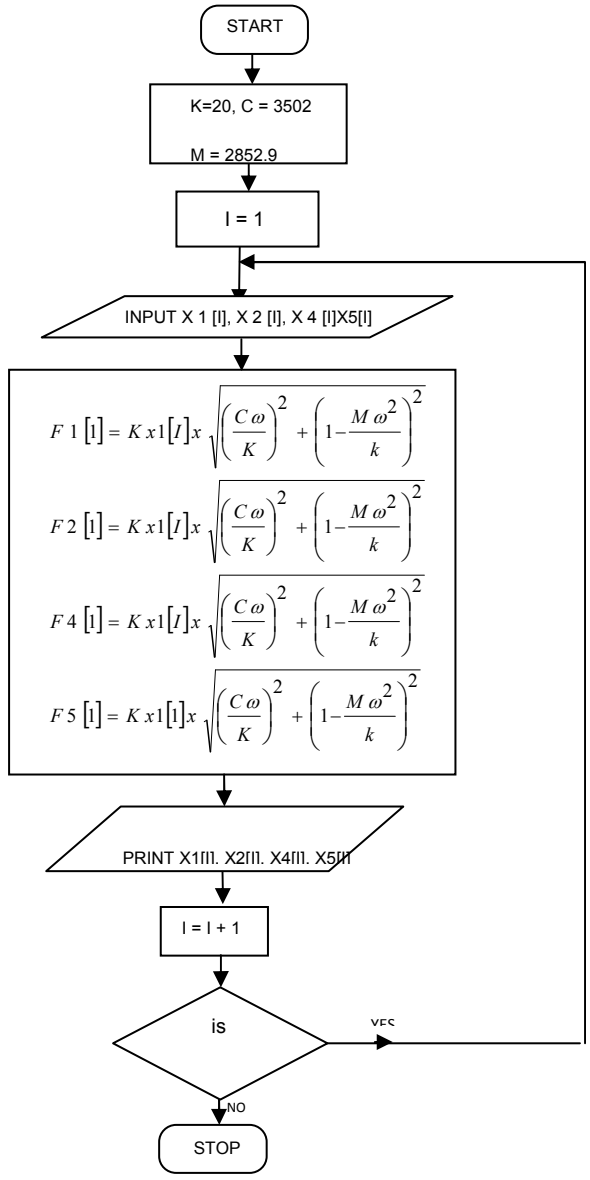

Fig. 4 : Flowchart for obtaining vibration amplitude for GT 18 in Operation

RESULTS AND ANALYSIS

The analyses done so far confirmed that vibration is very dynamic and cannot be predicted. It happens either in sinusoidal form or wavy spectrum and appear sometime in a straight line. This is to say that whenever a mechanical system is in operatiion vibration always manifest itself. The vibration velocity amplitude data was collected at various loads as well as the turbine speed. In an attempt to investigate the effect of load on the turbine, various speeds was also taken on this load.

Simulation of Delta (Ughelli) GT Unit 18

GT 18 is a healthy engine. The MATLAB program result of the vibration velocity amplitude (mm/s) is shown in table 1, Bearing 3 is not shown because it exists as a thrust bearing in which no transducer is fitted.

Table 1: Vibration Readings taken on healthy GT unit 18

			Vibration Velocity Amplitude (mm/s)			
Time (hrs)	Load (MW)	Speed %	Bearing 1	Bearing 2	Bearing 4	Bearing 5
0100	93	90	3.8	2.6	0.5	-0.2
0200	95	92	3.5	3.0	0.4	-0.1
0300	95	94	3.6	2.0	0.6	-0.2
0400	90	96	3.9	2.7	0.7	-0.1
0500	40	98	3.7	2.1	1.0	-0.2
0600	75	100	3.8	2.5	0.5	0.3
0700	75	102	4.8	2.0	0.8	0.5
0800	74	104	3.8	2.1	0.5	0.3
0900	79	106	3.5	2.6	0.4	0.1
1000	90	101	3.6	2.7	0.6	-0.2
1100	95	103	3.9	2.4	0.8	-0.1

ANALYSIS OF SIMULATION RESULTS

Fig. 5 below shows that the turbine speed fluctuates with increasing vibration velocity amplitude.It is caused by excessive vibration at the bearing.

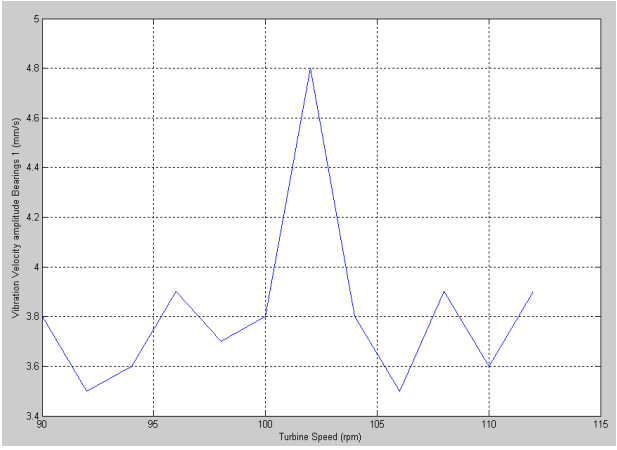

Fig. 5: Vibration amplitude against turbine speed

Figure 6 shows the spectral results after applying the same resampling technique used for rotor shaft detection.The vibration amplitude increased sharply at bearing 2, due to a sudden increase of speed.

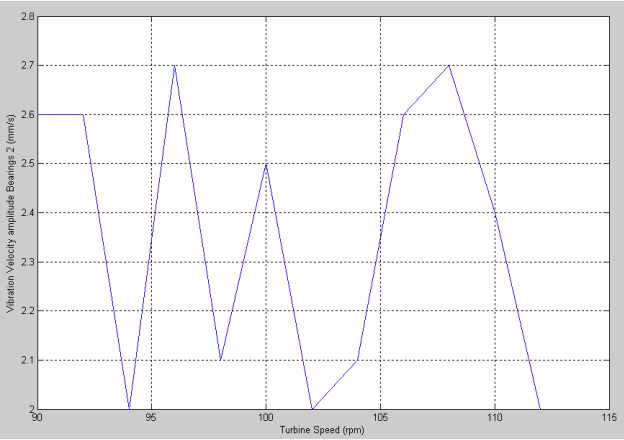

Fig. 6: Vibration amplitude of bearing 2 against turbine speed

Fig. 7: vibration amplitude for bearings 1, 2, 4 and 5 against turbine speed. This is the graph of the four bearings against turbine speed, It is observed that the vibration amplitude increases from bearing 1 to 5. This is so because bearing 5 is closed to the gas generator. The high vibration amplitude at the bearing housing might be due to misalignment.
The graph shows sthat as the load increases, the speed decreases and then falls suddenly.

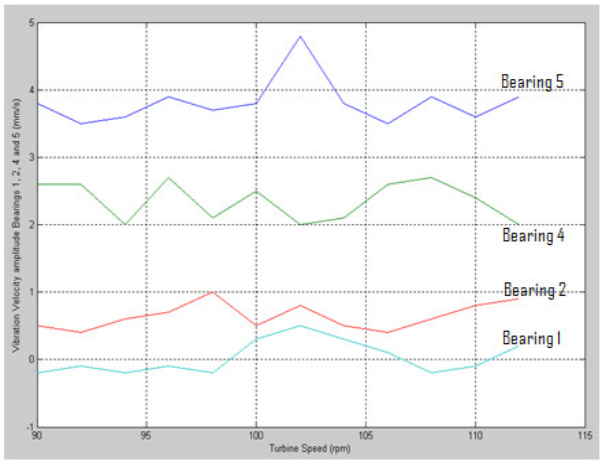

Fig. 7: Vibration amplitude of bearings 1, 2, 4 and 5 against turbine speed

The vibration amplitude increased sharply at bearings 1 and 2 due to highest at point 5.1 in fig.8 and 2.8 in fig.9 respectively.

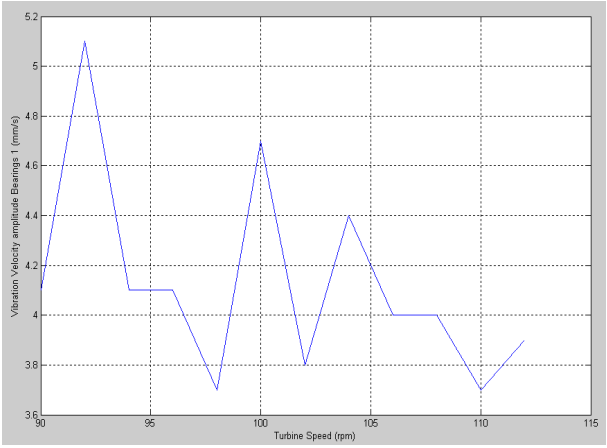

Fig. 8: Vibration amplitude of bearing 1 against turbine speed

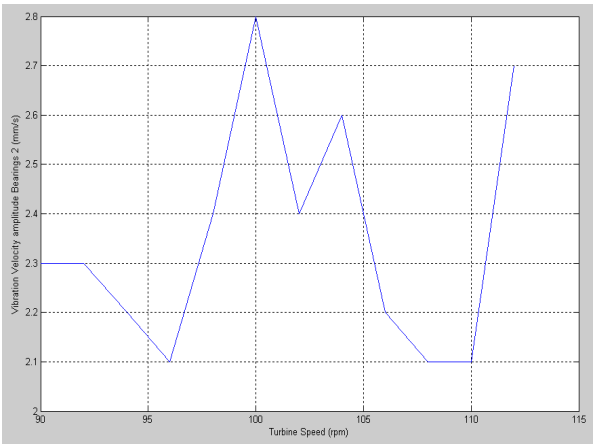

Fig. 9: Vibration velocity amplitude bearing 2 against turbine speed

The vibration amplitude increase in fig. 10 implies that the level of vibration in an engine while in operation is determined by the loading. Hence the sudden removal of the load resulting in either reduction or increase in vibration amplitude.This is caused by increase in torsional stresses that the load brought about in the shafting system.

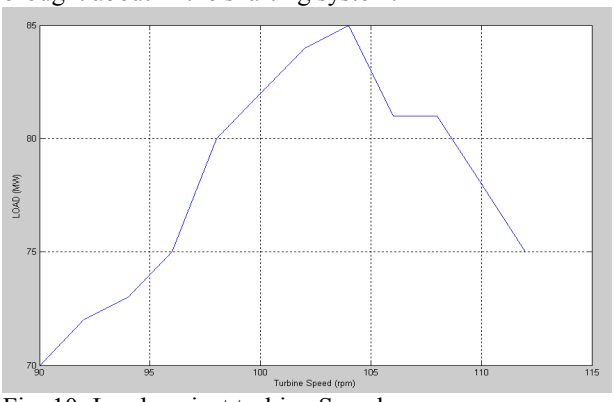

Fig. 10: Load against turbine Speed

The speed fluctuates with a steady increase in vibration amplitude causing unusual waveform due to mass imbalance as shown in fig.11. Hence, it is observed also that the vibration of bearing 4 increases rapidly. This result due to zero eccentricity in the rotor shaft. This could lead to downtime of the component and until proper measure is taken.

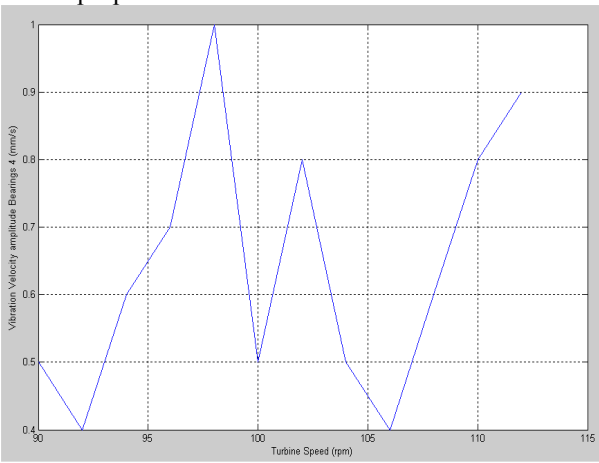

Fig. 11: Vibration velocity amplitude of bearing 2 against turbine speed

CONCLUSIONS

The amount of defect meted to the GT is a function of the amplitude of the vibration on the rotor shaft and bearings. It is observed that the bearing at the turbine side carries too much load. This brings about the high vibration amplitude on the bearing. Therefore the vibration amplitude on the bearing along the rotor shaft depends solely on the distance of the bearing from the load end of the shaft.

Furthermore, the above graphical presentations show that it is easier to use FDIA to move from one bearing to another in an attempt to identifying faults. Vibration analysis therefore helps in monitoring the health of a GT rotating component through the enunciated FDIA in this research. Vibration signature also helps alert equipment operators of engine health condition. The result shows that the engine is not to be operated beyond 4.8mm vibration displacement amplitude. A speed of 7860 rpm is also be avoided in order to prevent resonance.

RECOMMENDATIONS

Based on the research work carried out, the following recommendations are made.

1. A software to monitor the thrust bearing and its vibration amplitude should be developed.
2. In order to validate modeling in a gas turbine rotor shaft fault, more attention should be given to automated techniques.
3. The relationship between vector and vibration in modeling rotor shaft should be developed

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