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Advanced Gas Turbine Rotor Shaft Fault Diagnosis Using Artificial Neural Network

Ezenwa A. Ogbonnaya[1], Emmanuel M. Adigio[1], Hyginus U. Ugwu[2] and Magnus C. Anumiri[2,*]

[1]Department of Mechanical and Marine Engineering, Niger Delta University, Wilberforce Island, Bayelsa State, Nigeria

[2]Department of Mechanical Engineering, Michael Okpara University of Agriculture Umudike, Umuahia, Abia State, Nigeria

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Abstract

The effect of vibration in plant leads to catastrophic failure of a system. This is why vibration monitoring of a system constitutes a very key practice of ensuring power plant availability. Force, Amplitude and Resonance a program written in Visual Basic Programming language was utilized in this study to monitor the vibration level of the Gas Turbine (GT17) in Afam thermal station and to calculate the force causing vibration on the bearing. The program was also run using the data obtained from the plant. Results show that vibration velocity amplitude of bearing 2 on weeks 5 and 8 were 6.7mm/s and 6.6mm/s and the forces causing vibration were $2.545\text{x}10^4$N and $2.272\text{x}10^4$N respectively. The comparison of results obtained with maximum vibration velocity amplitude of the machine (7mm/s) showed that the vibration of the machine was tending towards the maximum value. Therefore, proper attention should be given to bearing 2 to avoid failure of the Gas Turbine.

Keywords: artificial intelligence, condition monitoring, excitation force, amplitude, resonance

1. Introduction

The performance efficiency of power plants is of great importance to engineers and every nation. Organizations have thus embarked on different maintenance management strategies to ensure high reliability of plant availability. To meet these challenges, one way is to check the growing energy demand and to develop new maintenance strategy for power plant's availability [1]. In-view of this, it became necessary to propose a new additional strategy to save the gas turbine from catastrophe through the consideration of its rotor shaft vibration.

The new methodology took into consideration most of the existing monitoring methods and integrating Artificial Intelligence (AI) method of Artificial Neural Network (ANN) into it [2-4]. ES is an intelligent agent that perceives its environment and takes action that maximizes its chances of success. ANN is a branch of AI that is suitable for establishing intelligent fault diagnostic systems [5].

Inspired by the structure of the brain, ANN consists of a set of highly interconnected entities called nodes or units [6]. Each unit is designed to mimic its biological counterpart, the neuron and accepts a weighed set of inputs and responds with an output. A typical diagram of how the neuron works is shown in Fig.1 [7].

* Corresponding author. E-mail address: anumiric@gmail.com
Tel.: +2348069235467; Fax: +2347081183785

Thus, the application of ANN in GT running and maintenance becomes a proper tool when integrated into vibration monitoring.

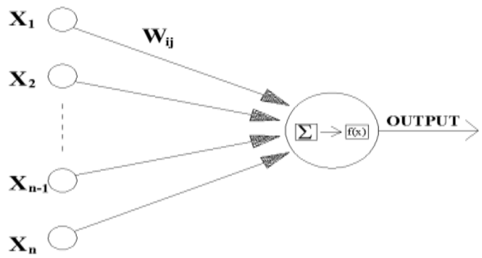

Fig. 1 Operation of a neuro [8]

2. Materials and Methods

The engine used to actualize this work is a 75MW plant popularly called GT17 in the series of Afam IV plants. It is a Type 13D equipment manufactured in 1981. Other characteristics of the plant relevant to this work are as presented in Table 1.

The vibration analysis method used consists of five steps:

(1) Formulation of mathematical model for vibration analysis.

(2) Design of computer program for mathematical models.

(3) Data collection.

(4) Analysis of data; and

(5) Data storage for future.

These steps were configured to form a continuous online system (periodic analysis system using portable equipment) for GT running and maintenance. The determination of which configuration would be more practical and suitable depends on the critical nature of the equipment, and also on the importance of continuous or semi continuous measurement data for that particular application.

2.1. Mathematical Model for Vibration Analysis

2.1.1. Engine vibration monitoring

Measurements of vibration parameters are important vibration monitoring. The parameters desired may be displacement, velocity, or acceleration; in time or frequency domain. These quantities are useful in predicting the fatigue failure of a particular component of machine and play important role in analysis, which are used to reduce equipment vibration.

When measurement of both amplitude and frequency are available, diagnostic methods can be used to determine the magnitude of a problem and its probable cause since each mechanical defect generates vibration in its own unique way. This makes it possible to identify a mechanical problem by measuring and noting its vibration signature.

Also, when vibration measurements and analysis are performed systematically and intelligently, they will not only allow determination of machine health but also permit the prediction of the mechanical fault [11].

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Table 1 Characteristics of Afam GT 17 System Relevant to this Work

S/No	Description	Specification
1	Name of Equipment	Brown Boveri Sulzer Turbo Maschinen
2	Manufacturer	Asea Brown Boveri
3	Capacity	75 MW
4	Year of Manufacture	1981
5	Year of Installation	1982
6	Year of Commission	Nov 1982
7	Type	13D
8	No of Turbine Rows	5
9	No of Compressor Rows	17
Particulars of Rotor		
10	Length of Rotor shaft:	8000mm
11	Moment of Inertia, I	586.2m4
12	Modulus of Elasticity, E	207GN/m2
13	Mean Diameter, δ	1400mm
14	Density, p	7850kg/m3
15	Modulus of Rigidity, G	80GN/m2
16	Mass of Turbine shaft	23500kg
17	Mass of Compressor shaft	24000kg
18	Natural frequency, ω_n	350 rad/s
19	Damping Ratio	0.9
20	Spring stiffness, K	1.5x106 N/mm
21	Maximum vibration limit	7.0mm/s

2.1.2 Vibration signal pick up

The first step of the program, detection, simply involves measuring and trending vibration levels at marked locations on each machine included in the program on a regularly scheduled basis [12]. The instrument includes a transducer that is held or attached to the bearing cap of the machine. The transducer converts the machine vibration into an equivalent electrical signal that is read on the meter as a vibration level. It is very important to know where and how to take vibration readings. Any noted increase in the level of vibration is a positive warning of developing problems.

2.1.3. Data analysis

The collected data were analyzed using mathematical models for the structural member of the rotor system. In order to obtain a mathematical model for vibration analysis of the GT plant, the following equations were considered:

Excitation force: Assuming a shaft of mass ‘M’ with stiffness ‘K’ and damping coefficient ‘C’ rotating with a speed of, subjected to an excitation force $F = F_0 \sin\omega t$, as shown in Figs. 2.and 3, by considering section A-A, the following expressions are obtained:

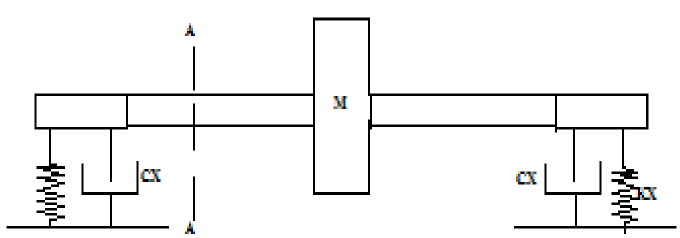

Fig. 2 Turbine shaft with spring and damper

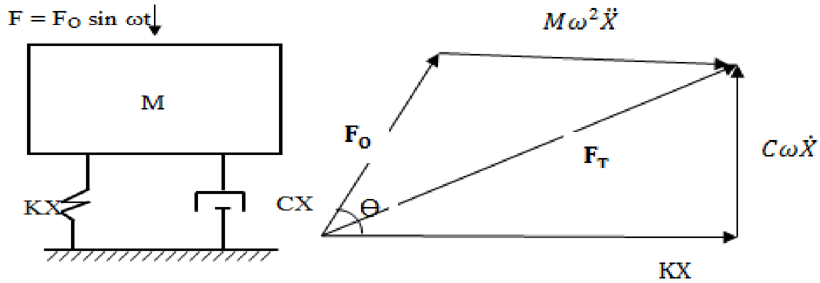

Fig. 3 Free body diagram of a vibrating element

The equation of motion for the system can be written as;

$$M\ddot{x} + C\dot{x} + Kx = F_0 Sin\omega t \tag{1}$$

The expression (1) is a linear non homogenous second order differential equation. The solution of this equation consists of two parts, the complementary part, X_c and the particular integral part X_p. Therefore the solution may be written as:

$$X = X_c + X_p \tag{2}$$

The complementary function is obtained by setting the equation (1) to zero. i.e.,

$$M\ddot{x} + C\dot{x} + Kx = 0 \tag{3}$$

Thus, the complementary solution becomes

$$X_c = Be^{-a\,t} + Sin(\omega_d t - \theta) \tag{4}$$

$$X_P = A_1 Sin\omega t + A_2 Cos\omega t \tag{5}$$

$$\dot{X} = \omega A_1 Cos\omega t - \omega A_2 Sin\omega t \tag{6}$$

$$\ddot{X} = -\omega^2 A_1 Sin\omega t - \omega^2 A_2 Cos\omega t \tag{7}$$

Substituting equations (5), (6) and (7) into equation (3) and rearranging gives,

$$(K - \omega^2 M)(A_1 Sin\omega t + A_2 Cos\omega t) + C\omega(A_1 Cos\omega t - A_2 Sin\omega t) = F_0 Sin\omega t \tag{8}$$

Comparing coefficients of $\sin\omega t$ and $\cos\omega t$ on the left and the right hand side separately yields:

$$(K - \omega^2 M)A_1 - C\omega A_2 = F_0 \tag{9}$$

$$(K - \omega^2 M)A_2 + C\omega A_1 = 0 \tag{10}$$

From (10), making A_2 the subject of the formula and substituting into equation (9)

$$(K - \omega^2 M)^2 A_1 + (C\omega)^2 A_1 = F_0 (K - \omega^2 M)$$

$$A_1 = F_0 \frac{(K - \omega^2 M)}{(K - \omega^2 M)^2 + (C\omega)^2} \tag{11}$$

$$A_2 = F_0 \frac{(-C\omega)}{(K - \omega^2 M)^2 + (C\omega)^2} \tag{12}$$

The particular integral of the differential equation (4) is

$$X_P = A_1 \mathrm{Sin}\varpi t + A_2 \mathrm{Cos}\varpi t$$

$$X_P = F_0 \frac{(K - \varpi^2 M)}{(K - \varpi^2 M)^2 + (C\varpi)^2} \mathrm{Sin}\varpi t + F_0 \frac{(-C\varpi)}{(K - \varpi^2 M)^2 + (C\varpi)^2} \mathrm{Cos}\varpi t$$

$$X_P = \frac{(F_0)}{(K - \varpi^2 M)^2 + (C\varpi)^2} \times [(K - \varpi^2 M) sin\varpi t - C\varpi cos\varpi t] \quad (13)$$

$$C\varpi = XSin\theta \text{ ; and } (K - \varpi^2 M) = XCos\theta$$

$$X = \sqrt{(K - \varpi^2 M)^2 + (C\varpi)^2}$$

$$Tan\theta = \frac{C\varpi}{(K - \varpi^2 M)}$$

Hence, the particular integral becomes

$$X_P = \frac{(F_0)\, X}{(K - \varpi^2 M)^2 + (C\varpi)^2} Sin(\varpi t + \theta) \quad (14)$$

Conversely, the complete solution of (2) becomes:

$$X = Be^{-a\,t} + Sin(\varpi_d t - \Theta) + \frac{(F_0)\, X}{(K - \varpi^2 M)^2 + (C\varpi)^2} Sin(\varpi t + \theta) \quad (15)$$

This implies that the complementary function is small compared to the particular integral. Therefore, the displacement *X* at any time is given by the particular integral, X_p only. Hence, the amplitude for forced vibration is given as:

$$X = \frac{(F_0)}{\sqrt{(K - \varpi^2 M)^2 + (C\varpi)^2}} \quad (16)$$

Also, the excitation force, F_0 causing vibration of the body can be calculated as:

$$F_0 = X\sqrt{(K - \varpi^2 M)^2 + (C\varpi)^2} \quad (17)$$

Let stiffness $K = \frac{\varpi_n^2}{M}$; damping ratio $\zeta = \frac{C}{Cc} = \frac{C}{2M\varpi_n}$; where: C_c is the critical damping coefficient and ϖ_n is the natural frequency of the system, then dividing equation (17) by stiffness, K becomes:

$$F_0 = KX\sqrt{\left(1 - \frac{\varpi^2}{\varpi_n^2}\right)^2 + (2\zeta \frac{\varpi}{\varpi_n})^2} \quad (18)$$

Moreso, converting vibration velocity to displacement gives:

$$x = \frac{\dot{x}}{2\pi f} \quad (19)$$

Resonance: Resonance occurs in a vibrating body when the frequency of the excitation force equals that of the natural frequency of the body. Let the natural frequency of the body,

$$\omega_n = \sqrt{\frac{K}{M}} \text{ (in radians per seconds)} \tag{20}$$

$$n = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \text{ [In Hertz (Hz)];} \tag{21}$$

and the shaft speed = N [in revolutions per minutes (rpm)]; then, the frequency of a vibrating body is given

$$= \frac{2\pi N}{60} \text{(in radians per seconds)} \tag{22}$$

$$f = \frac{\omega}{2\pi} \text{ [In hertz (Hz)]} \tag{23}$$

Hence, the equation for resonance of a vibrating body is given as:

$$\beta = \frac{\omega}{\omega_n} \tag{24}$$

2.2. Computer program for mathematical model

The mathematical expressions (18), (19), (22) and (24) were used in writing the program that calculates Excitation Forces, Amplitude and Resonance, on each bearing units of the GT shaft. The computer program is called "FAR" meaning Force, Amplitude and Resonance and was written with Visual Basic. The flowchart that led to the program is shown in Fig. 4 while its proposed on-line vibration monitoring schematic is presented in Fig. 5.

2.3. Data collection technique

Data were collected on hourly basis for a period of six months from an operational GT machine used for electricity generation. The data were sampled and divided into a period of 2weeks intervals to obtain 12weeks. The statistical mean for each was taken to obtain the data shown in Table 2 while some of the characteristics of the GT plant used as a case study in this work are as previously given in Table 1.

Table 2 Vibration Data from Afam IV Unit GT 17

Weeks	**Active load (MW)**		**Shaft speed (rpm)**	**Vibration velocity amplitude (mm/s)**			
				Compressor bearings		**Turbine bearings**	
	A	**R**		**X**	**X_2**	**X_3**	**X_4**
1	50	75	3063	4.8	6.2	0.9	-5.0
2	50	75	3074	4.6	6.2	0.9	-5.0
3	50	75	3056	4.9	6.2	0.8	-5.0
4	45	75	3063	5.1	6.5	1.0	-5.0
5	40	75	3028	5.0	6.7	1.2	-5.0
6	35	75	3053	5.2	6.5	1.0	-5.0
7	37	75	3005	5.2	6.6	1.7	-5.0
8	41	75	3056	5.3	6.6	0.9	-5.0
9	40	75	3077	5.2	6.6	0.9	-5.0
10	42	75	3074	5.0	6.5	0.9	-5.0
11	40	75	3065	5.0	6.6	1.2	-5.0
12	50	75	3077	5.0	6.5	1.3	-5.0

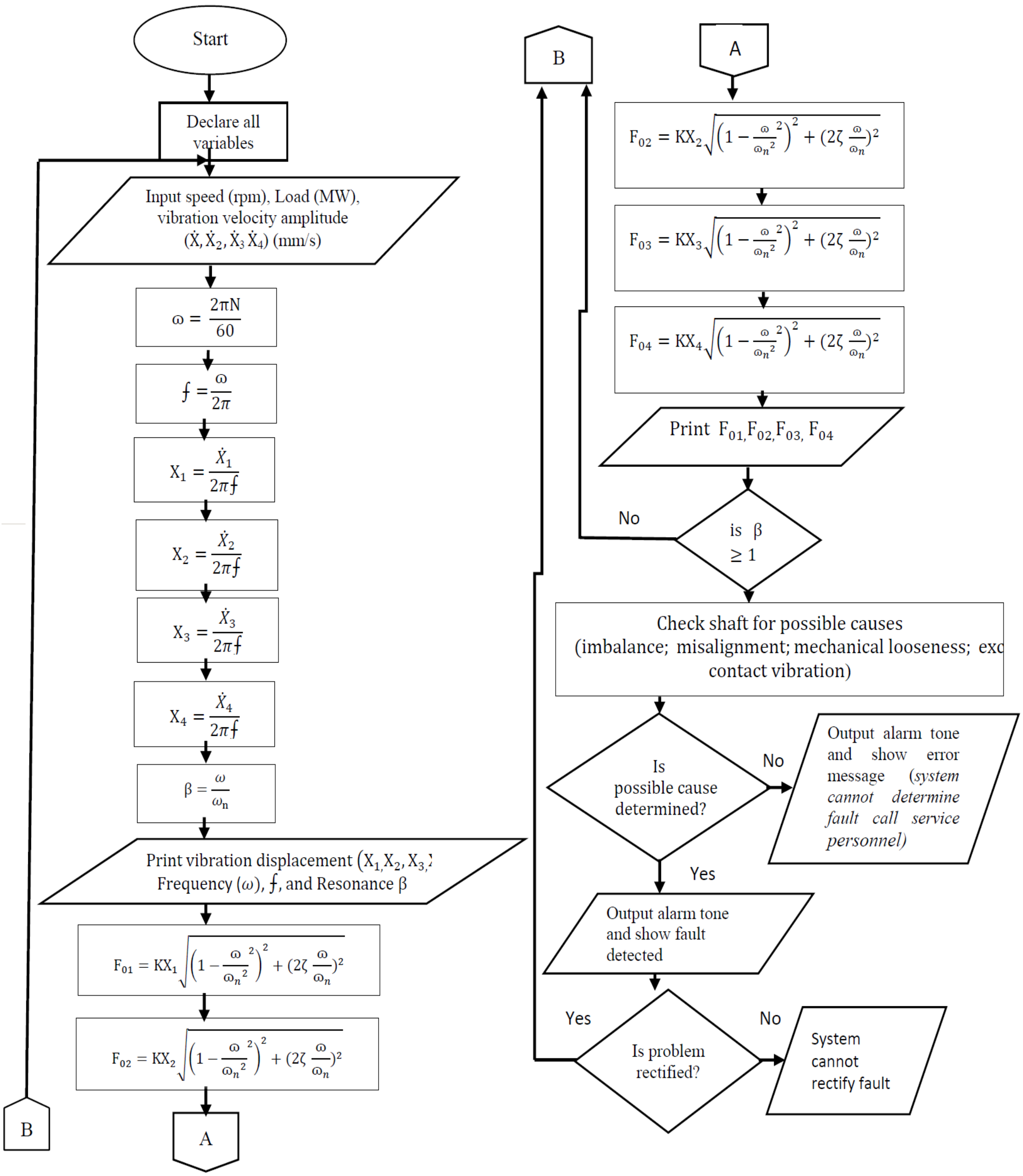

Fig. 4 Flowchart for 'FAR' program

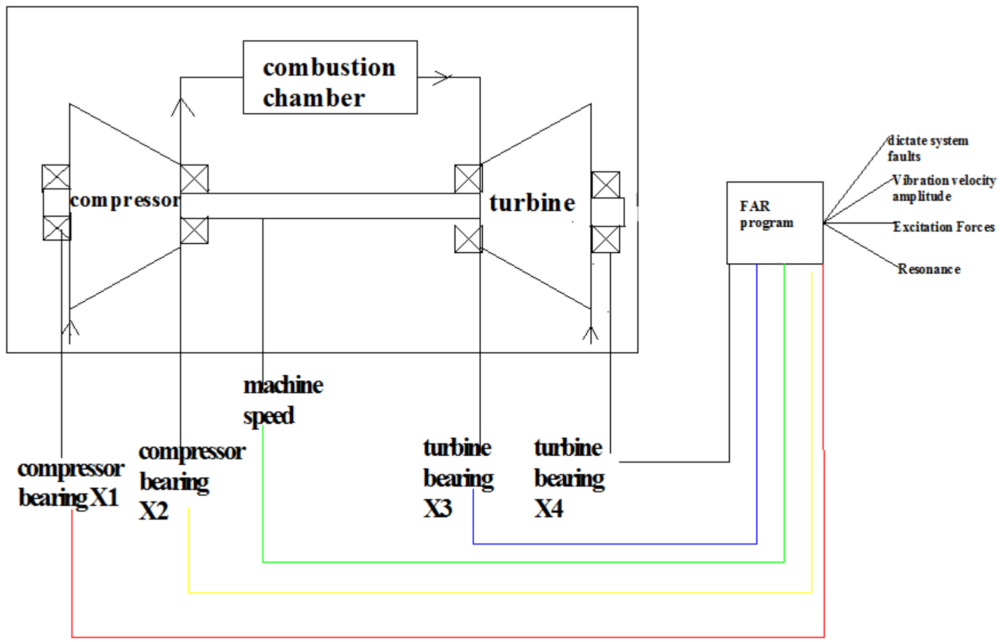

Fig. 5 Proposed online vibration monitoring with FAR

3. Results and Analysis

The steps in the flow chart on Fig. 4 were implemented to obtain the program used for monitoring the vibration level shown in Fig. 6 of the GT engine and also converting the velocity of the vibration to displacement as shown in Fig. 7. The program has the ability to raise an alarm once the vibration of the system is tending towards the reference vibration limit set by the operator. The program also has a knowledge-based code where the control system and the reference value can be changed and a historian Table 3 that keeps record of the vibration data and the condition of the GT system. From Table 3, the Table 4 was used in plotting the graphs shown in Figs. 8 to 12 was generated. These trajectories represent the vibrations on bearings 1 to 4 of GT 17 during the time under consideration, based on the data collected.

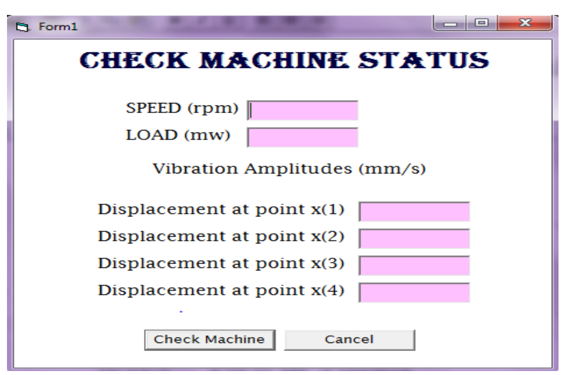

Fig. 6 FAR program for vibration monitoring

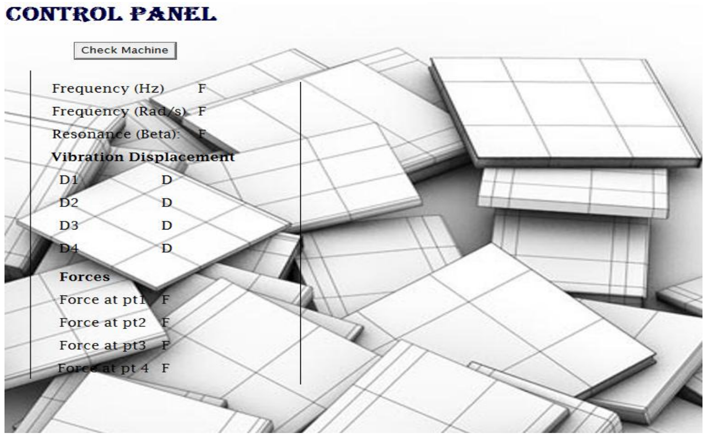

Fig. 7 Typical page of FAR program

Table 3 Historian report for the vibration readings

SN	Date	Time	BetaValue	ErrorMsg
1	10/12/2...	10:19:00 ...	0.9	Machine ...
2	10/13/2...	3:51:05 AM	0.1047619	
3	10/13/2...	4:45:03 AM	0.1047619	
4	10/13/2...	4:47:18 AM	0.1047619	
5	10/13/2...	4:52:38 AM	0.1047619	
6	10/13/2...	4:53:25 AM	0.1047619	
7	10/13/2...	5:01:56 AM	0.1047619	
8	10/13/2...	5:50:29 AM	0.1047619	
9	10/13/2...	5:52:57 AM	0.1047619	
10	10/13/2...	5:54:55 AM	0.1047619	
11	10/13/2...	5:57:32 AM	0.1047619	
12	10/13/2...	6:04:26 AM	0.1047619	
13	10/13/2...	6:18:43 AM	0.1047619	
14	10/13/2...	6:24:15 AM	0.1047619	
15	10/13/2...	8:59:31 AM	0.1047619	
16	10/13/2...	9:33:37 AM	0.1047619	
17	10/13/2...	9:34:35 AM	0.1047619	
18	10/13/2...	9:47:00 AM	0.9266939	
19	10/13/2...	9:47:18 AM	0.9266939	
20	10/13/2...	9:47:58 AM	1.047619	
21	10/13/2...	9:57:21 AM	1.047619	
22	10/13/2...	9:59:56 AM	1.047619	
23	10/13/2...	10:14:19 ...	1.047619	
24	10/13/2...	10:15:30 ...	1.047619	

Table 4 Historian report for the vibration readings

Weeks	Active load (MW)		Shaft speed, N (rpm)	Frequency, ε (rad/s)	Frequency, f (Hz)	Vibration velocity amplitude (mm/s)				Vibration displacement amplitude (mm) $\times 10^{-2}$				Excitation force F_x (N) $\times 10^4$			
						Compressor bearing		Turbine bearing		Compressor bearing		Turbine bearing		Compressor bearing		Turbine bearing	
	A	R				X_1	X_2	X_3	X_4	X_1	X_2	X_3	X_4	F_1	F_2	F_3	F_4
1	50	75	3063	320.79	51.05	4.8	6.2	0.9	-5.0	1.50	1.933	0.28	1.558	1.609	2.008	0.302	1.677
2	50	75	3074	321.95	51.23	4.6	6.2	0.9	-5.0	1.43	1.926	0.28	1.553	1.468	1.989	0.289	1.607
3	50	75	3056	320.06	50.93	4.9	6.2	0.8	-5.0	1.53	1.937	0.25	1.562	1.687	2.134	0.275	1.720
4	45	75	3063	320.79	51.05	5.1	6.5	1.0	-5.0	1.59	2.026	0.31	1.559	1.710	2.180	0.335	1.677
5	40	75	3028	317.13	50.47	5.0	6.7	1.2	-5.0	1.58	2.113	0.38	1.577	1.899	2.545	0.455	1.899
6	35	75	3053	319.75	50.88	5.2	6.5	1.0	-5.0	1.63	2.033	0.31	1.564	1.810	2.262	0.348	1.740
7	37	75	3005	314.72	50.08	5.2	6.6	1.7	-5.0	1.65	2.097	0.54	1.589	2.128	2.702	0.696	2.047
8	41	75	3056	320.06	50.93	5.3	6.6	0.9	-5.0	1.66	2.062	0.28	1.562	1.824	2.272	0.310	1.721
9	40	75	3077	322.26	51.28	5.2	6.6	0.9	-5.0	1.61	2.048	0.23	1.552	1.652	2.097	0.285	1.588
10	42	75	3074	321.95	51.23	5.0	6.5	0.9	-5.0	1.55	2.019	0.28	1.553	1.689	2.090	0.289	1.608
11	40	75	3065	321.01	51.08	5.0	6.6	1.2	-5.0	1.56	2.056	0.37	1.557	1.664	1.997	0.399	1.663
12	50	75	3077	322.26	51.28	5.0	6.4	1.3	-5.0	1.55	1.985	0.40	1.552	1.589	2.034	0.413	1.589

3.1. Analysis under bearing 1

Fig. 8 is the vibration spectrum of displacement amplitude (mm) versus frequency for bearing 1. Through the readings obtained during the engine operation, the highest vibration amplitude occurs at a frequency of about 50.88Hz. The impact of this is that the engine could be run safely at other frequencies while frequencies between 50.7 and 50.9Hz should be avoided. These frequencies are otherwise called “barred zones” for bearing 1.

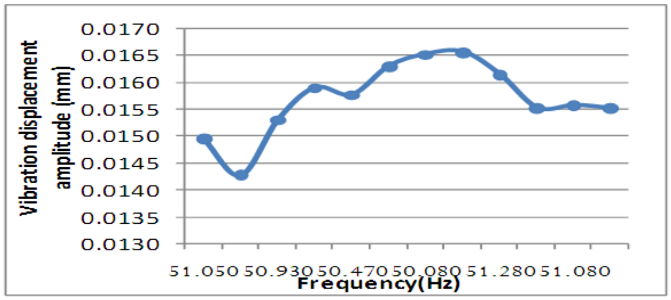

Fig. 8 CVibration displacement amplitude on bearing 1 against frequency

3.2. Analysis under bearing 2

The trajectory for bearing 2 is shown in Fig. 9. The 2 peaks for the operation of the GT on this bearing are 0.0211 and 0.0209mm. Even though these frequencies fall below the reference values, avoiding them could keep the operational parameter and the engine within the safe limits.

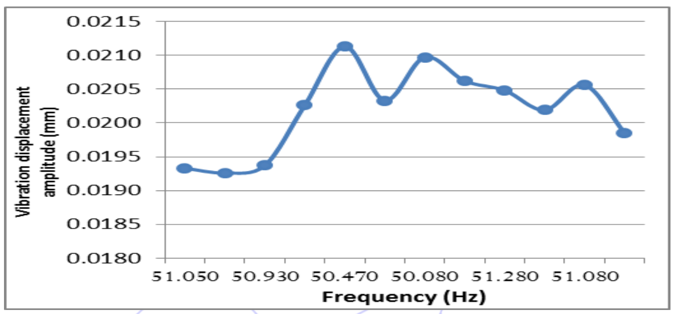

Fig. 9 Vibration displacement amplitude on bearing 2 against frequency

3.3. Analysis under bearing 3

Fig. 10 represents a graph of vibration on bearing 3 against frequency of the turbine. It was observed that the bearing 3 of the GT had the highest vibration amplitude of 0.0054mm at a frequency of 50.08Hz.

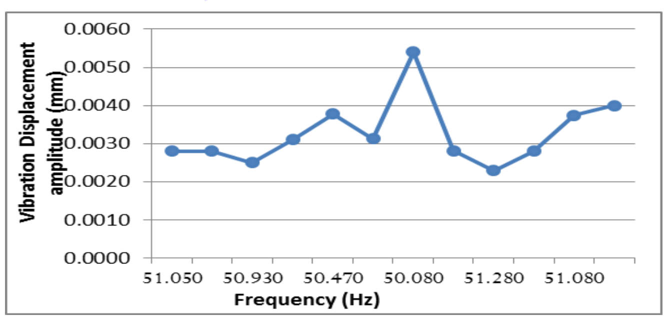

Fig. 10 Vibration displacement amplitude on bearing 3 against frequency

3.4. Analysis under bearing 4

Fig. 11 represents a graph of vibration displacement amplitude on bearing 4 against frequency of the GT. The peak vibration level for the bearing is 0.01589mm as against 50.08Hz. This portends that the bearing needs to be checked since it gives equal vibration velocity readings at different frequencies.

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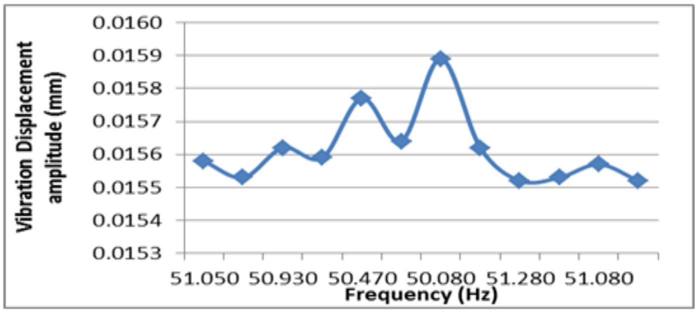

Fig. 11 Vibration displacement amplitude on bearing 4 against frequency

3.5. General Analysis

Fig. 12 represents a graph of vibrations on bearing 1 to 4 against frequency. From the graph, it was observed that the bearing 2 has the highest vibration level, which tends towards the reference vibration limit of the system. Hence, proper proactive care needs to be given to bearing 2 to avoid excessive vibration of the system.

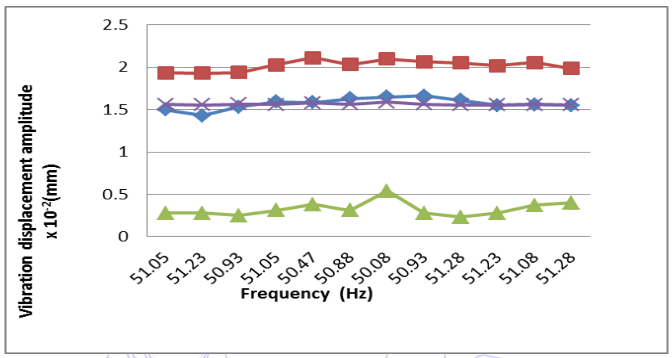

Fig. 12 Vibrations on bearings 1 to 4 against frequency

4. Conclusion and Recommendations

A research work has been carried out on the application of artificial intelligence in the operations of gas turbine. A computer program code named “FAR” for vibration monitoring was written in Visual Basic Programming language to monitor gas turbine vibrations and record data for future purposes. The program was run with data obtained from Afam gas turbine power plant. The results obtained show that the vibration levels on bearing 2 in weeks 5 and 8 were tending towards the maximum vibration limit of the gas turbine and requires attention.

On a general note, the work equally shows that trending of vibration signals from operational machinery is a very effective tool in the upkeep of engines especially the gas turbine machine of interest. The study also identified the fact that mechanical faults detection, identification, analysis and solution can be handled using the artificial neural network.

The recommendations are as follows:

(1) Far should be interfaced with gas turbine engines and other vibrating equipment.

(2) Vibration monitoring and analysis should be taken more seriously in gas turbine engine operations, and

(3) The monitoring of the vibration on the bearings carrying the compressors in gas turbine engines should be used to determine the holistic health of the gas turbine.

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