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Vibration Appraisal of a Rotor Shaft of a Marine Gas Turbine Utilizing Excel Software Based Approach

IGOMA, E.N.[1], TONLAGHA, O.R.[2]

[1, 2] *Marine Engineering Department, Faculty of Engineering, Nigeria Maritime University, Okerenkoko, Delta State, Nigeria.*

Abstract- This work focuses on the vibration appraisal of a rotor shaft of a marine gas turbine utilizing excel software-based approach for condition monitoring. The study took Ten (10) weeks during which data were measured, collated and studied. The gas turbine in which this investigation was done is the Marine Type Engine: Aero – Derived (ABB: ALSTOM): 75MW used in Afam III Power Station. The results obtained show that vibration causes the rotary parts like the four bearings in the rotor shaft to misaligned and load (energy) reduction. It was recommended that proactive condition-based maintenance is required so as to avert unexpected unwarranted expenditures for yet – to – be conventional maintenance.

Indexed Terms- Vibration, Gas Turbine, Rotor Shaft, Excel Software, Condition Monitoring

I. INTRODUCTION

Vibration analytical appraisal of rotary machineries like the gas turbine is generally utilized to guarantee untimely faulty diagnosis analysis before these machines may downtime. Gas turbines are very vital rotary machinery which also has relationship with the internal combustion engines which are intensively utilized in the oil and gas industrial plants. Indeed, it is used to accomplished the principal function of producing mechanical energy in the form of shaft vie the kinetic energy of the gas created in the combustion chamber [1][2][3][4][5][6][7]. Meanwhile, the gas turbine when in operation are principally influenced by the rotor shaft faults like misalignment, looseness, distortion, bearing vibration, unbalance and eccentric journal and so on [8]. The capability for the identification of these components that has the faults under intensive condition permits the users to take suitable controllable steps in order to rectify the faults and thereby provide an appraisal for ahead-of-time detection of the engine faults that could take to ruinous deterioration [9].

II. LITERATURE REVIEW

Rotor unbalancing is the commonly known cause of machine vibrations. However, most of the rotary machinery problem can be resolved by the used of rotor balancing and misalignment. Meanwhile, a very small amount of unbalance can result to a serious problem in high-speed rotating machinery like gas turbine [10]. These problems are reduced by an appropriate balancing of rotary element and appropriate alignment of the driven shaft. But if the problems persist, they will lead to bearings, seals, gears and couplings failure. Due to this conspicuous factual fact, condition monitoring is required. Some approaches to condition monitoring are discussed below.

The proactive condition monitoring of a rotor shaft of a gas turbine using vibration analysis was conducted by [11]. The work gives analysis of a case of condition monitoring through its thermodynamic and rotary shaft vibration. Selectively, data were assembled from an industrial gas turbine for detection of the variations in the operational conditions of the plant which helps in putting place the controllable means of the plant. The Java Computer Programming Language was used for the simulation and analysis of data obtained. Hence, the study revealed that vibration reduces the active load and a controllable monitoring maintenance programme is needed thereby debarring the unwanted vibration effect.

Vibration – based crack diagnostic diagram in a rotary shaft during accelerated resonance was investigated by [12]. The study investigates analytical and mathematical dynamic reaction of a cracked Jeffcott rotor passing through a vital speed with invariable -

constant acceleration. The results show that the developed framework gives enablement to analyze the cracked rotor dynamical responses with /without weight ascendancy.

The vibration analysis of tie rod and bolt rotor utilizing FEM studied by [13]. In the study, a new methodical approach of the finite elemental model was initiated for a distinctive tie-rod/tie-bolt rotor. However, the study presumed that the flange of the disk as a supernumerary rotor which encircles the principal rod and the Timoshenko shaft element was utilized in the modeling of the tie-rod/tie-bolt rotor. The results expose that the axial force due to compression in the rotor has diminution effect on the critical speeds. Also, there is a significant agreeable correspondence when the results when comparably compared with the experimental data obtained in the literature.

The investigation of the vibration analysis of turbo generator in Kota super thermal power station was done by [14]. The study presents an analysis of steam turbine vibration monitoring, monitoring tools, vibration analysis of turbo generator of 195MW: Unit-7, and how to assess the performance of the turbine. An elaborated report on the vibration of bearings comparable to the bearing temperature was accomplished using IRD 880 instruments. The results achieved show that the maximum vertical displacement measured in the bearing number is 4 and the maximum axial velocity measured in the bearing is number 7.

The application of the artificial neutral network (ANN) and wavelet transform (WT) for vibration analysis of combined faulty unbalance and shaft bow was performed by [15]. The study investigates ANN and WT utilized for the prognostication of the effect of combined faulty of unbalance and shaft bow on the frequency parts rotary machinery vie vibration signature. Precisely, it was discovered that marquardt algorithm was far better effective when compared to the other techniques and when used for the diagnosing and measuring of faults. The overall achievements rates based for unbalance, shaft bow and combine faulty unbalance – shaft bow are 99.78%, 99.81% and 99.45% respectively.

The advancement of vibration condition monitoring system using optical sensors for generator winding integrity of power facilities was conducted by [16]. The work focuses on diagnosing system vibration - condition monitoring advancement for a stator and rotor winding integrity appraisal of 100MW gas turbine. The vibration condition monitoring system is needed for the improvement of the operational and the maintenance cost savings and also in terms of reliability in the power plant.

Furthermore, after proper analysis of the various approaches enumerated and paraphrased above, this work will study the vibration analysis of a rotor shaft of a marine gas turbine using Excel Software based approach to analyze the characteristic features of the turbine speed on the load and the vibration displacement amplitude.

III. METHODOLOGY

The investigation took Ten (10) Weeks during which data were collated and studied. The data were obtained from a marine type engine: Aero - Derived of 75 MW capacity used for generating power and the Plant is in Afam, Oyigbo Local Government, Rivers State [11]. The average of the data for the turbine was collated and those that were not obtained directly were calculated using appropriates assumptions [17].

ANALYTICAL MODEL OF VIBRATION OF THE ROTOR SHAFT OF THE MARINE GAS TURBINE ENGINE

Vibration happened rectilinearly or torsionally. The torsionally kind of vibration is of most consequential effect to rotary systems. Because, it is of the form of periodical angular motion of elastic shafts with circular rotors firmly connected to the gas turbine rotors [18]. Practically, a spring – mass system movement strained to the vertical direction coupled with the activation of unbalanced rotary machinery. From Figure 1.1, the unbalance is denoted by eccentric mass (m) with the change in the eccentricity (e) with a rotating angular speed ($\boldsymbol{\omega}$). Given, ‘x’ equals to the displacement of the non-rotating mass (M – m) from the static equilibrium position[11][17]:

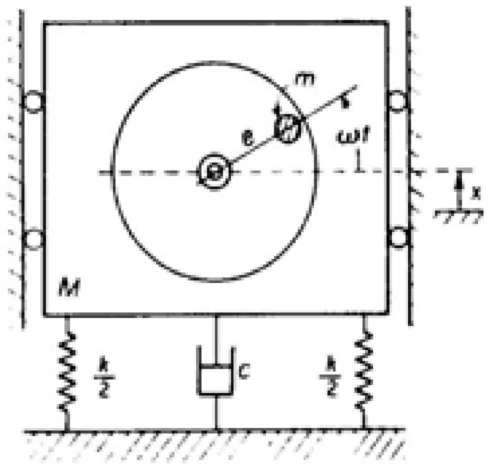

Figure 3.1: Representation of Harmonic Disturbance Force Due to Rotary Unbalanced [11][17][19].

$$x = cx + \dot{e}\sin\omega t \tag{1}$$

Hence, equation of motion is given by;

$$\ddot{m}x + k\dot{x} + kx = me\omega^2 \sin\omega t \tag{2}$$

From [20],the vibration displacement Amplitude (X)

$$= \frac{\boldsymbol{m\omega^2}}{\sqrt{(\boldsymbol{k - me\omega^2}) + (\boldsymbol{c\omega})^2}} \tag{3}$$

Meanwhile, from equation iii, X_1, X_2, X_3 and X_4 can be deduced;

$$X_1 = \frac{me_1^2}{\sqrt{(k - me_1^2)^2 + (c)^2}} \tag{4}$$

$$X_2 = \frac{me_2^2}{\sqrt{(k - me_2^2)^2 + (c)^2}} \tag{5}$$

$$X_1 = \frac{me_3^2}{\sqrt{(k - me_3^2)^2 + (c)^2}} \tag{6}$$

$$X_1 = \frac{me_1^2}{\sqrt{(k - me_1^2)^2 + (c)^2}} \tag{7}$$

IV. RESULTS AND DISCUSSION

- RESULTS

Table 3.1 contains the values of the turbine speed which was obtained from direct readings of the turbine logsheets while the values of vibration amplitude on the bearings were obtained from computation using Equations: 4, 5, 6 and 7 respectively. Precisely, the assumption of angular speed for the process started by the assumption of a unit amplitude for the free mass. With respect to the assumption made, the amplitude coupled with inertia force of the leftover masses was then computed. The oscillatory frequency of different load was achieved by the used of the rotary speed of the turbine. Meanwhile, it must not surpass the original frequency thereby debarring the system from resonating. Mathematically, by substituting the speed of the turbine engine at different eccentricity comparable to the place of the bearing taking from reference point, the comparable load, frequencies and various vibration displacement amplitudes were obtained. The values obtained after computation were compared to simulated data of [11] and they show correspondence correlation. The characteristic features for the turbine speed on the load, frequencies, and vibration displacement amplitudes on the bearing were plotted using Excel Software as shown in Figures 4.1-4.3 respectively.

Table 3.1: Calculated and Compared Vibration Results

S/No.	Turbine Speed (RPM)	Load (MW)	Frequency (Hz)	X_1 (mm/s)	X_2 (mm/s)	X_3 (mm/s)	X_4 (mm/s)
1	3023	28	50.40	3.32	2.23	2.77	1.19
2	3025	50	50.41	3.43	2.28	2.67	1.12
3	3027	52	50.44	3.34	2.45	2.80	1.13
4	3028	51	50.47	3.22	2.26	2.65	1.15
5	3029	52	50.48	3.25	2.25	2.40	1.14
6	3030	48	50.49	3.40	2.42	2.54	1.13
7	3030	12	50.49	3.52	2.37	2.70	1.19
8	3032	53	50.51	3.24	2.24	2.64	1.15
9	3032	48	50.51	3.38	2.40	2.80	1.16

10	3033	49	50.54	3.34	2.39	2.78	1.11

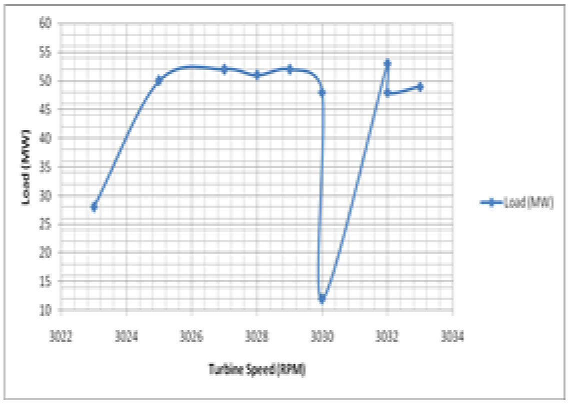

Figure 4.1: Characteristic Feature of Turbine Speed on the Load

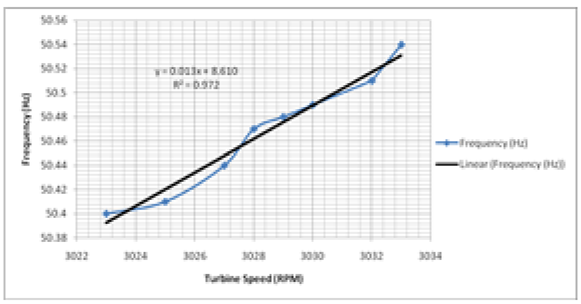

Figure 4.2: Characteristic Feature of the Turbine Speed on the Frequency

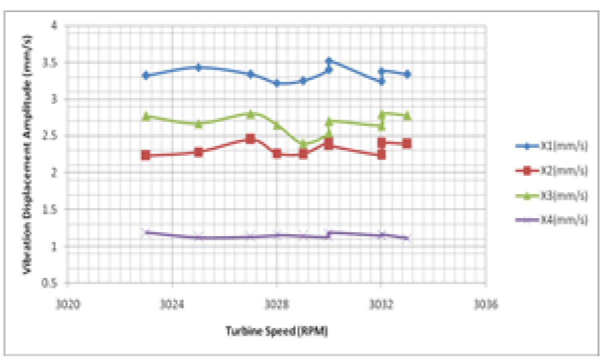

Figure 4.3: Characteristic Feature of the Turbine Speed on the Vibration Displacement Amplitude

- DISCUSSION

Figure 4.1 shows the characteristic feature of the turbine speed on the load. It shows that as the turbine speed increases the load increases and haphazardly slowed down and later picked up. This depicts that vibration causes haphazard movement of the rotor shaft thereby causing reduction in energy. Because, in a normal situation, the turbine speed verses the load supposed to produce a straight line graph. Figure 4.2 represents the characteristic feature of the turbine speed on the frequency. The graph illustrates that as the turbine speed increases the frequency increases slightly (snake like). Also, it shows that the turbine speed is directly proportional to the frequency and it increases by 0.013% with a certified square root of 0.972. Figure 4.3 depicts the characteristic feature of the turbine speed on the vibration displacement amplitude showing how the unbalance rotor shaft oscillates with their bearings (1-4) as the turbine speed increases. It also shows that the vibration displacement amplitude of the different bearings were oscillating at almost the same range while the turbine speed increases thereby showing danger that necessitates condition monitoring to avert gas turbine downtime that can occur due to shaft misalignment.

CONCLUSION

The paper which focuses on vibration analysis of a rotor shaft of a marine gas turbine thoroughly investigated how Excel Software Approach will be used for condition monitoring in order for the gas turbine to perform effectively. After the whole analyses of the characteristic features, it was found that vibration causes load (energy) reduction in the marine gas turbine. Precisely, the rotary parts which include the rotor shaft and the four bearings are the most that were affected by the vibration thereby resulting in misalignment if not early check will cause engine failure.

RECOMMENDATION

It is recommended that a proactive condition monitoring based maintenance is required so as to save unexpected expenditures for yet – to – be conventional maintenance.

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