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Vibration Analysis and Condition Monitoring Of a Marine Gas Turbine for Efficient Performance

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ABSTRACT: *The vibration analysis and condition monitoring of a marine gas turbine for efficient performance has been conducted. The work look at vibration displacement amplitude in which collations of data were done through direct measurement from the real gas turbine and generator performance logsheet. The gas turbine under investigation is the Trans- Amadi Gas Turbine utilized for power generation. The procedural process for the vibration data collection continues for twenty-five weeks in which each week a minimum of five days hourly readings were taken and the average taken and recorded for that week. The revelations from the study reveals that vibration influence the performance of the gas turbine. Precisely, the number of deficiency apportioned to the gas turbine is a function of the vibration displacement amplitude on the rotor-shaft and the four bearings. Observably, the bearings at the turbine bears very high force as a result of the power generation thereby bring about very high vibration displacement amplitude. The vibration appraisal also displays that using vibration for diagnosis in machinery like gas turbine is the best effective way for detecting together with identifying deficiency in bearing thereby making it very vital in order to debar ruinous breakdown of the gas turbine plant. It is recommended that performance appraisal and condition monitoring of the plant show be done steadily in order to avoid unscheduled shutdown.*

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I. INTRODUCTION

The gas turbine is an important and major source of generating power for different demands likeaerospace, maritime, electricity, compression of gases, coupled with high – scale services. The vibration in gas turbine is impacted by mechanical imbalances of the sundry engine compartments in addendum to the different mechanical faults occurrence in the gear boxes, bearing, pumps, gears, compressor rotors together with stators coupled with the shafts [1][2][3][4].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[5]. The ability of identifying the constituent that has the faults under intense condition allows the users to choose appropriate measures in order to correct the faults and thereby providing an appraisal for ahead-of-time detection of the engine faults that could take to ruinous deterioration[6][7]. The phenomenon of impairment in gas turbine engines can make their operation uneconomical and dangerous. A Cognition of the actual true state of the gas turbine is vital to evaluate the performance ability so that it can meet the operational and maintenance demands. An early detection-diagnosis of the cause of the declension allows suitable maintenance activity to be carried out which can restore the engine performance and in-adversely ensure a dependable-safe operation. The performance efficiency of gas turbine power plants is immensely essential to the engineers and every nation that have it. Meanwhile, organizations have therefore, ventured into sundries maintenance management strategies to guarantee high reliability of plant availability. In other to meet these myriads of demands, checking the growing energy demands and developing a new maintenance approach for gas turbine plant availability becomes necessary[8].The hypothetical rudimentary of this idea is sum up in Fig **1**[9][10].

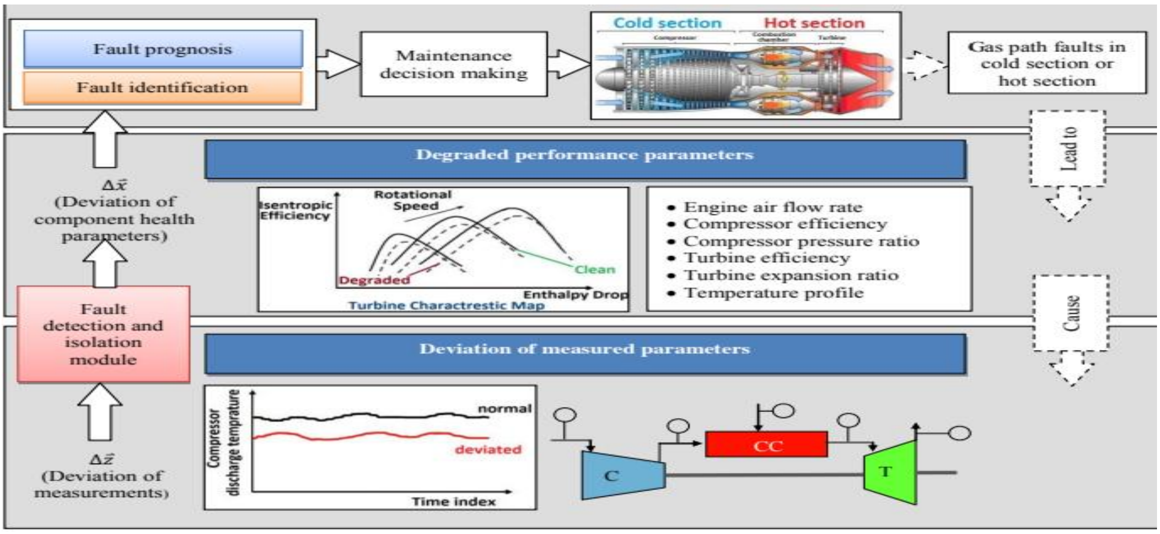

Fig. 1 Gas turbine performance Based condition monitoring [9]

Precisely, in-view of these conspicuous challenges, it became pertinent to develop a new mythological and workable strategy to salvage the plant from untold disaster through the consideration of its rotor shaft vibration. However, the fault detection, identification together with analysis is linked to the rotational motion. This depends on the turning speed of the rotor coupled with the dynamic interaction betwixt the shaft and other items that is in contact with it. The use of vibration measurement for the optimization and the diagnosis of problem with machinery that have immense impact on the output, quality and production stoppage [11].Actually, when fault happens, the gas turbine parameters are subjected to alteration. These alterations depend on the degree of faults and interaction with the prevailing parameters. Meanwhile, efficient condition monitoring can be investigated when the gas turbine is in operation and it is called on-line while the off-line is when it is down or probably showdown (i.e. not in operation). Precisely, it is at this point that optimization of a model-based monitoring is very essential and the use of signature analysis can be implored.Vibration signature analysis entails monitoring the health condition of the gas turbine rotor shaft by recording the systematic signals obtained in the form of mechanical vibration, relative displacement and so on [12]. Notwithstanding from study, it has been seen that the rotor shaft is one of the most pertinent parts of the gas turbine engine systems. Because it has several other components joined to it whose health condition are monitored through vibration signature analysis. In addition, vibration analysis is comprised to be the processing of the sensed vibration signal displacement, velocity or acceleration amplitude. Sensing: detection entails the processing of vibration signal using an accelerometer or velocity transducer. It is very pivotal to take into consideration the type and the range of transducers or pickup for capturing vibration signals [13]. The vibration signature of a machine is the feature: characteristic pattern of the vibration it produces when the machine is in operative state [14]. The real signal from the vibration transducer can be regarded as the signature but the spectrum of the vibration signal is what is called signature. Efficient vibration analysis commences by getting an exact signal from the standard vibration transducer with the help of an accelerometer. The analog signal is changed into digital signal with the aid of a converter. The obtained digital signals can then be treated straightaway or can be treated using some formula depending upon the need of the user [15]. The diverse forms of measured signal possible condition of machinery are wavelength transform, genetic algorithms, neutral network, fuzzy logic and machine support vectors and they are largely used in laboratory test established to detect, locate and quantify the faults. And based on it, the lifespan of the machinery can be prognosticated. However, every fault has a particular signature in the measured signal and that is the very commodious and economical mode to discover potential faults thereby optimizing it effectively. Economically, these rotating machines (rotor shaft) have high capital cost and thus, developing a condition monitoring techniques is very pertinent [16]. Precisely, the Trans – Amadi gas turbine power station Phase II (GT-II) operates on an open cycle consisting of a compressor, a combustor and a turbine combined in series as shown in **Fig 2** The combustor increases both the temperature and the specific volume of air [17]. A net positive shaft work transfer is produced because the negative shaft work transfer required for powering the compressor is less than the positive work transfer produced by the turbine [18].

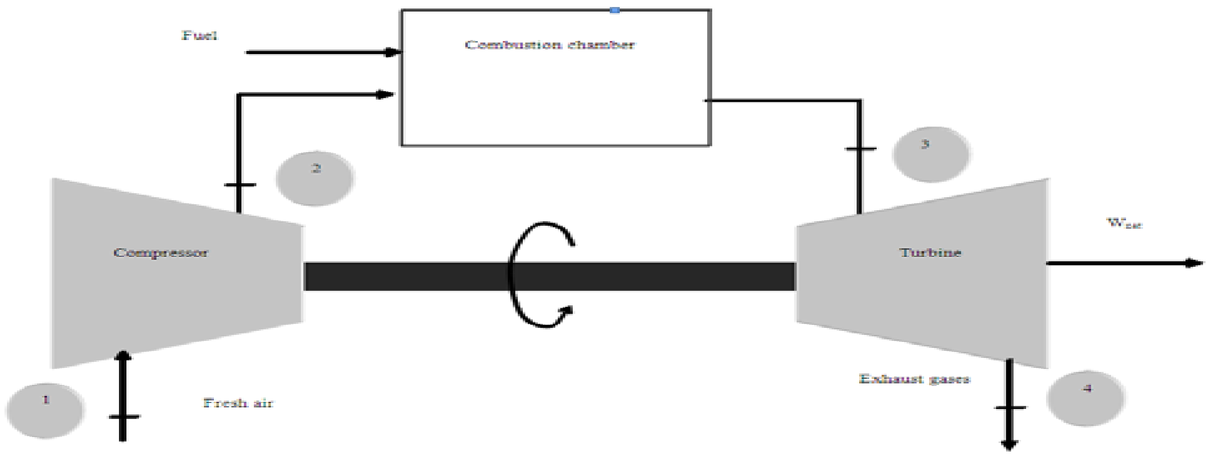

Fig. 2 Simple-Cycle, Single-Shaft Gas Turbine [7][13**]**

Also, the major sections of the Trans-Amadi gas turbine power station phase II depicting the single shaft line is shown in **Fig. 3** [10].

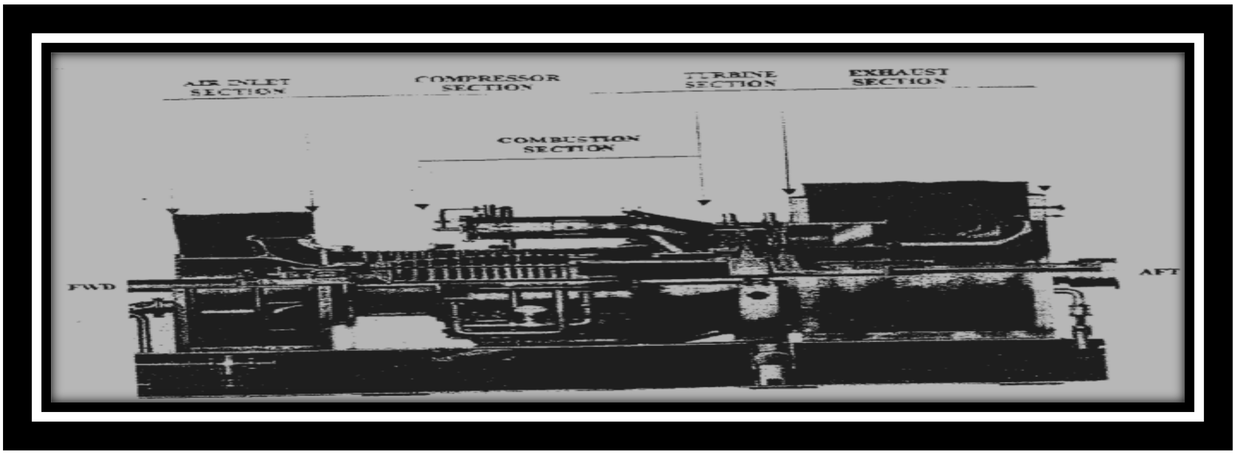

Fig 3: Major Sections of the Trans-Amadi Gas Turbine Assembly depicting the Shaft line [17]

Gas Turbine Engine Condition Monitoring: The condition based monitoring of gas turbine engine refers to as a set of chore to accomplish the detection of the beginning of a fault of the machine: gas turbine engine for the preventative of any failure or catastrophic breakdown that will stop the normal operational schedule of the machine. Also, the condition based monitoring entails online and offline monitoring [19]. The offline monitoring involves measurements produced as a free run or synchronal sampling and the online monitoring is performed at real time. Both techniques have their merits and demerits. The offline technique normally provide a simplify analyzers (data collector). Chiseled measurements path with well – organized apparatus registration, utilize medial technical workforce whose primary job is the exactness in the measurement procedure. The primary merits of this type of equipment are the choice but it is essential to emphasize that data collectors are primarily advantageous only for the assessment of the vibroactivity. While, the vibration online diagnosis is sometimes referred to as monitoring module because it assure unaltered inspection of the technical state of the gas turbine engine which entails records analyzer, alarms and prediction. The online monitoring permits main signs of the permanent, technical state varying with the potentiality of the trendy remarks. The structural depiction of the vibration diagnostics of a gas turbine engine comprising of both techniques is shown in **Fig. 4**[17][20].

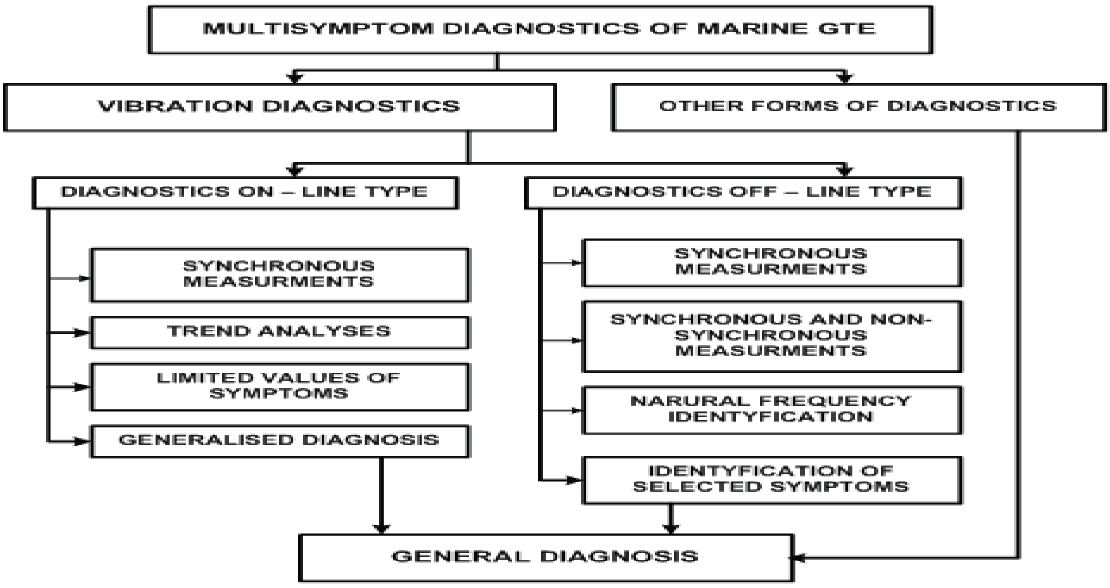

Fig .4 The Structural Depiction of Vibration Diagnosis of Both Offline and Online Monitoring [21]

However, the condition monitoring system depends on diversity of sensors that can aid in observing when the machine is degrading [22][23][24]. Precisely, an increment in the justification of fuel costs to maintenance has moved underline from mechanical health to monitory engine performance [25][26]. The conditioned based maintenance refers to measurable data on the gas turbine engine that is being monitored. The data differs as the health or condition of the gas turbine gets impaired. When the change in the qualities of the data is ascertained, the analysis and the diagnosis is disposed to clear the problem [28]. And by the utilization of this kind of maintenance, one doesn't need to wait until the occurrence of the failure but considerable condition of the gas turbine engine is put in place to maximise the usage of the apparatus. There are different condition monitoring methods such as Vibration Monitoring, Oil Analysis, Manuel Inspection, Current, Thermal, Corrosion and Performance Monitoring[27][28].

Vibration Analysis: This is one of the established and effectual technological methods being utilized in condition monitoring. In measuring vibration, vibration transducers are used in agreement with the frequency limit or range as the case may be. The vibration measurement is usually performed in gearbox, turbines, bearings and shaft. Several variety of sensors are being used in measuring the vibration and they include the displacement sensor, velocity sensor and acceleration sensor **[**22]. One of the merits in vibration analysis is that it can discover emerging problems before they become too severe and cause unplanned downtime. This is achievable by the conduction of regular monitoring of machine vibration either at planned time or on uninterrupted bases. Regular vibration monitoring can discover degrading or bad bearings, mechanical looseness or gears that are broken [29]. Also, vibration analysis can expose misalignment and unbalance before these conditions give way for bearing or shaft degradation [30]. The vibration measurement is an effectual, uninstructive technique in monitoring condition of machines during upstarts, shutdowns and normal operational activities. Vibration analysis is utilized mainly on rotary equipment like steam and gas turbines, pumps, motors, compressors and so on, and this rotary equipment require other methods to amply monitor their operations. The vibration analysis system normally comprises of four (4) fundamental parts namely: Signal pickups (transducer), a signal analyzer, analysis software and a computer for data analysis and storage.

Vibration Analysis and Measurement Equipment

Online Data Collector: Accurate machines are virtually ever furnished with continual online monitoring systems. Precisely, here, the sensors like Eddy Current probes introduced in Turbomachinery are unalterably fixed on the machines at appropriate measurement place and is plugged to the online data collection and analysis apparatus. Automatically, the vibration data are obtained for each level and the appraisal is shown on the local monitoring equipment. This capability gives timely faults detection and provides defensive moves on exactness of the machinery. A defensive move appropriated by online data collection and appraisal equipment is in the pattern of producing alarms to caution the person operating of anomalous condition. Vibration appraisal and the managerial of database software can also be networked to many computers utilizing the local area network and the wide area network to admit many users to do condition monitoring. Hence, the machines at different physical centers can perform the condition monitoring from that single center and the information can be transmitted from the host computer to the local monitoring unit. The advantage is that this type performs monitoring on critical machinery, measurements are taken automatically without the hindrances of human and it provides almost immediacy of detection of faults while the disadvantageous part is that the machine is expensive and it requires special skills and online data collector is depicted in **Fig.5** [31].

Fig.5: Online Data Collector [31].

Hand-held Vibration Meter: This equipment is not expensive and it is easily used during vibration program. Practically, an operator of plant together with vibration technicians carries hand – held meters and analysers when on routine checks. When these are closely – held in juxtaposition with the machine under investigation, they produce a show of the vibration stratums either in analog or digital form. The read – out produces contiguous information that can be utilized in the determination of the vibration stratums whether normal or anomalous. The hand – held vibration meters are distinctively battery – powered and is utilized as accelerometer for sensing. Occasionally, a velocity pick- me - up is utilized. The equipment can give the following data depending on the particular mode: acceleration (pk)(g); velocity(pk – rms) (mm/s); displacement(pk – pk)(microns or mils) and bearing condition (gSE, dB and others). The advantage in using this hand – held vibration meter is that they are commodious and conciliatory, and it needs very small skilful usage while the disadvantage part is that they lack data storage abilityand the equipment is illustrated in **Fig. 6** [31].

Fig.6A hand – held Vibration Meter [31]

Diagnosis of Common Vibration Problems

Unbalance: An unbalance is refer to as by the International Standard Organization as “the condition which exist in a rotor when the vibrating force is conveyed to its bearings as a result of centrifugal forces”. And it can also be described as: “the mismatched distribution of mass about a rotor rotary center line”. The vibration due to unbalance of a rotor is plausibly the most common machine’s abnormality. It is fortuitously also very simple to discover and correct. The unbalance is a situation when the geometric centre line of a rotating axis does not agree with the mass center line. A pure unbalance will produce a signal at the rotating speed and preponderantly in the radial direction. The rotating center line is referred to as the axis about which the rotor will rotate if not forced by its bearings and it is also known as Principle Inertia Axis: PIA whiles the geometric center line: GCL is the physical center line of the rotor. However, a state balance occurs when the two center lines are in agreement with each other. The types of unbalance that can be faced on machinery are: static unbalance, couple unbalance and dynamic unbalance respectively and are depicted in **Fig.7**. The overhung rotors can have both static and couple unbalance and must be tried and fasten utilizing analysers [31][32].

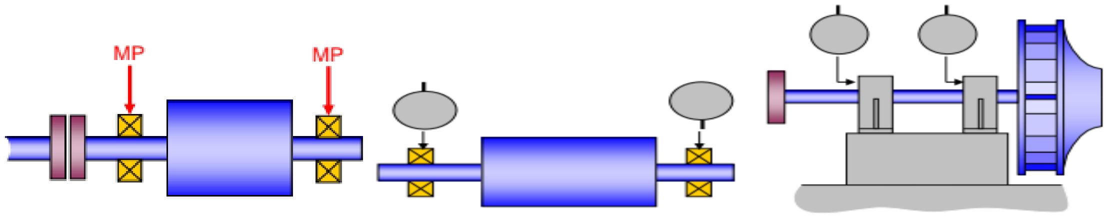

Fig.7: Pictorial View of the Static and Couple Unbalances with OverhungRotor [32]

Eccentric Rotor: The eccentricity happens when the center of rotation is at beginning from the geometric center – line of a gear, bearing, sheave, motor armature or any other rotor. The maximal amplitude happens at one revolution per minute (1RPM) of the eccentric constituents in a path through the centers of the dual rotors. Notwithstanding, in eccentricity, the phase perusal vary by 0^0 or 180^0 when calculated in the vertical and horizontal directions and a trial to balance the eccentric rotor sometimes results in the reduction of the vibration in one direction but increases the vibration in the radial direction dependent on the sereneness of the eccentricity as shown **Fig.8**[32].

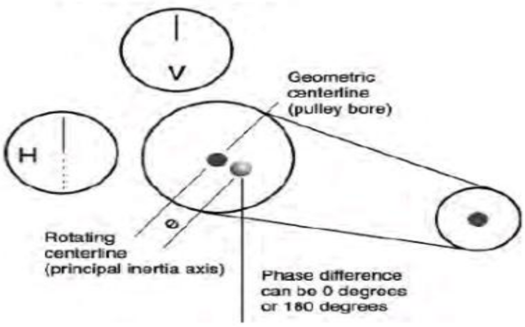

Fig. 8 Depiction of Eccentric Rotor [31]

Bent Shaft: The vibrations in the radial likewise in axial direction will be larger when bent shaft is engaged. Precisely, the axial vibrations maybe above the radial vibrations and the Fast Fourier Transform will unremarkably have the constituents of the amplitude of one revolution per minute (1RPM) that is ascendant, then the bend is near the shaft center and the amplitude of double revolution per minutes (2RPM) that is ascendant, then the bend is near the shaft end respectively and it is illustrated in **Fig. 9**[31].

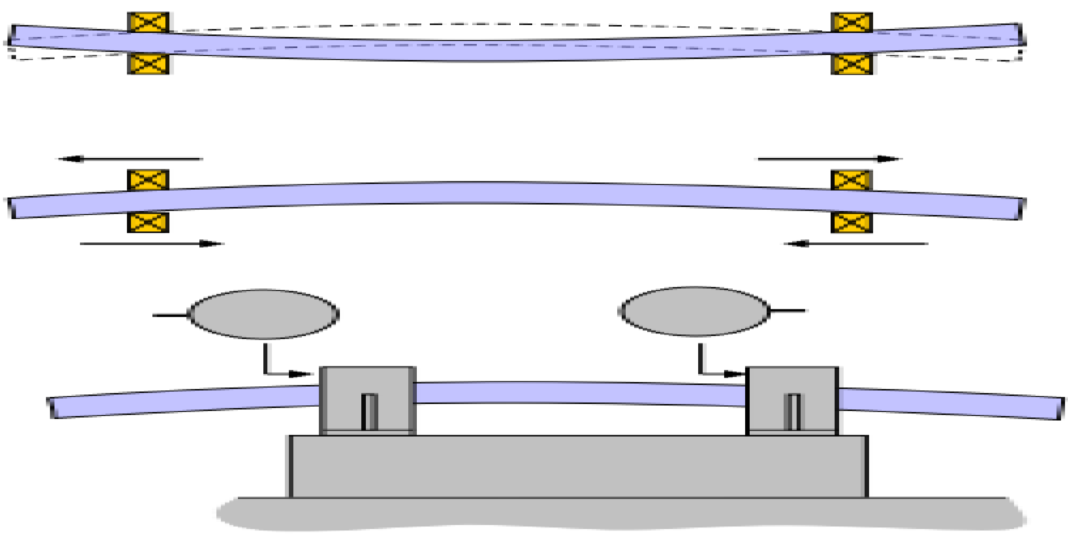

Figure 1.9: Depiction of a Bend Shaft [32]

Misalignment: This is a major source of machine vibration. Meanwhile, some machineries has - been integrated with self – positioning bearing and flexible couplings that can take very a trifle of misalignment. Nevertheless, despite these facts, it is not rare to come across very high vibrations due to misalignment. The misalignment is categorize into two namely - angular misalignment: the shaft center line of the two shafts come together at an angle with each other and parallel misalignment: the shaft center line of the two machinery is parallel to each other and have an offshoot and it is shown in **Fig.10** and **Fig.11** respectively[31][32].

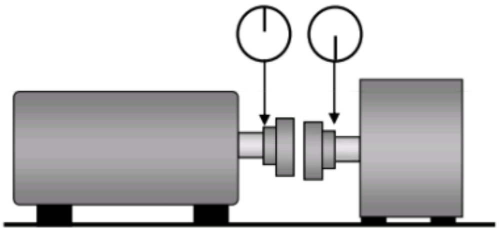

Figure 1.10: Digram of Misalignment [32]

	RPM	Alignment Tolerance inch [mils] acceptable	excellent
Short "flexible" couplings			
Offset	900	6.0	3.0
	1200	4.0	2.5
	1800	3.0	2.0
	3600	1.5	1.0
Angularity (gap difference at coupling edge per 10" diameter)	900	10.0	7.0
	1200	8.0	5.0
	1800	5.0	3.0
	3600	3.0	2.0
Spacer shafts and membrane (disk) couplings			
Offset (per inch spacer length)	900	2.0	1.2
	1200	1.5	0.9
	1800	1.0	0.6
	3600	0.5	0.3
Angularity [mrad]		[mrad]	
	900	2.0	1.2
	1200	1.5	0.9
	1800	1.0	0.6
	3600	0.5	0.3
Soft foot	any	2.0 mils	

Figure 1.11: Tabulation Depicting Different Types of Misalignments and Alignment Tolerances Based on Experiment [32].

Looseness: The looseness otherwise known as mechanical looseness happens in three places - the internal assembly looseness, looseness at the machinery to based interface and structural looseness respectively. The internal assembly looseness is experience betwixt the cap of a bearing liner, the rolling element bearing, a sleeve or an impeller on the shaft. The internal assembly looseness is cause by the inappropriate fittings betwixt the constituent's components which generate several harmonics in the Fast Fourier Transform due to the non – linear signal response of the loose component to the excitation forces from the rotor. And the looseness at the machinery to based interface is accompanied with the loose pillow block bolts, cracks in the frame form or bearing base. While the structural looseness as the name implies is caused by structure's weakness in the machinery foundation or feet. This can also happen due to impaired grouting loose hold down bolts at the structure or stand together with the frame and it is illustrated in **Fig.12**[31].

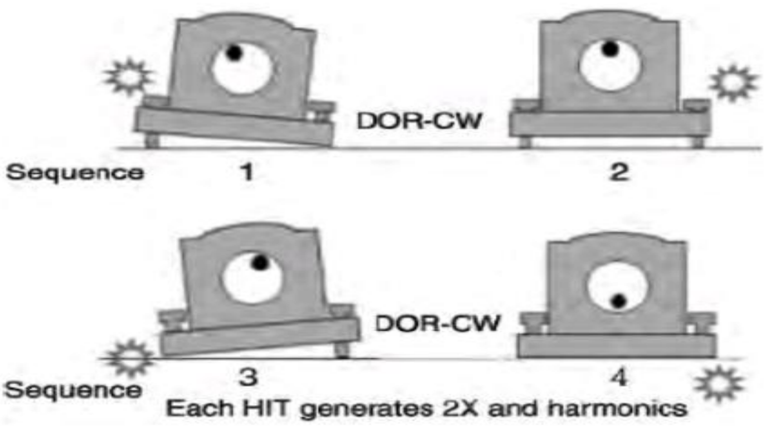

Fig.12Depiction of Looseness [31].

II. METHODOLOGY

Precisely, this work will look at the vibration displacement amplitude andthe collations of data were done through direct measurement from the real gas turbine and generator performance Log sheet data. The log sheet data obtained for the gas turbine and generator cover a period of twenty five weeks when the engine was in operation. The marine gas turbine engine under investigation is the 25MW Trans Amadi Gas Turbine used for power generation application [17]. The essence of obtaining these data is for the purpose of conducting a real field assessment of the peculiar constituent workability, which may be effectively monitored. Also, in the treatment of data, the computation of average values of daily parameters for every week was done for the twenty five weeks. After proper computation of the various averages of the vibration analyses parameters, a simulation model was develop through EXCEL SOFTWARE PROGRAM for the rotor shaft vibration utilizing the vibration displacement amplitude of the four Bearings, A, B, C and D.

Modeling the Gas Turbine Engine Rotor Shaft System

The gas generator of the Trans – Amadi Power Station II, Unit II: MS5001 Nuovopignone engine operates on a single shaft system mechanism and its rotational speed is 3000rpm to 7600rpm.In modeling the single shaft gas turbine, the following assumptions were considered:

i. That the system is a single – degree – of – freedom
ii. That the system is linear since the weight of the compressor and turbine compartments are lumped as a mass acting at the centre of the shaft linked by bearings at both sides of the assembly iii. That the damping of the system is viscous only
iii. That the system is running at a constant speed.

Analytically, utilizing the free body analysis: Single Degree of Freedom together with basic engineering principles, the relationship to each other components can be analyzed considering the unbalance mass ‘m’ positioned at a radius ‘r’ together with rotor spinning at angular velocity ‘ω’. The unbalance force can be computed taking cognizance of the displacement of rotor shaft in the reaction of the unbalances as it is depicted in **Fig. 13.** While the force vibration experienced as a result of damping of the system together with the vector in relation to the forced vibration and damping is illustrated in **Fig. 13** and **Fig. 14** respectively [33].

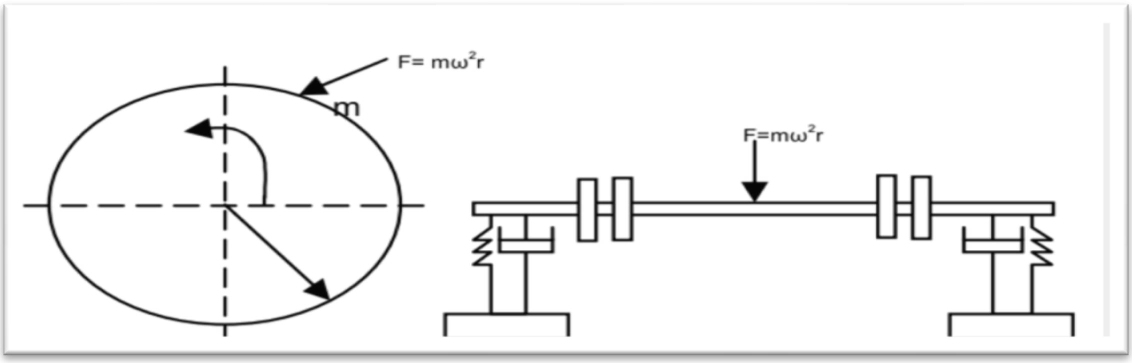

Fig. 13 Rotor Shaft Experiencing Unbalance force ($m\omega^2 r$) [34]

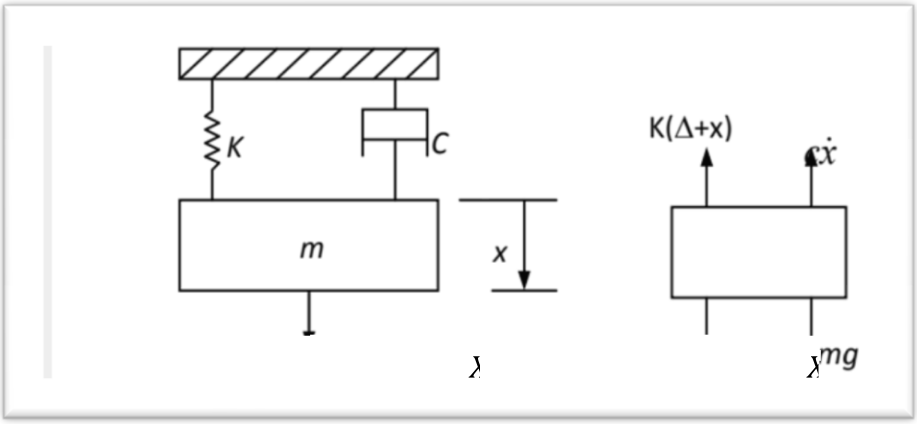

Fig. 14 Free Body Diagram of Force Vibration with Damping System [34]

Utilizing the Ordinary Differential Equation (ODE) Method, the differential equation controlling equal mass – spring and viscous – damping system is expressed as [34][35]:

$$m_{eq}\ddot{x} + C_{eq}\dot{x} + K_{eq}x = F_{eq}(t) \qquad (1)$$

Also, it can be expressed in the angular coordinate as:

Equation 3.3 is obtained by utilizing **Fig. 14**andEquation 1.

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin \omega_t \qquad (3)$$

Dividing Equation 3.3 with "m"

From Equation 1 [38].

The natural frequency is expressed as:

$$\omega_n = \sqrt{\frac{k}{m}}(5)$$

The damping ratio is expressed as:

$$\xi = \frac{c}{2\sqrt{km}}(6)$$

The standard form of the differential equation for single Degree of Freedom, substituting Equations 5 and 6 into Equation 1 is expressed as:

Equation 7 is supplementary to two initial conditions of:

$$x(o) = x_0 \ and \ \dot{x}(0) \qquad (8)$$

Since Equation 8 is a linear and ordinary homogenous differential equation with constant coefficients likes the damping ratio and the natural frequency and is expressed as:

$$x(t) = Ae^{\alpha t}(9)$$

Putting Equations 3.7 into Equation 9, Equation 10 can be expressed as:

$$(\alpha^2 + 2\xi\omega_n\alpha + \omega_n^2)Ae^{\alpha t} = 0 \qquad (10)$$

Solving the quadratic equation

α^2

Equation 12 is obtained using the Almighty Formula for quadratic equation 11;

$$\alpha = \frac{-2\xi\omega_n \pm \sqrt{(2\xi\omega_n)^2 - 4\omega_n^2}}{2} \qquad (12)$$

$$\alpha = \omega_n(-\xi \pm \sqrt{\xi^2 - 1} \qquad (13)$$

Equation 313 depends on the value of α, the root of the characteristic equation (defining i$= \sqrt{-1}$) in the case of when $\boldsymbol{\xi}$ = 1

$$x(t) = B_1 e^{i\omega_n t} + B_2 e^{-i\omega_n t} \qquad (14)$$

B_1 and B_2 are constants of integration.

The Euler's identity states that [37]:

$$e^{i\theta} = \cos\theta + i\sin\theta \qquad (15)$$

Applying Euler's identity to Equation 3.14, that is substituting Equation 15 into Equation 14.

$$x(t) = B_1(\cos\omega_n t + i\sin\omega_n t) + B_2(\cos\omega_n t - i\sin\omega_n t) \quad (16)$$

Taking cognizance of Equation 3.16 and putting the inhomogeneous function of Equation 3.3 into consideration, the steady state solution can be expressed as:

Taking the first order differentiation of Equation 3.17, the velocity is expressed as:

$$\frac{\partial t}{\partial x} = \dot{x} = \omega B_1 \cos\omega_t - \omega B_2 \sin\omega_t \qquad (18)$$

Taking the Second order differentiation of Equation 3.18, the acceleration is expressed as:

$$\frac{\partial^2 t}{\partial x^2} = \ddot{x} = -\omega^2 B_1 \sin\omega_t - \omega^2 B_2 \cos\omega_t \quad (19)$$

Comparing Equation 3.17 to Equation 3.18, Equation 19 can be reduced to:

$$-\omega^2 (B_1 \sin\omega_t + B_2 \cos\omega_t) = -\omega^2 x(t) \quad (20)$$

Substituting Equations 17; 18 and 19 respectively into Equation 3;

$m(-\omega^2$

$$= F_0 \sin\omega_t \qquad (21)$$

This implies that;

Equation 22 can also be expressed as

$$(k-\omega^2 m)(B_1 \sin\omega_t + B_2 \cos\omega_t) + c\omega(B_1 \cos\omega_t - B_2 \sin\omega_t) = F_0 \sin\omega_t \quad (23)$$

This implies that:

$$\{(k-\omega^2 m)B_1 - c\omega B_2\} \sin\omega_t + \{c\omega B_1 + ((k-\omega^2 m)B_2\} \cos\omega_t = F_0 \sin\omega_t \ (24)$$

From Equation 24, the first Algebraic Equation can be expressed as:

$$\{(k-\omega^2 m)B_1 - c\omega B_2\} \sin\omega_t = F_0 \sin\omega_t \quad (25)$$

Dividing Throughout Equation 25 by $\sin\omega_t$, the Equation 25 will be reduced to:

$$(k-\omega^2 m)B_1 - c\omega B_2 = F_0 \qquad (26)$$

From Equation 24, the second Algebraic Equation can be expressed as:

$$\{c\omega B_1 + ((k-\omega^2 m)B_2\} \cos\omega_t = 0 \qquad (27)$$

Dividing Throughout Equation 3.27 by $\cos\omega_t$, the Equation 3.27 will be reduced to:

$$c\omega B_1 + (k-\omega^2 m)B_2 = 0 \qquad (28)$$

Dividing Equations 26 with the stiffness coefficient "k" taking cognizance of $r = \frac{\omega}{\omega_n}$; $\xi = \frac{c}{c_c} = \frac{c}{2m\omega_n}$ and $x_0 = \frac{F_0}{k}$ respectively [35]

$$\frac{(k-\omega^2 m)B_1 - c\omega B_2}{k} = \frac{F_0}{k} \quad (29)$$

$$\left(\frac{k}{k} - \frac{\omega^2 m}{k}\right)B_1 - \left(\frac{c\omega}{k}\right)B_2 = \frac{F_0}{k} \quad (30)$$

$$(1-r^2)B_1 - 2r\xi B_2 = x_0 \ (31)$$

While for Equation 28

$$\frac{c\omega B_1 + (k-\omega^2 m)B_2}{k} = \frac{0}{k} \ (32)$$

$$\left(\frac{c\omega}{k}\right)B_1 + \left(\frac{k}{k} - \frac{\omega^2 m}{k}\right)B_2 = 0 \ (33)$$

$$2r\xi B_1 + (1-r^2)B_2 = 0 \ (34)$$

Where B_1 and B_2 are constants

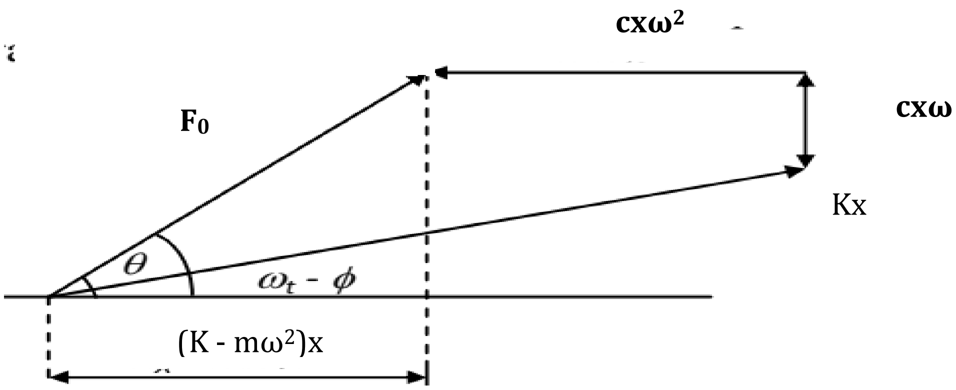

Fig. 15 Vector Representation of **Fig. 14**

Utilizing Figure 15 which is the vector resolution of the reaction forces of the free body diagram (FBD) of Figure 14 and applying Pythagoras Law for the sectional part;

$$F_0^2 = (cx\omega)^2 + (kx - mx\omega^2)^2 \quad (35)$$

$$F_0^2 = x^2\{(c\omega)^2 + (k - m\omega^2)^2\} \quad (36)$$

$$x^2 = \frac{F_0^2}{(c\omega)^2 + (k-m\omega^2)^2} \qquad (37)$$

$$x = \sqrt{\frac{F_0^2}{(c\omega)^2 + (k-m\omega^2)^2}} \qquad (38)$$

Dividing the RHS of the Equation 39 the stiffness coefficient "k"

$$x = \frac{\frac{F_0}{k}}{\sqrt{\frac{(c\omega)^2}{k} + \left(\frac{k}{k} - \frac{m\omega^2}{k}\right)^2}} \quad (39)$$

$$x = \frac{\frac{F_0}{k}}{\sqrt{\frac{(c\omega)^2}{k} + \left(1 - \frac{m\omega^2}{k}\right)^2}} \quad (40)$$

From Equation 41, the Phase Angle can be expressed as:

$$\tan\theta = \frac{c\omega}{k} - m\omega^2 = \frac{\frac{c\omega}{k}}{1 - \frac{m\omega^2}{k}} \quad (41)$$

The vibration displacement amplitude can be expressed as:

$$F_0 = \frac{kx}{\sqrt{\left(\frac{c\omega}{k}\right)^2 + \left(1 - \frac{m\omega^2}{k}\right)^2}} \quad (42)$$

III. RESULTS AND DISCUSSION

RESULTS

The Table 1 contains values of the gas turbine vibration parameters like the turbine speed, vibration displacement amplitude of the bearings A, B, C and D which was taken at each reference vibration displacement amplitude which serves as a minimum vibration displacement the gas turbine rotor shaft can embark upon; the power generated in Megawatts (MW). All these vibration parameters are found in the Turbine Logsheet (See Appendix A) and Equation 41 is the vibration displacement amplitude which is being provided in the Turbine Logsheet as well. Painstakingly, the procedural process for the vibration data collection continues for twenty-five weeks in which each week a minimum of five days hourly readings were taken and the average taken and recorded for that week. Notwithstanding, the presumption of the angular velocity (w) for the vibration operation commenced via assuming that the one unit of the amplitude for the free mass. However, this presupposition of the amplitude together with the force of the remaining masses that are at were given and the frequency of the turbine in Hertz is gotten via the conversion of the turbine speed in revulsion per minutes (RPM) to Hertz (Hz) (1rpm = 0.02Hz). The obtainable results were validated when put in comparison with [11][36] and they display closed relationship. The simulation of the results was performed using Excel Software Program in order to get the characteristics vibration depiction of turbine speed on the frequency, reference vibration displacement amplitude, power generated and the vibration displacement amplitudes of the four bearings A, B, C, and D. Also, a characteristics depiction of the power generated on the frequency and reference vibration displacement amplitude were also simulated as well. These vibration characteristics depictions are displayed in Figures 16 – 21.

Table 4.1: Turbine Vibration Parameters

S/N	Turbine Speed (RPM)	Frequency (Hz)	Reference Vibration Displacement Amplitude mm/s	Power Generated (MW)	Bearing A mm/s	Bearing B mm/s	Bearing C mm/s	Bearing D mm/s
1.	7265	145.30	2.43	14	2.32	2.20	1.01	0.21
2.	7328	146.56	2.94	19	2.90	2.10	0.93	0.24
3.	7329	146.58	2.90	11	2.84	2.12	0.94	0.28
4.	7332	146.64	3.31	19	3.70	3.50	1.23	0.48
5.	7342	146.84	2.54	19	2.84	2.10	0.81	0.28
6.	7343	146.86	2.94	19	2.90	2.40	1.04	0.28
7.	7344	146.88	3.40	19	3.40	3.10	0.80	0.40
8.	7349	146.98	2.94	19	2.91	2.24	1.10	0.21
9.	7351	147.02	2.94	19	2.74	2.49	0.80	0.28
10.	7353	147.06	2.64	19	2.60	2.64	1.10	0.34
11.	7354	147.08	2.84	11	2.70	2.30	1.10	0.34
12.	7354	147.08	2.94	11	2.84	2.67	0.79	0.33
13.	7356	147.12	3.10	19	2.90	2.79	0.80	0.33
14.	7357	147.14	3.00	19	2.94	2.68	0.87	0.34
15.	7360	147.20	2.74	18	2.79	2.64	0.88	0.34
16.	7361	147.22	3.58	18	2.64	2.14	0.74	0.34
17.	7364	147.28	2.89	18	3.20	2.64	0.54	0.30
18.	7365	147.30	3.24	18	2.79	2.99	1.10	0.33
19.	7366	147.32	2.99	11	3.10	2.98	0.80	0.39
20.	7368	147.36	3.20	11	2.92	2.67	0.82	0.40
21.	7370	147.40	3.00	19	3.10	2.00	0.82	0.44
22.	7374	147.48	3.30	19	3.10	2.64	0.83	0.39
23.	7382	147.64	3.30	19	3.14	2.70	0.90	0.38
24.	7391	147.82	3.40	19	3.34	3.10	0.84	0.74
25.	7394	147.88	3.30	19	3.30	2.69	0.74	0.40

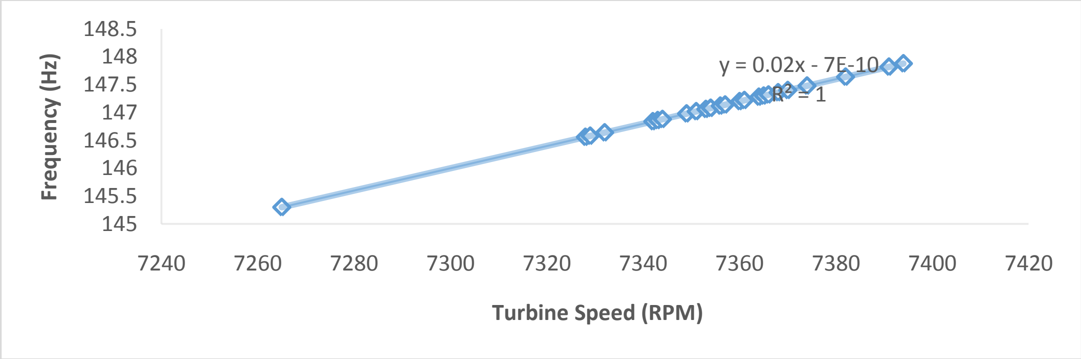

Fig. 16Characteristic Vibration Depiction of the Turbine Speed on the Turbine Frequency

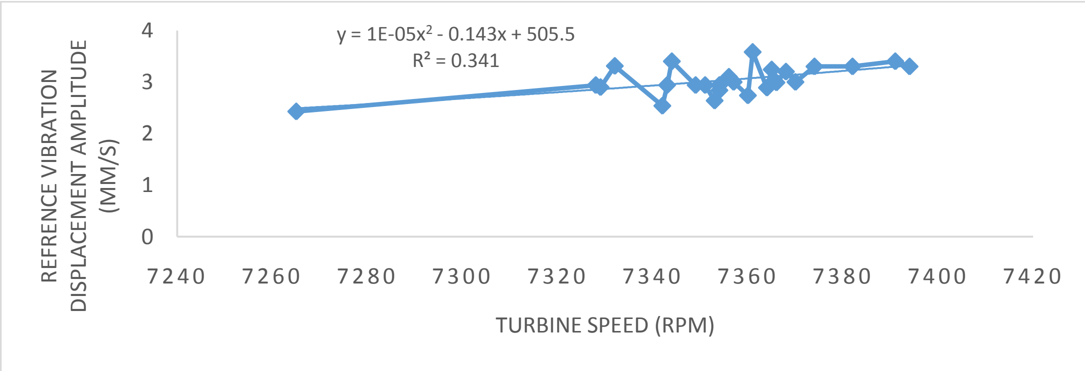

Fig. 17 Characteristic Vibration Depiction of the Turbine Speed on the Reference Vibration Displacement Amplitude

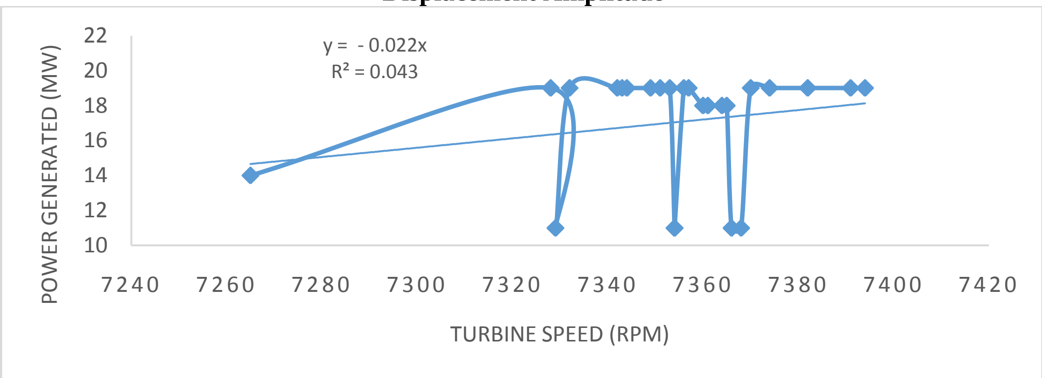

Fig. 18 Characteristic Vibration Depiction of the Turbine Speed on the Power Generated

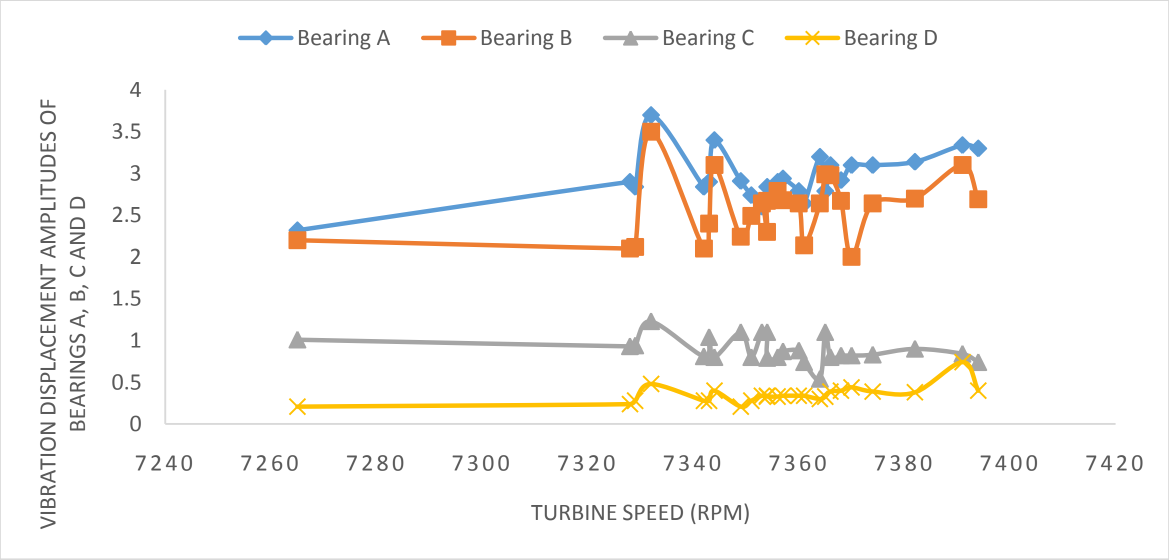

Fig. 19 Characteristic Vibration Depiction of the Turbine Speed on the Vibration Displacement Amplitudes of Bearing A, B, C and D.

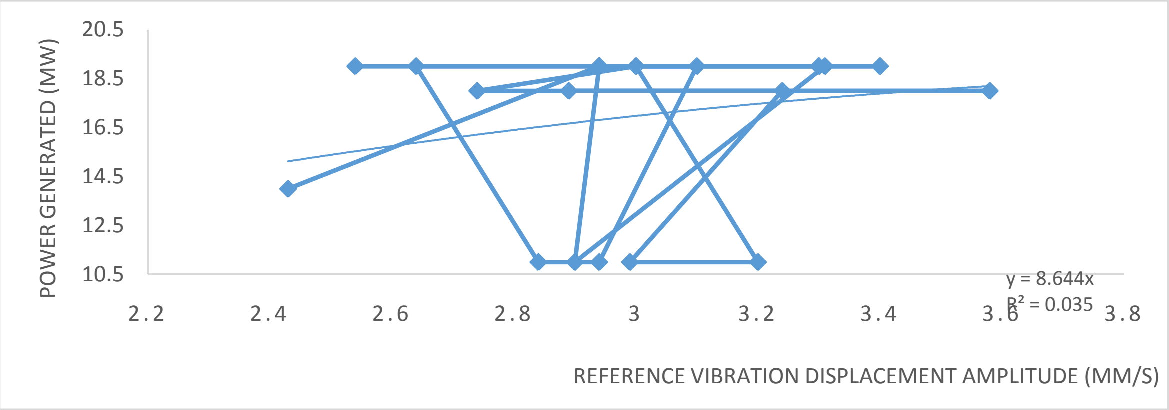

Fig. 20 Characteristic Vibration Depiction of the Power Generated on the Reference Vibration Displacement Amplitude

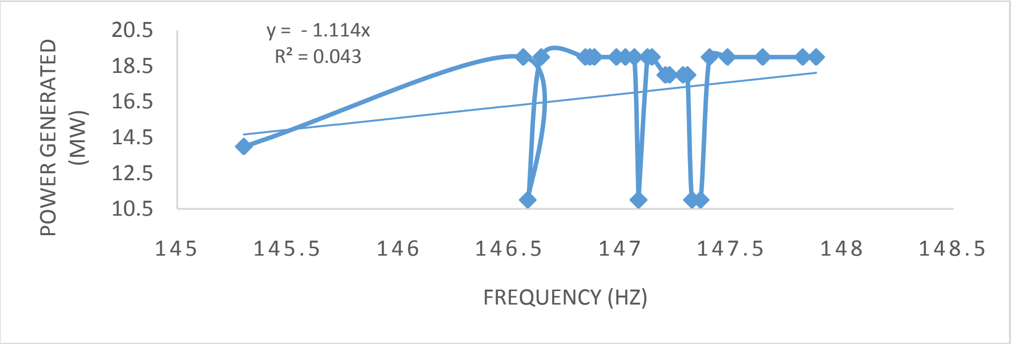

Fig. 21 Characteristic Vibration Depiction of the Power Generated on the Turbine Frequency

DISCUSSION

Fig. 16 is the characteristic vibration depiction of the turbine speed on the turbine frequency and it show that as the speed of the turbine increases the frequency also increases as well. The characteristic vibration depiction of speed versus frequency revealed the comprable relatonship of both vibration parameters. However, **Fig. 17** is the characteristic vibration depiction of the turbine speed on the reference vibration displacement amplitude and it shows a fluctuating nature of the graph as the speed increases indicting effect of vibration on the plant.Also, the reference vibration amplitude is the minimum vibration displacement amplitude expected in the four bearings A, B, C and D respectively. Notwinstanding, the **Fig. 18** is the characteristic vibration depiction of the turbine speed on the power generated and it displays that as the turbine speed increases, there is rise and fall of power supply due to the cause of vibration. It also reveals that from 7332 rpm to 7353 rpm; there was a steady power supply of 19MW which translates to be 76%. Precisely, **Fig. 19**is the characteristic vibration depiction of the turbine speed on the vibration displacement amplitudes of bearing A, B, C and D,and unlike the reference vibration displacement amplitude which serves as a guide, the vibration of the Bearing A, B, C and D affects the plant. The graph also reveals that the vibration amplitude increases from Bearing A to Bearing D. It happens like that due to the reason that Bearing D is closer to the gas turbine generator and the high vibration displacement amplitude must be as a result of misalignment. While **Fig. 20** is characteristic vibration depiction of the power generated on the reference vibration displacement amplitude. It revealed the uncertainty nature of power generation and vibration motion and the graphical representation depicts that vibration really affect power generation. Finaly, **Fig. 21**is characteristic vibration depiction of the power generated on the turbine frequency and it look similar to **Fig. 18** which is characteristic vibration depiction of the turbine speed on the power generated thereby showing the comparable nature of gas turbine speed and the gas turbine frequency. The graph also show that both the turbine speed and frequency affect the gas turbine performance.

IV. CONCLUSION

The revelations from the vibration analysis and condition monitoring of the marine gas turbine for efficient performance of Trans- Amadi Gas Turbine utilizing available vibration parametric data reveals that vibration influence the performance of the gas turbine (See Figures 16 – 21).Precisely, the number of deficiency apportioned to the gas turbine is a function of the vibration displacement amplitude on the rotor-shaft and the four bearings A, B, C and D respectively. Observably, the bearings at the turbine bears very high force as a result of the power generation thereby bring about very high vibration displacement amplitude. Consequently, the vibration displacement amplitude on the bearing attached to the rotor-shaft is dependent entirely on the distance of the bearing from the power side of the rotor shaft.The vibration appraisal also displays that using vibration for diagnosis in machinery like gas turbine is the best effective way for detecting together with identifying deficiency in bearing thereby making it very vital in order to debar ruinous breakdown of the gas turbine plant. It is recommended that appraisal and monitoring of gas turbine should be steadily conducted in order to avoid unscheduled shutdown.

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