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# Single Plane Mass Balancing of Generator Rotor Contemplating Vibration Amplitude Response

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**ABSTRACT:** Vibration of the rotating equipment's is a critical issue when shafts are being rotated at high speeds has a huge potential in causing a bearing failure and it can even turn into a massive catastrophic failure. All of this can be prevented by monitoring and controlling the vibration of the rotating equipment. Vibration has different sources, out of all the prominent source is the mass and force unbalance of the rotor. The top gas pressure recovery turbine is auxiliary unit in the iron making process by blast furnace the residual pressure at the furnace top is utilized by this reaction turbine, which is coupled to a generator to produce a power of 12 MWH. After overhauling of the whole unit which is planned periodically (once in 4 years) the Generator DE and NDE bearing was found suffering from the high vibration amplitudes. The above mentioned problem was taken as the live example in order to design a suitable methodology for reducing the vibration level at bearings using the dynamic mass balancing.

**KEYWORDS:** Mass balancing, High speed rotating equipment, Vibration response, Catastrophic failure.

### I. INTRODUCTION

High speed of about 3000 rpm the whole unit of 15 tons mass rotating at 3000 rpm rotary equipment's have huge potential in creating hazardous catastrophic failures which require frequent and continuous monitoring of various parameters that govern the state of the rotating equipment. Vibration is one of the prominent factor which affects the equipment performance by increasing the load on its bearings. The main causes for the vibration are misalignment & mass imbalance.

Misalignment is the difference in the rectilinear position of two shafts in order to transmit the power from one shaft to the other both shafts have to be perfectly aligned. It could be an angular one or an offset. Apart from misalignment there are many number of causes like uneven radial run-outs, insufficient bearing clearances and interferences, shaft blade clearances, inappropriate axial float setting, inappropriate bearing mounting, uneven force on the blades. Out of the entire mass imbalance plays a major role in affecting the vibration of the generator rotor.

Mass Imbalance for the huge mass rotating on a particular axis requires the total mass of the rotor to be uniformly distributed around the axis which causes zero resultant force when the shaft rotates. This can be achieved by evenly distributing the mass around its centre of gravity which is very much essential.

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## II. PROBLEM DEFINITION

Gas turbine are always prone to the dust content in the gas which they operate, deposition rate of the dust on both runner and guide blades increases with the running hours of the turbine which is need to be cleaned periodically by taking up overhauling jobs in order to ensure safe running of the gas turbines. One of such overhauling job was planned for the top gas recovery turbine which uses the residual top pressure of the blast furnace in order to generate power. After all overhauling jobs the various parameters that govern the smooth working of both turbine and generator rotor was taken care (i.e. Bearing clearances& Interferences, Coupling run-outs, Alignment of the rotors etc.) and set to the optimal required levels. During the trail run of the turbine it was found that the vibration levels at both DE (Drive end) and NDE (Non Drive End) was found on the higher side i.e.3.86 mm/sec and 5.45 mm/sec respectively, which was not acceptable for safe running of generator. The vibration limits for generator rotor is between 2 to 4 mm/sec for safe operation as per ISO-10816-3. The only suspect for the present increase in the vibration level of generator is the mass imbalance of the generator rotor. This is taken as the problem statement to develop a methodology to reduce the vibration level through single plane mass balancing of the rotor with in the field by contemplating vibration amplitude response.

Table 1. Vibration amplitude response during trail run of the turbine

Time			10:27 AM	10:33 AM	10:44 AM	10:53 AM	11:08 AM
SPEED (rpm)			715	994	2501	2995	3000
Generator	DE Bearing	Vibration (velocity mm/sec)	0.58	0.61	1.88	3.79	3.86
		Temperature (°c)	37.6	38.6	43.4	48.2	49.8
	NDE Bearing	Vibration (velocity mm/sec)	0.47	0.32	2.31	5.12	5.45
		Temperature (°c)	41.2	43.9	50.8	56.6	58.3

## III. BASIC THEORY AND DEFINITIONS

Unbalance in a rotor is because of the resultant unbalanced force which generates due to the mass imbalance. The imbalance in the rotor causes the variation in the centrifugal forces acting in when the shaft rotates thus creating imbalance which generates increased load on the bearings which is more uneven due to fatigue nature of it the bearing wears out soon.

Balancing is the process through which the unbalanced force is nullified by placing a residual mass at a suitable position, which generates a resultant force which is equal and opposite to the unbalanced force thereby nullifying it. There are three types of balancing methods [1]. They are

- [1] Static balancing
- [2] Couple/Moment balancing
- [3] Dynamic balancing

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## 3.1 Static Balancing

Static balancing is defined as the balance the rotor exhibits at its rest it is a characteristic of eccentricity of rotor centre to the centre of gravity of the rotor shown in figure 1. the difference in the eccentricity causes the imbalance in the shaft and affects the primary forces acting on it, so the fundamental principle of static balancing is to resolve the primary forces in one single plane and add the compensating mass in the same plane .there are many examples of various equipment's which have their imbalance in one plane such as flywheel, grindstone, car wheels, etc., can be taken up for static balancing. Usually if the diameter of the rotor is 7 to 10 times greater when compared to its width such rotors are considered for static balancing problems.

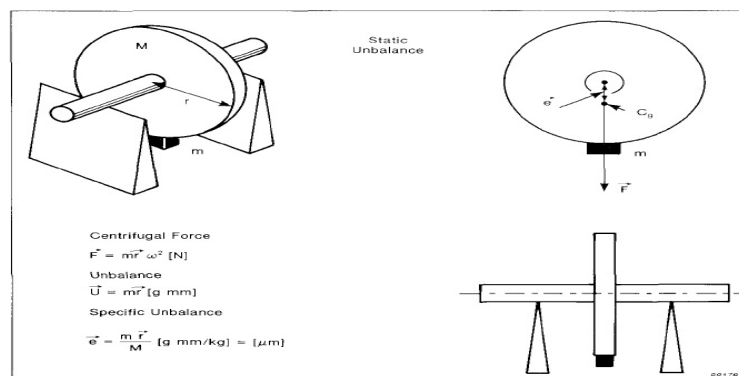


Fig.1 Static Balancing

## 3.2 Couple/Moment Unbalance

Apart from static balancing wherein unbalance is in single plane there are many other engineering problems where the unbalance is observed in two different planes. Usually if a shaft diameter is less than 7 to 10 times of it when compared to its width of the rotor would usually result in imbalance in more than one plane. If the rotor is statically balanced the eccentricity of the rotor with the centre of gravity would be zero. But if the same statically balanced rotor is rotated at certain speed due to which moment or couple unbalance takes place, which in turn causes unbalanced centrifugal forces resulting in shifting of centre of gravity, as shown in the Figure 2.

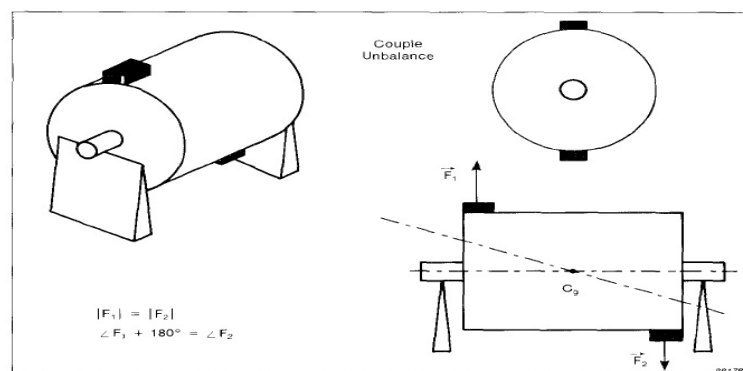


Fig.2 Couple (Moment) Unbalance

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## 3.3 Dynamic Unbalance

Dynamic balance occur when there is combination of both static and moment unbalance. Generally most of the rotors exhibit dynamic unbalance, in order to balance the rotor completely the rotor must be balanced dynamically [2]. Dynamic balance can be achieved by measuring the value of vibration while the rotor is running based on it we will calculate the residual mass and the phase angle which is mounted on the mass in two planes of the rotor (i.e. both DE and NDE) the rotor is balanced dynamically as shown in figure 3. The rotors can be classified into two types, Rigid and Flexible. Rotor is considered to be rigid if the service speed is more than 50 percent of it critical speed. Balancing of rigid rotors is easy when compared to the flexible rotor where the service speed is less than 50 percent of its critical speed [3].

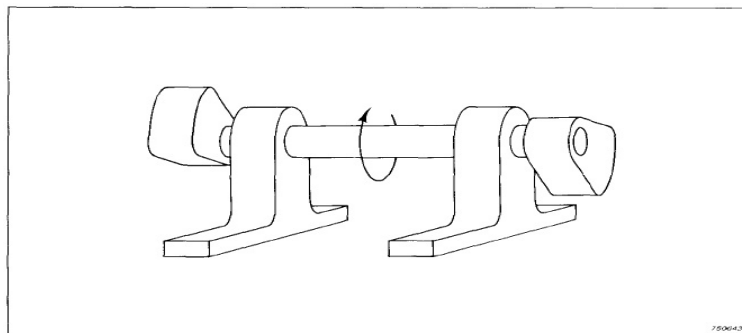


Fig.3 Dynamic Unbalance

## IV. PRINCIPLE OF BALANCING

The principle of balancing the rotor dynamically is that “correction mass is calculated by means of the suitable methodology along with the phase angle at which it has to be mounted by using a trial mass”. The suitable trial mass is selected initially according to the weight of rotor and is placed on the rotor balancing ring at a known desired position as shown in figure 4. Its effect of the present unbalance is accounted by means of measuring vibration amplitude at the drive end and Non-Drive end bearings of the rotor, the trail mass is moved to cover all phase angles of the rotor and vibration amplitude reading of all the cases is recorded, based on the reading obtained exact correction mass and phase angle is calculated [4].

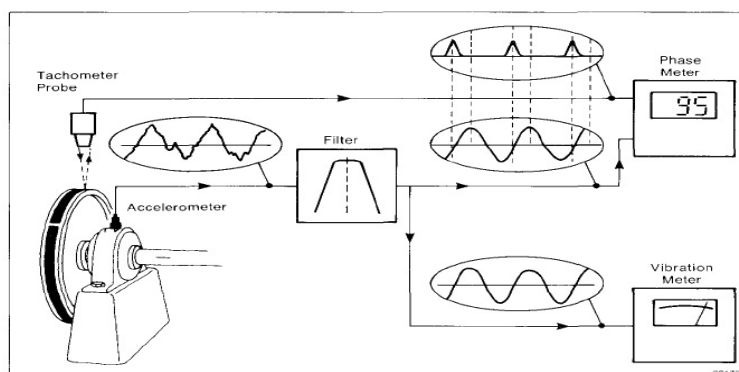


Figure 4 Balancing Set Up

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Different types of instruments are necessary for the mass balancing of the rotor which are mentioned below

- Tachometer – it is an instrument which is used to measure speed at which the shaft rotates, generally laser tachometers are used.
- Accelerometer – it is an instrument which is used measure acceleration of the vibration amplitude, it is mounted on bearing with a vibration probes.
- Vibrometer – these are instruments through which the displacement and velocity of vibration amplitude can be measured from the bearing which are mounted using vibration probes.
- FFT analyser – it is an instrument which is used to study the frequency response of the vibration amplitude through Fast Fourier Transfer which converts the vibration amplitude response in time domain to the frequency domain resulting in signature of vibration amplitude response.
- Phase meter– it is an instrument by which the phase at which peak of the vibration amplitude lie is determined. It works on the principle of frequency synchronisation of tachometer and vibrometer probe.
- Filters– these are used within the circuit of FFT and accelerometer in order to avoid unwanted noise to enter into the circuit. Which ensures the easy recognition of the vibration amplitude response and frequency response.

## V. GENERAL BALANCING PROCEDURE

### 5.1 Performing a Frequency Analysis

Before performing the mass balancing operation it is necessary to carry out the frequency analysis or signature analysis on the present vibration amplitude response from the bearings, the vibration amplitude response will be in the time domain in order to study the signature analysis of it, it has to be converted into frequency domain. This process is carried out by FFT analyser which uses Fast Fourier Transform for the same cause. Performing the frequency analysis is mandatory before mass balancing because.

- [1] In order to ensure the present increase in the vibration amplitude response is because of imbalance of the mass of the rotor By analysing the frequency response, if the peak in the vibration amplitude level corresponds to rotation frequency of the rotor, i.e. 1x peak is observed in the frequency response of the FFT which is dominating all other peaks in the signature. By this we ensure that the present increase in the vibration amplitude response is due to mass imbalance [5].
- [2] For selecting the best measure of vibration amplitude for the purpose of analysis. The dynamic mass balancing of the rotor requires the measure of the vibration amplitude response for the purpose of analysis; the vibration amplitude response can be measured in three domains 1) displacement 2) velocity and 3) acceleration. One of the above parameters has to be selected for the balancing purpose which yield lowest dynamic range and ensure the signal to noise ratio is minimal. This can be done by selecting the parameter which a flattest response in the graph has generated by plotting the vibration amplitude to that of frequency, as shown in the figure below.

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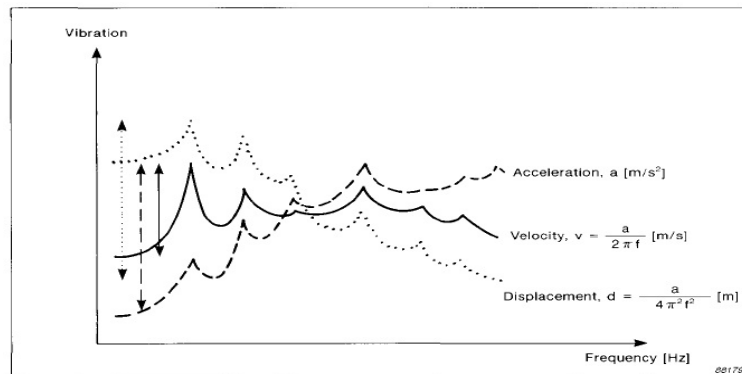


Figure 5 Selection of Suitable Parameter for Vibration Amplitude Measurement

Based on experience the velocity remains as flattest curve in most of the cases so it is often selected for further analysis the acceleration response can be used where low frequency noise can cause the problem as it emphasises the higher frequency components. The displacement can be used where the reduction in the high frequency noise is essential as it emphasises lower frequency components.

For every rotor there would be maximum allowable specific residual unbalance which depends on the weight of the rotor and its distribution about its centre of gravity. The calculations become complex when the mass of the rotor is unevenly distributed

## 5.2 Selection of Trial Masses

The most important prerequisite for the mass balancing of the rotor is selection of trial mass. This is used during the balancing of the rotor to alter the mass distribution thereby changing the vibration level at both drive end and non-drive end bearing thereby establishing a relationship between the induced mass imbalances to the vibration amplitude response at the bearing end [6].

The maximum residual mass  $M_{MR}$  in grams is determined by the following relation

$$M_{MR} = \frac{SU \times M_R}{R_C} \dots\dots\dots (1)$$

Where,

S.U = specific unbalance required (in %) (Taking it as 2%)

$M_R$  = rotor mass (kg). (14.85 tons)

$R_C$  = correction radius (mm) (Correction ring diameter - 1200mm)

$M_{MR} = (2) (14850) / (600)$

$M_{MR} = 49.5$  grams ~ 50 grams

A suitable trial mass is 5 to 10 times the maximum residual mass. The Trial mass is approximated to be 7 times the Maximum residual mass, so the Trial mass of 350 grams is selected.

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## VI. SINGLE – PLANE MASS BALANCING OF A GENERATING ROTOR (STATIC - BALANCING)

After carrying out the frequency analysis and selecting the best parameter for the further calculations (velocity) and calculating the value of suitable trial mass. The procedure for single plane mass balancing of a generating rotor is as follows:

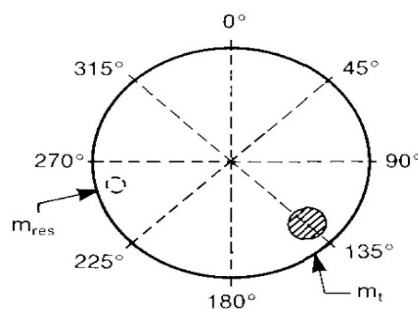


Figure 6 Placement of Trail Mass

- Mount the required instruments onto the rotor for example tachometer and accelerometer using suitable probes; ensure the calibration and proper connection of probes to the instrument.
- Start the turbine and increase the speed to the service speed ensure from the tachometer that the speed of the generator rotor is 3000 rpm.
- At this speed obtain the value of velocity of vibration and phase angle.
- Divide the rotor into equal intervals and mark the relevant points, example divides the rotor and mark at every 45° step.
- Mount the calculated trial mass at 0° position, rotate the rotor at its service speed and note down the vibration amplitude value as shown in figure 6.
- Now move the trail mass to 45° and repeat the procedure.
- Note down vibration amplitudes for every position of trial mass.
- Calculate the mean vibration amplitude value and highest amplitude peak values shown in table 2.
- Plot a graph with trial mass position on x axis and vibration amplitude on Y axis; mark the mean vibration line and highest peak line is shown figure 7.
- The linear distance from the mean vibration value to the highest peak of vibration amplitude gives the value of  $V_{res}$ , which represents the magnitude of unbalance
- The linear distance from the Mean vibration value to the zero line gives the value of  $V_t$ , which represents the magnitude of vibration developed due to trial mass

Table 2. Calculation of Mean Vibration Value & Highest Peak Value

Position	Vibration	Mean Vibration value	Highest peak value
0	10.5	10.075	11.85
45	8.8	10.075	11.85
90	8.3	10.075	11.85
135	9.1	10.075	11.85
180	10.5	10.075	11.85
225	11.6	10.075	11.85
270	11.85	10.075	11.85
315	10.5	10.075	11.85

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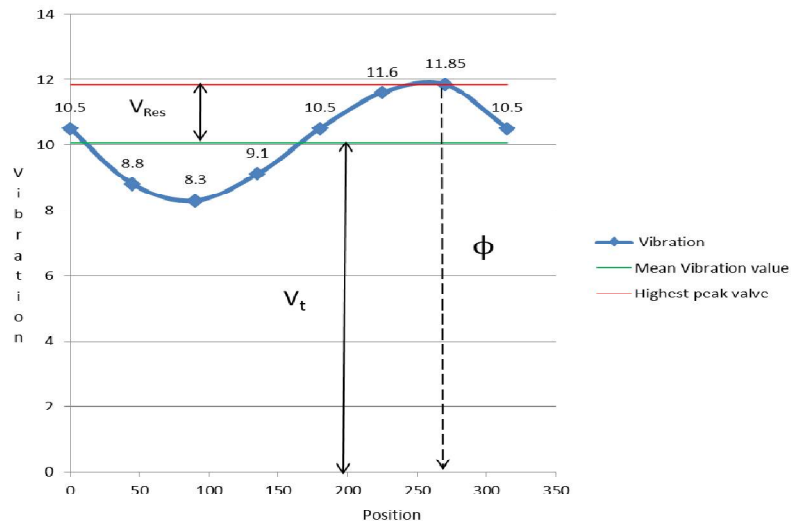


Figure 7 Graphical Method of Single Plane (NDE) Mass Balancing of Generator Rotor

Finally the magnitude of mass required for rotor to clear the present unbalance is calculated as

$$M_{res} = \frac{|V_{res}|}{|V_t|} \times M_T \quad \dots\dots\dots (2)$$

From the Figure 7

The value of  $V_{res} = (11.95 - 10.075) = 1.875$

The value of  $V_t = (10.075 - 0) = 10.075$

$M_{res} = (1.875)(350)/(10.075)$

$M_{res} = 65.13647$  grams,  $M_{res} \sim 65$  grams

The phase angle ( $\phi$ ) at which it has to be mounted on the correction ring is obtained by the Graph by drawing a linear vertical line from maximum vibration amplitude to the X axis to note the value of  $\phi$ . From graph the value of  $\phi = 264^\circ$ .

The Top gas recovery turbine was stopped and the generator covers were removed in order to add the calculated mass to generator rotor in the mass balancing ring [7] [8].

The mass of  $M_{res} = 65$  grams was added at a phase angle of  $\phi = 264^\circ$ .



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## VII. SINGLE PLANE BALANCING OF GENERATOR ROTOR

The machinery was started after fixing back the end covers and the vibration level at the different speeds were noted in table 3.

Table 3. Vibration Response after Single Plane Mass Balancing

Time			11:21 AM	11:39 AM	11:48 AM
SPEED (rpm)			1000	2400	3000
Generator	DE Bearing	Vibration (velocity mm/sec)	1.23	3.2	4.85
		Temperature (°c)	39.6	44.2	47.9
	NDE Bearing	Vibration (velocity mm/sec)	0.46	1.5	1.98
		Temperature (°c)	45.4	51.3	55.3

## VIII.CONCLUSION

As it was observed that due to single plane mass balancing of generator rotor at NDE bearing, the vibration level was successfully brought down to 1.98 (mm/sec). But it was observed that due to CROSS EFFECT of balancing in the NDE the DE vibration level is increased up to 4.85(mm/sec) at service speed (i.e. 3000 rpm). Therefore the single plane mass balancing yields good results in the plane selected for the balancing(i.e. NDE) but due to Cross effect it will increase the unbalance in the other end (i.e. DE). For the effective balancing of the rotor systems two plane mass balancing is recommended using the same methodology in order to minimize the CROSS EFFECT.

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