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Identification of Type of Unbalance Present in Rigid Rotor

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Abstract **- Unbalance is one of the most common causes of machinery vibrations, present in rotating machines. This paper gives a brief description about the common types of unbalance. The importance of identifying type of rotor unbalance (static, couple, quasi-static, or dynamic) is presented. This paper describes the method to detect type of unbalance present in the rotor. The vibration data can be measured with the help of FFT analyser. For experimental verification of new method, a test rig is designed and developed. Fundamental terms that are used throughout the balancing procedure are explained. These terms are the back bone to why and how balancing is performed.**

Keywords: Unbalance, FFT Analyser, Static, Couple, Quasi-static, Dynamic.

I. INTRODUCTION

Rotor unbalance is the most common malfunction in rotating machinery. Due to commonly occurring uneven radial distribution of mass along the rotor, unbalance is the most common in rotating machinery. At a constant rotational speed, the rotor responds to the unbalance force with synchronous (1x) lateral vibrations in a form of an elliptical orbit. Unbalance related vibrations account for approximately eighty percent of all vibration problems in rotating equipment. Unbalance can be the cause of not only excessive vibration, but also short bearing life and noise.

There are many procedures used to balance rotating systems. Randall L. Fox [1] provided essential information needed to solve the majority of balancing problems. This information provided enable to solve the majority of balancing problems both in the field and in the balancing shop.

Cunningham [2] has given a brief comparison of shop vs. in-field methods of dynamic balancing. Also need of balancing, different types of unbalance and balancing tolerance standards explained. This work gave idea about how to choose which balancing method, in-shop or in-field.

Bruel and Kajer [3] provided basic theory and principles of dynamic balancing of rigid rotors. The criterion for selecting best measurement parameter for balancing is explained.

Feese and Grazier [4] has given six case histories, which illustrate actual problems that are encountered while trying to field balance different types of rotating machinery.

Neville F. Rieger [5] presented an introduction to the principles of balancing, certain basic balancing procedures.

II. CHARACTERISTICS OF UNBALANCE

Unbalance is defined as simply the unequal distribution of the weight of a rotor about its rotating centerline. According to International Standard Organization (ISO) unbalance is – “That condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces” [1].

The ability to distinguish between unbalance and other sources of vibration is an essential first step in any analysis. The basic characteristics of unbalance are straightforward and easy to understand.

a. The dominant vibration will occur at the rotating speed of the rotor (1*RPM).
b. The Vibration will be highest in the radial direction (Horizontal and vertical); axial vibration will be low except for cantilevered rotor.
c. The amplitude and phase angle of the vibration will be steady and repeatable.
d. The phase analysis of the vibration at the rotational speed (1*RPM) will show a 90° shift from horizontal to vertical at the inboard or the outboard bearings of the machine.

III. CAUSES OF UNBALANCE

There are many reasons that unbalance may be present in a rotor. The most common causes are as follow.

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a) Eccentric component.
b) Incorrect assembly.
c) Loose parts.
d) Deposit built-up.
e) Wear and tear of parts.
f) Blow holes in casting.
g) Thermal and mechanical distortion.
h) Addition of keys and keyways.
i) Machining tolerances and fits allowing assembly error.
j) Structural damage such as bent or broken components.
k) Change in operational conditions.

Other problems can occur at 1*RPM. While unbalance always causes vibration at 1*RPM, 1*RPM vibration is not always caused by an unbalance. Other problems are misalignment, bent shaft, cracked shaft, open rotor bars in motors, partial rubs, looseness, resonance etc.

IV. TYPE OF UNBALANCE

The location of the mass center and the principal inertia axes are determined by the distribution of mass within the part. Unbalance exists when the axis of rotation is not coincident with a principal inertia axis. The International Standards Organisation (ISO) defines four basic types of unbalance in their standards 1925 that covers balancing technology. These are static unbalance, couple unbalance, quasi-static unbalance, and dynamic unbalance. Each type of unbalance is defined by the relationship between the central principal axis and the rotating centerline of the machine. The central principal axis is the axis about which the weight of a rotor is equally distributed.

V. EXPERIMENTATION

The test rig for analysis of dynamic unbalance of rotors is as shown in figure 1. It has 3 HP motor with speed of 1440 rpm. It is possible to operate the motor with the help of VFD (Variable Frequency Drive) at above 1440 rpm. The test rig consists of four rotors mounted on shaft with equal distance apart. The twelve holes are provided on all rotors so that unbalance can be created and balance weight can be added at any location. The shaft is provided with throughout keyway of 430 mm so that rotors can be fixed at various locations in axial direction. The shaft is supported between two self-aligning ball bearings. The rotor shaft and motor shaft are connected by jaw type rigid coupling.

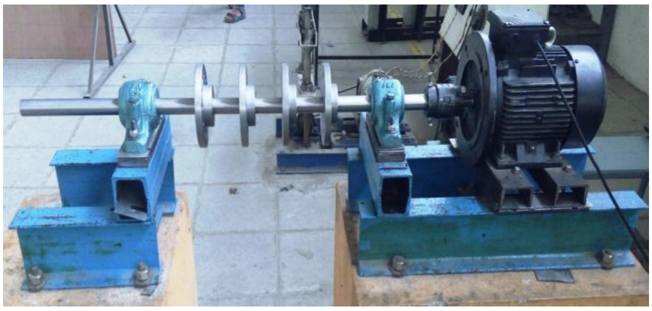

Fig.1 Experimental Test Rig

The type of unbalance can be detected by comparing vibration parameters on both the bearings. The four types of unbalance are created one by one, by attaching masses at different location on four rotors, according to definition of unbalances. For each type of unbalance created, vibration parameters are measured on both the bearings and graphs are plotted as shown below.

A. Static unbalance

A condition of static unbalance exists when the mass center does not lie on the axis of rotation. Static unbalance is also known as force unbalance. As defined, static unbalance is an ideal condition, it has the additional condition that the axis of rotation be parallel to the central principal axis. The static unbalance is created by attaching five trial masses on five rotors at same location on same straight line as shown in figure 2.

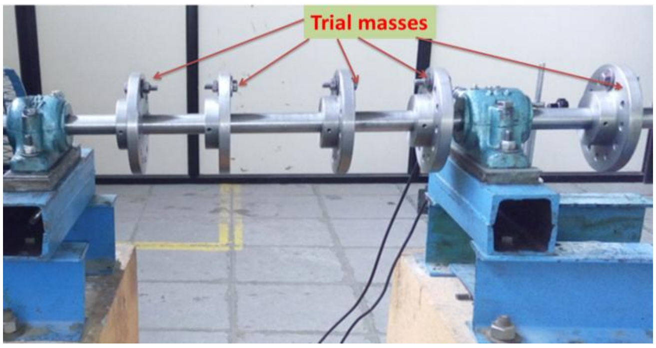

Fig. 2 Static unbalance

Table 1 Effect of static unbalance on vibration parameters at various rotor speeds.

Speed	DE Bearing		NDE Bearing	
	Velocity	Phase Angle	Velocity	Phase Angle
600	0.357	12	0.829	21
800	1.81	31	0.675	21
1000	1.26	30	2.61	38
1200	2.16	36	3.86	43
1400	3.92	44	4.9	50
1600	8.01	56	10.2	59

B. Couple Unbalance

A couple is simply two parallel equal forces acting in opposite directions but not on the same straight line. Couple unbalance is that condition of unbalance where the central principal axis intersects the rotating centerline at the rotor center of gravity. Couple unbalance condition is created by attaching trial masses at each end of the rotor but on the opposite sides of the centerline as shown in figure 3.

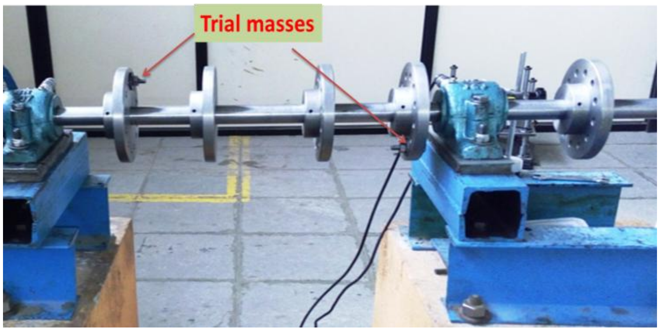

Fig. 3 Couple unbalance

Table II Effect of couple unbalance on vibration parameters at various rotor speeds.

Speed	DE Bearing		NDE Bearing	
	Velocity	Phase Angle	Velocity	Phase Angle
600	0.177	10	0.117	186
800	0.329	21	0.272	199
1000	0.496	31	0.564	207
1200	0.945	34	0.892	212
1400	1.49	40	1.63	218
1600	2.49	50	2.73	226

C. Quasi static unbalance

The quasi-static unbalance is combination of static and couple unbalance, where the static unbalance is directly in line with one of the couple moments. Quasi-static Unbalance is that condition where the central principal axis intersect the rotating centerline but not at the rotor center of gravity. The quasi-static unbalance is created by attaching trial masses at each end of the rotor but on opposite side of centerline same as couple unbalance. One more trial mass is attached on another rotor in line with one of end rotor as shown in figure 4.

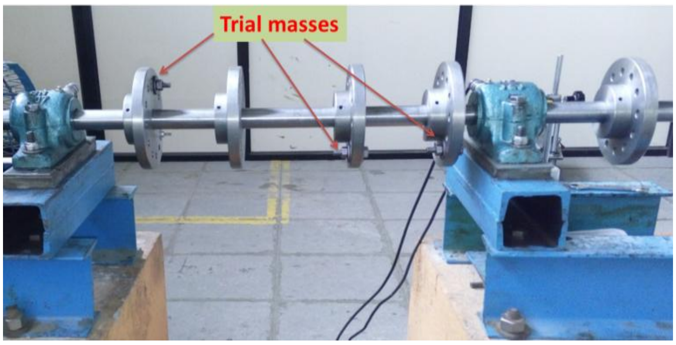

Fig.4. Quasi-static unbalance

Table III Effect of Quasi-static unbalance on vibration parameters at various rotor speeds.

Speed	DE Bearing		NDE Bearing	
	Velocity	Phase Angle	Velocity	Phase Angle
600	0.131	11	0.164	197
800	0.213	17	0.471	203
1000	0.311	27	1.08	210
1200	0.446	30	2.45	219
1400	0.739	35	4.33	225
1600	1.02	38	6.86	230

D. Dynamic unbalance

Dynamic unbalance is the most common type of unbalance. The dynamic unbalance exist whenever static and couple unbalance are present but where the static unbalance is not in direct line with either couple component. In dynamic unbalance central principal axis and rotating centerline do not coincide or touch. The dynamic unbalance is created by attaching trial weights as shown in figure 5.

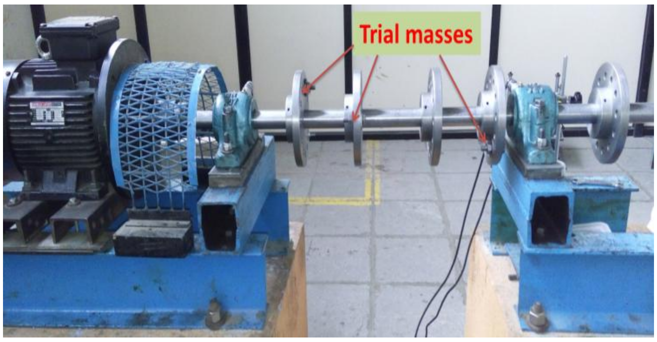

Figure 5 Dynamic unbalance

Table IV Effect of dynamic unbalance on vibration parameters at various rotor speeds.

Speed	DE Bearing		NDE Bearing	
	Velocity	Phase Angle	Velocity	Phase Angle
600	0.167	24	0.191	142
800	0.424	33	0.471	153
1000	0.696	48	0.836	150
1200	1.14	54	1.61	159
1400	1.89	63	3.46	176
1600	2.81	80	5.91	184

E. Graphs of unbalance parameters

1. **Effect of static unbalance on vibration parameters at various rotor speeds**

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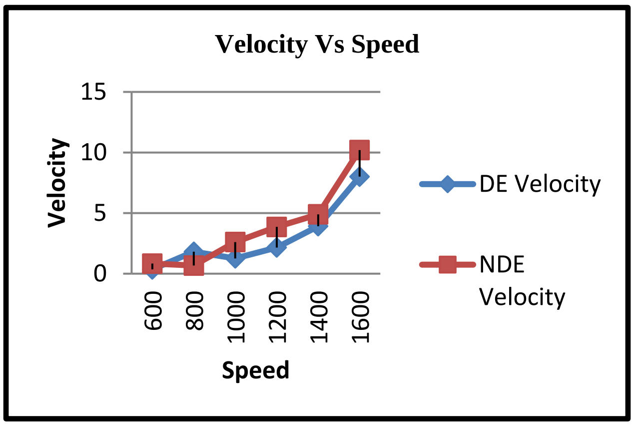
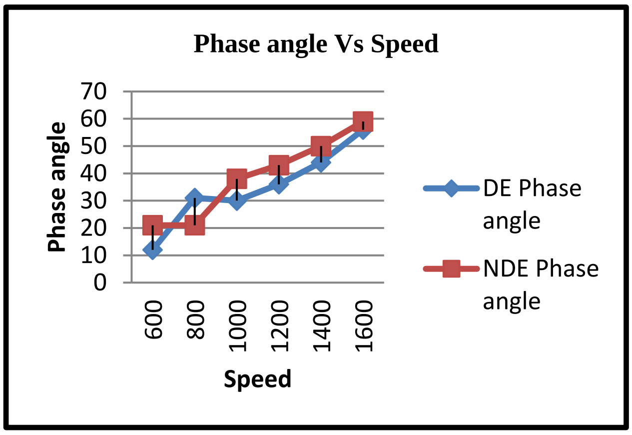
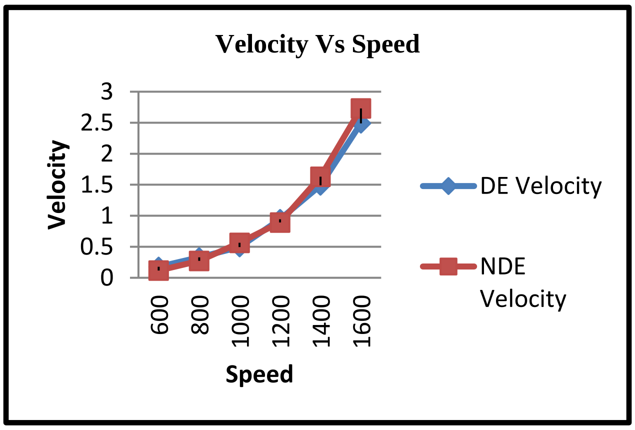


Graph 1

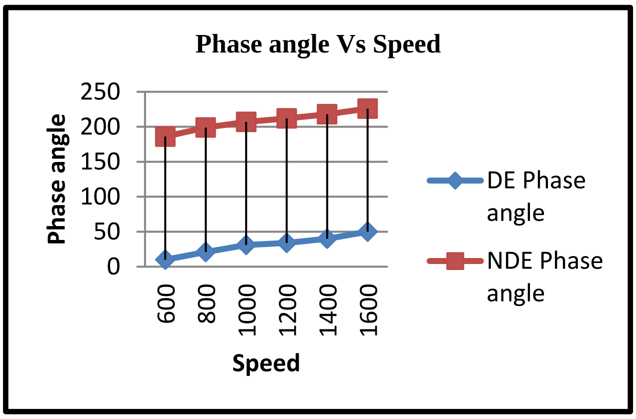


Graph 2

2. Effect of couple unbalance on vibration parameters at various rotor speeds

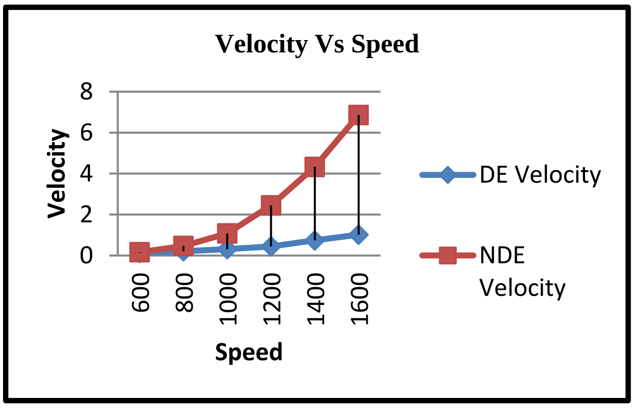


Graph 3

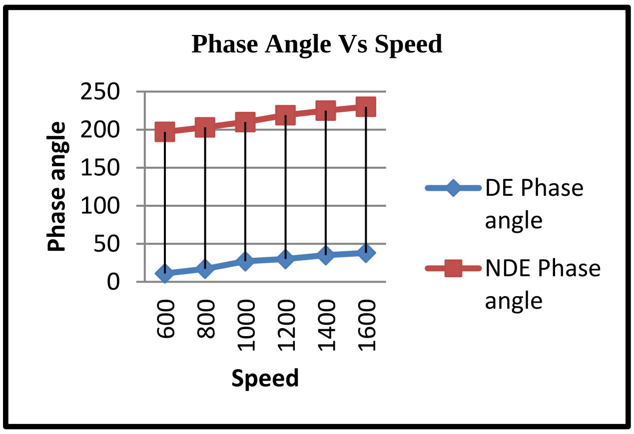


Graph 4

3. Effect of quasi-static unbalance on vibration parameters at various rotor speeds

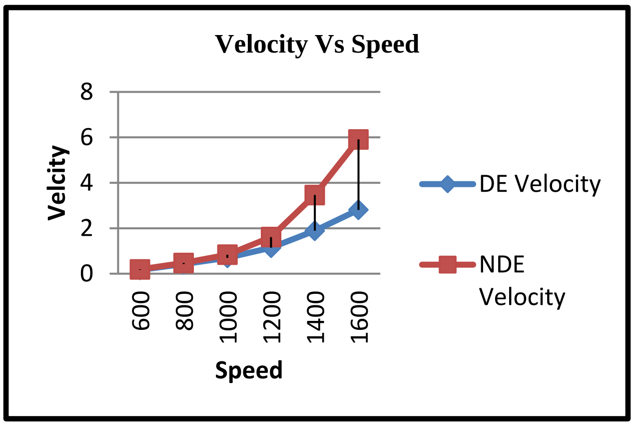


Graph 5

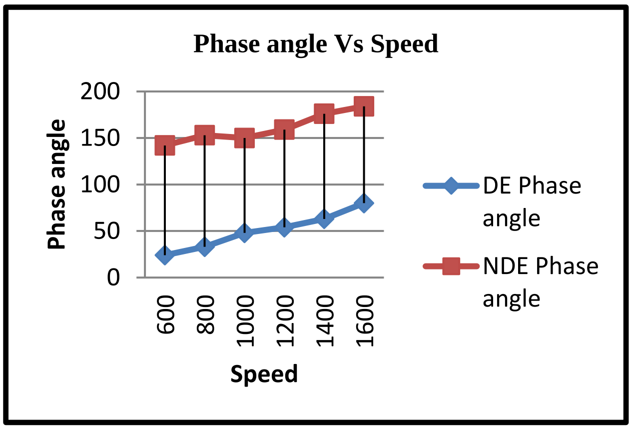


Graph 6

4. Effect of dynamic unbalance on vibration parameters at various rotor speeds



Graph 7



Graph 8

CONCLUSION

The graphs plotted for different types of unbalance gives following conclusions-

1. The rotors supported between the bearings shows approximately equal vibration amplitude and phase angle readings measured at the bearings at each end of the shaft. Thus static unbalance can be detected by observing equal vibration amplitude and phase readings at both bearings.
2. The rotors supported bearings shows equal amplitudes of vibration but the phase readings differ by 1800. Thus couple unbalance can be identified by comparing the bearings vibration amplitude and phase readings at each end of the rotor.
3. The rotors supported between bearings shows phase readings differ by approximately 1800. The amplitude of vibration is higher on non-drive end bearing. This is because of trial mass attached on plane 3 is in line with trial mass on plane 4 and near to non-drive end. Thus if the bearing shows the phase readings differ by 1800 and higher vibration amplitude on one of the bearing, then quasi-static unbalance is present in the rotor.
4. The Dynamic unbalance shows phase readings which are neither the same nor directly opposite to one other.

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