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DYNAMIC BEHAVIOR OF ROTOR WITH TWO DISCS

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ABSTRACT

Vibrations are found almost everywhere in rotating machines. Rotating machinery vibrates due to unbalances, misalignments and imperfect bearings. Study performed on a rotor to predict the unbalance in rotor for threshold instability. In this paper vibration characteristic of a rotor having one disc and two discs is investigated. The vibrations are measured at different speeds by vibration sensor. Rotating speed is used as the control parameter to observe vibrations.

Keywords- *Unbalance, Threshold instability, Critical speed. etc.*

I. INTRODUCTION

Rotor dynamics is the study of rotating machines and has a very important part to play throughout the modern industrial world. Rotating machinery is used in many Applications, e.g., Turbo machines, Power Stations, Machine tools, Automobiles, Household machines, Aerospace applications, Marine propulsion, and Medical equipment. The interaction these machines have in their surroundings is of great importance as if these machines are not operating at the correct speed ranges, vibration can occur which may ultimately cause failure. Failure of machinery in applications such as aero engines, turbo machines, space vehicles, etc. creates enormous repairing costs and more importantly may put human life in danger. This means industries put a great deal of resources into the study of rotor dynamics to calculate the safe operating ranges before the machine goes into service and also methods of detecting imminent failure. Recently, many active and semi active vibration control schemes have been proposed to compensate unbalance, misalignment, parametric uncertainties or exogenous excitations. Different devices such as electromagnetic bearings, auto balancers, squeeze film dampers, lateral force actuators, pressurized bearings, electro- and magnetoreohological dampers, piezoelectric actuators, shape-memory alloys, etc. have been used for this purpose.

Rotor Unbalance

There are two types of rotating unbalance:

1. *Static Unbalance*

The principal axis of the polar mass moment of inertia of the rotor is parallel to the centerline of the shaft. Static unbalance can be detected by placing the shaft between two horizontal rails and allowing the shaft to naturally roll to the position at which the unbalance is below the shaft axis.

2. *Dynamic Unbalance*

This is when the unbalanced masses lie in more than one plane. The static test will only detect the resultant force. The unbalance has to be detected by rotating the shaft and measuring the unbalance. The machine for carrying out this detection are called 'balancing machines' and consist of spring mounted bearings that support the shaft. By obtaining the amplitude and relative phase it is possible to calculate the unbalance and correct for it.

Threshold speed of instability:

When a rotor is operated beyond a certain critical speed, a very high & unstable vibration with a fractional frequency of the rotor speed will be developed. This rotor speed referred as the instability threshold speed. At a constant rotational speed, the rotor can experience the same unstable state when operating parameters is beyond a certain value. This limit is referred as the stability boundary of that operating parameter.

Sanxing Zhaoa et. Al. [8] established the motion equations for symmetrical single-disk flexible rotor-bearing system to obtain threshold speed and unbalance response. The rotor's stiffness and damping are considered to calculate it. Shows parameter Keqr–K(γ^2) versus rotating speed for two symmetrical single-disk flexible rotor-bearing systems, one of which considers the damping of rotor while the other ignores the damping of rotor. The graph shows that when the rotor's stiffness and damping are taken into account, the studies on system stability and response will approach to the practical situations. The threshold speed of the system is a bit larger compared with that when the

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rotor's damping is ignored. The threshold speed of the system is 6247 rpm if the damping of rotor is taken into account, or it will reduce to 5960 rpm. In this case, when the damping of rotor is considered the threshold speed of the system will increase by 4.8% compared with that when the damping is ignored. Therefore, the influence of rotor's damping on the threshold speed is not obvious. The damping of rotor plays a second role on the dynamic behavior of the system. S. R. Algule et al. [7] performed experimental study on a rotor to predict the unbalance in rotor. They observed unbalance in the rotor from spectrum analysis. The vibration was measured in terms of acceleration at five different speeds using FFT (Fast Fourier Transform) at initial condition. Based on vibration readings spectrum analysis and phase analysis was carried out to determine the cause of high vibrations. Then Rotor was balanced and found that vibrations were reduced. The experimental frequency spectra were obtained for both balanced and unbalanced condition under different unbalanced forces at different speed conditions. Md. Abdul Saleem et al. [4] performed experimental study on a rotor to predict the unbalance in rotor. The vibration was measured in terms of displacement at five different speeds using FFT (Fast Fourier Transform). They observed unbalance in the rotor from spectrum analysis and phase analysis. Based on vibration readings spectrum analysis and phase analysis was carried out to determine the cause of high vibrations and 1x frequency. Then Rotor was balanced and found that vibrations were reduced. The experimental frequency spectra were obtained for both balanced and unbalanced condition under different unbalanced forces at different speed conditions. Ritesh Fegade et al. [7] studied unbalanced response of rotor using ansys. They presented an alternative procedure called harmonic analysis to identify 1x frequency of a system through critical speed, amplitude and phase angle plots using ANSYS. The unbalance that exists in any rotor due to eccentricity has been used as excitation to perform such an analysis. Shamim Pathan and Pallavi Khaire [9] suggested to use flexible coupling to minimize the vibration. They studied that Rigid coupling is more prone to vibrations as compared to other couplings in unbalanced condition and flexible flange coupling can resists the vibration due to unbalance in better way. J.-J. Sinou [1] examined hardening-type non-linearity or softening-type non-linearity due to the effects of radial clearance and unbalance mass as excitation force. Mohammad Hadi Jalali et al. [5] analyze a high speed rotor in which critical speeds, operational deflection shapes, and unbalance response of the rotor are obtained in order to completely investigate the dynamic behavior of the rotating system. M. Chandra Sekhar Reddy and A.S. Sekhar [2] focus on torque measurements for misalignment diagnosis. Fourier and wavelet transforms are used to detect the misalignment fault. Slim Bouaziz et al. [10] presented research in which, the dynamic behavior of misaligned rotor is presented. He presented model which shows that the vibratory level of the angular misalignment decreases with the increase of the relative eccentricity, but increases when the imposed angle increases. W. J. Chen [11] studied that a very large and unstable vibration component typically ranging from 20% to 80% of the rotor speed, can be observed by dynamic signal analyzer or FFT emulator if the system is operated beyond the instability threshold or stability boundaries. The excessive vibration usually results in machine damage. Therefore, the determination of the instability threshold and stability parameters is critical for machine's safe operation. The instability threshold is usually determined from a stability map. A map of logarithmic decrements or damping coefficients versus the operating parameter is generated by repeated calculation of eigenvalues for a range of an operating parameter under study.

System description

The physical parameters for the rotor-bearing system are given in the following Table I.

S.No	Component	Parameter	Dimension(unit)
1	**Disc**	Diameter	120 mm
		Thickness	8 mm
		Density	7800 kg/m^3
		Mass	0.6 kg
2	**Rotor**	Diameter	23 mm
		Length of rotor	560 mm
		Density	7800 kg/m^3
		Mass	1.81 kg

		Bearing Span	420 mm
3	**Induction Motor**	-	1 hp A.C., 5000 rpm speed
4	**Vibration sensor**	-	Syscon SI-10, 230V, 50Hz.
5	**Tachometer**	-	-

The Experimental apparatus is shown in photograph of Figure 1 and Figure 2. It consists of a rotor having, single disc at 140 mm from MNDE in first case and two discs at equal and opposite direction from MNDE. The rotor shaft is supported by two identical ball bearings. The rotor is driven by induction motor. The speed of the motor is controlled by using speed regulator which is mainly used for A.C motors, to increase or decrease the speeds of the motor in the range of 0 to 5000 rpm. The instrument used in experiment includes tachometer for measuring speed of the rotor in RPM, vibration sensor connected to vibration indicator, measures the vibration in terms of velocity, displacement and frequency.

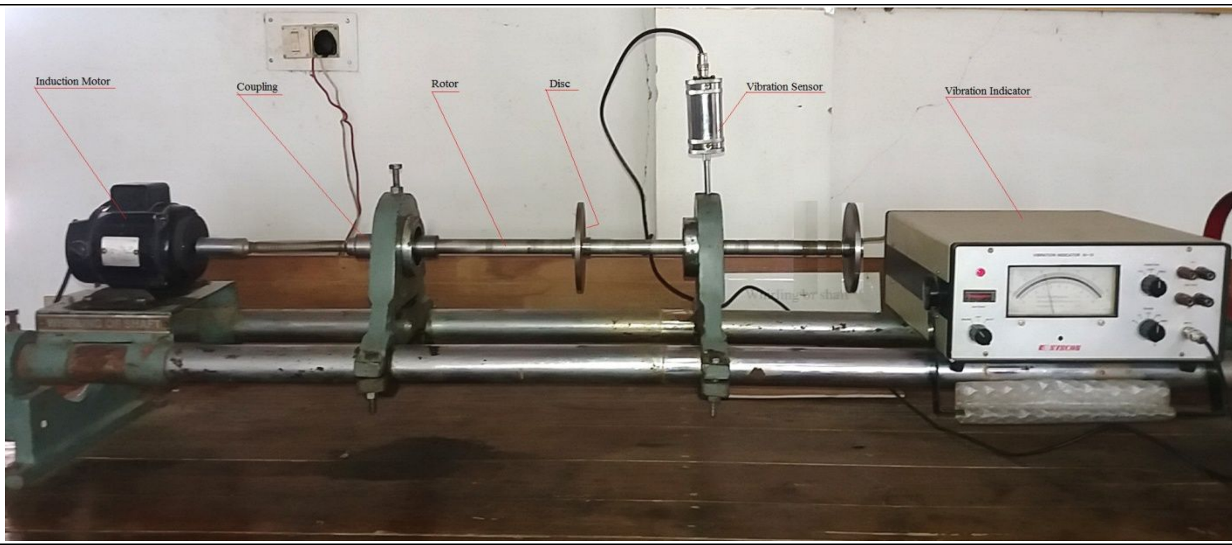

Fig. 1 Experimental Set-up

Experimental Procedure
First the setup is run for few minutes to settle down all minor vibrations. After that rotor having single disc at overhanging position, rotates at different speed. Then Vibration indicator acquires the vibration signals in terms of velocity, displacement and frequency. Vibration signals are measured at five different speeds 600, 800, 1000, 1200 and 1500 rpm at motor drive end (MDE) and motor non drive end (MNDE).
Then disc of same mass were added at exact opposite direction i.e. in balanced state and vibration signals were taken.

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Fig. 2 (a) Vibration Indicator (b) Tachometer

1) Comparisons of vibration amplitudes at MDE for unbalanced and balanced signals in terms of velocity:

S.No.	Speed	Unbalance	Balance
		Amplitude (mm/s)	Amplitude (mm/s)
1	600	6.89	2.48
2	800	13.36	7
3	1000	22.91	10.37
4	1200	83.53	18.76
5	1500	57.9	12.27

Above reading shows the vibration amplitude in terms of velocity at motor drive end for balanced and unbalanced condition. Above observation shows that vibration amplitude is much less in balanced condition than unbalance condition.

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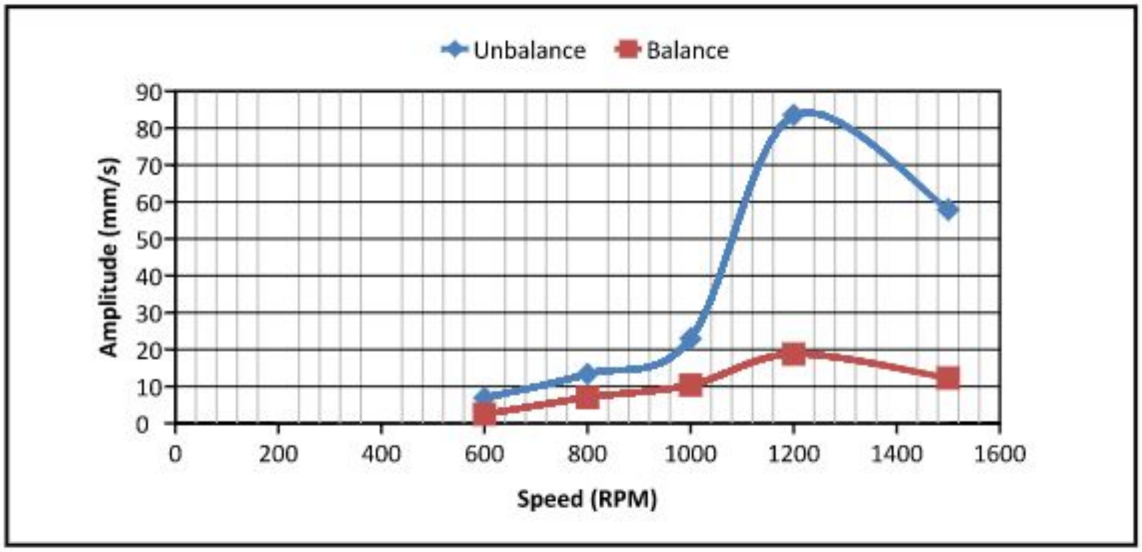
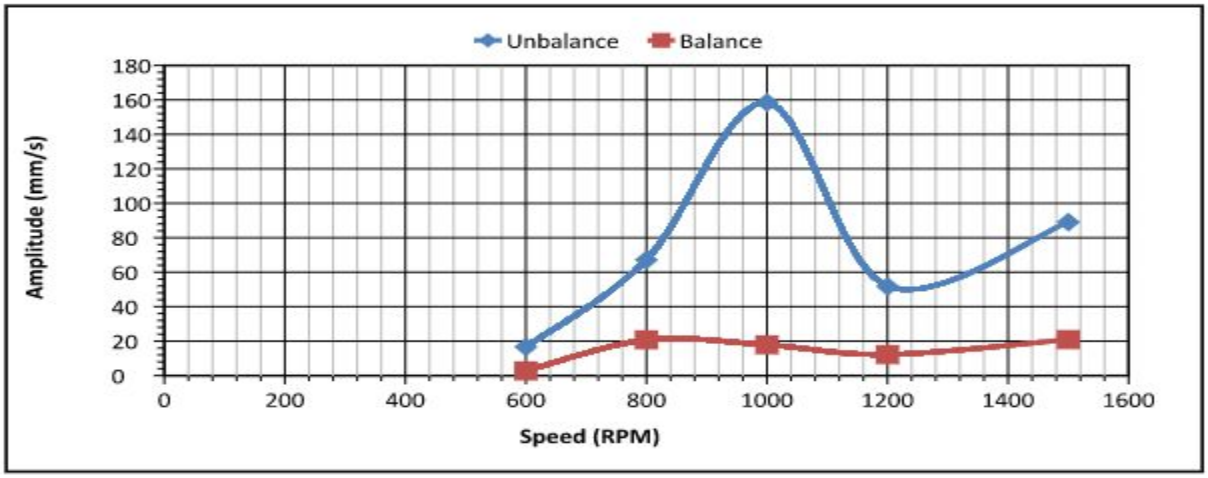


Above graph shows the vibration amplitude in terms of velocity at motor drive end for balanced and unbalanced condition. Graph shows that maximum amplitude occurs nearer to 1200 RPM in both cases (i.e. balanced and unbalanced condition). The amplitude suddenly increases in the range of 1000-1200 RPM.

2) **Comparisons of vibration amplitudes at MNDE for unbalanced and balanced signals in terms of velocity.**

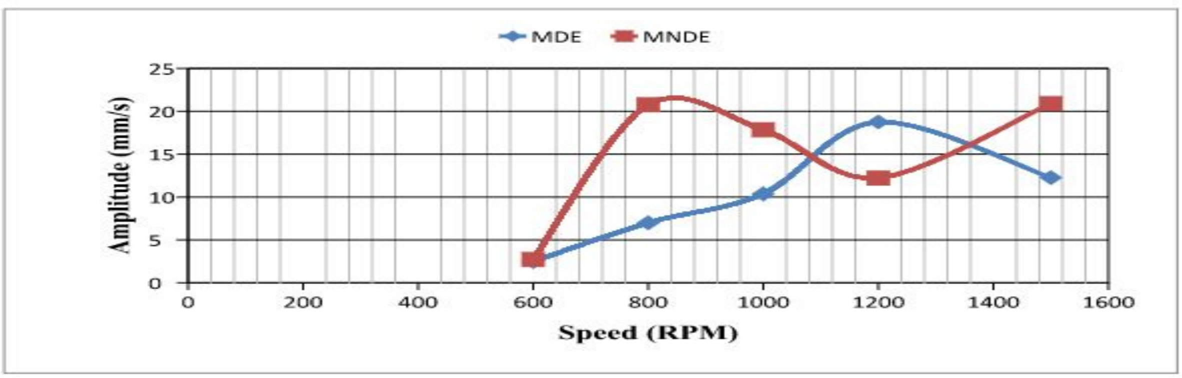
S.No.	Speed	Unbalance	Balance
		Amplitude (mm/s)	Amplitude (mm/s)
1	600	16.64	2.72
2	800	67.15	20.77
3	1000	158.5	17.85
4	1200	51.72	12.22
5	1500	89.06	20.9

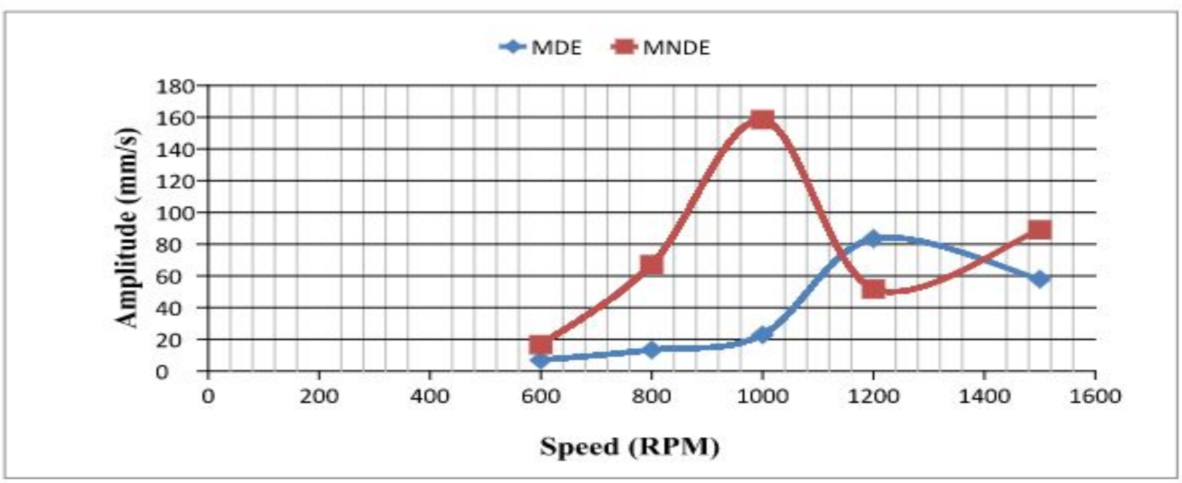
Above reading shows the vibration amplitude in terms of velocity at motor drive end for balanced and unbalanced condition. Above observation shows that vibration amplitude is much less in balanced condition than unbalance condition.



Above graph shows the vibration amplitude in terms of velocity at motor non drive end for balanced and unbalanced condition. Graph shows that maximum amplitude occurs nearer to 1000 RPM in both cases (i.e. balanced and unbalanced condition). The amplitude is high in the range of 800-1000 RPM.

3) Comparison of vibration amplitude for MNDE and MDE for balanced and unbalanced condition:





Above graph shows vibration amplitude for MNDE and MDE for balanced and unbalanced condition. By observing the graph we can see that the amplitudes are maximum at MNDE for both condition (balanced and unbalanced condition) i.e. maximum vibration occurs at overhung due to unbalance.

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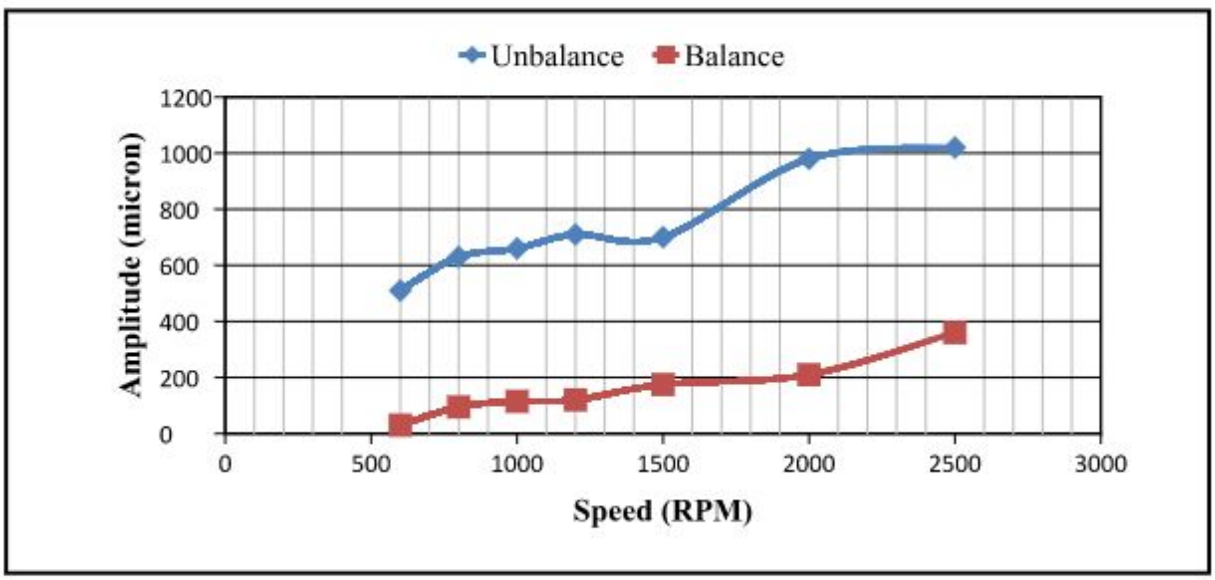
4) Comparisons of vibration amplitudes at MNDE for balanced and unbalanced signals in terms of displacement:

S.No	Speed	Unbalance	Balance
		Amplitude (micron)	Amplitude (micron)
1	600	510	30
2	800	630	95
3	1000	660	115
4	1200	710	120
5	1500	700	175
6	2000	980	210
7	2500	1020	360



Above graph shows the vibration amplitude for balanced and unbalanced condition in terms of displacement, which shows that the vibration amplitude is maximum in unbalance condition than balanced.

II. CONCLUSION

As the speed increases the amplitude at 1X is also increases for the same unbalance weight. This increase in amplitude value is because of the unbalanced force. Since the system frequency is nearer in the region of 1000-1200 due to the presence of resonance at this speed higher amplitudes were presented. Graphical analysis shows that there presents an unbalance in the rotor. Rotor is balanced and vibration readings are taken. It shows that amplitude of vibration is reduced drastically. This is a method to detect the fault in rotating machine. Hence Vibration monitoring method reduces the maintenance cost when it is applied to industries and improves the profit.

In this paper gyroscopic effect is not considered. If we consider the gyroscopic effect, the experimental values will be close to the theoretical values. The work may be extended for misalignment, looseness etc.

Nomenclature:

MNDE: Motor Non Drive End

MDE: Motor Drive End

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