International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 02 Issue: 01 | Mar-2015 www.irjet.net p-ISSN: 2395-0072

Vibration Signature Analysis of 4 Jaw Flexible Coupling Considering Misalignment in Two Planes.

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Abstract: Misalignment and unbalancing are the most possible causes of machine vibrations. A misaligned rotor always causes more vibration and generates excessive force in the bearing area and reduces the life of the machine. Coupling misalignment is a condition where the shaft of the driver machine and the driven machine are not on the same centerline. The non-coaxial condition can be parallel misalignment or angular misalignment. The more common condition is a combination of the two in both the horizontal and vertical direction. We are forced into this situation of coupling alignment because equipment from different suppliers must be mated together. Misalignment is temperature dependent. All materials expand with increasing temperature, and metal is no exception. Motors warm up several degrees, and the driven machine may warm up or cool down from ambient depending on the fluid it is handling. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs and increasing machine uptime. By the term two planes in our work, we mean that two rotors are used for the analysis of misaligned vibrations. If only one rotor is used then this system is called a single plane system. In this paper, experimental studies were performed on a 2 rotor dynamic test apparatus to predict the vibration spectrum for rotor misalignment. A 4 Jaw flexible coupling was used in the experiments. The rotor shaft accelerations were measured at rotor speed of 30 Hz i.e. 1800 RPM using accelerometer and a Dual Channel Vibration Analyzer (DCVA) under the aligned (baseline) and misaligned conditions. The experimental frequency spectrum was also obtained for both baseline and misaligned condition under different misaligned forces. The experimental results of aligned and misaligned rotors are compared at two different rotor locations.

Key words: 4 Jaw flexible Coupling • Misaligned Rotors • DCVA • Ball Bearings, FFT

1. INTRODUCTION

Rotor misalignment is the most common reason for machine vibrations. Most of the rotating machinery problem can be solved by aligning the rotors and balancing the rotors. Misalignment in a rotating system often produces excessive synchronous forces that reduce the life span of various mechanical elements. A very small amount of misalignment may cause severe problem in high speed rotating machines. Overhung rotors are used in many engineering applications like pumps, fans, propellers and turbo machinery etc. The vibration signature of the overhung rotor is totally different from the midway or intermediate rotors. The vibration caused by misalignment may destroy critical parts of the machine, such as bearings, seals, gears and couplings. Rotor misalignment is a condition in which the central axis of a rotating assembly, typically the shaft and its fixed components like disks and blades etc. is not coincident with the centre of rotation. In practice, rotors can never be perfectly aligned because of manufacturing errors such as porosity in casting, non- uniform density of material, manufacturing tolerances and gain or loss of material during operation. As a result of misalignment, a centrifugal force is generated and must be reacted against by the bearings and support structures.

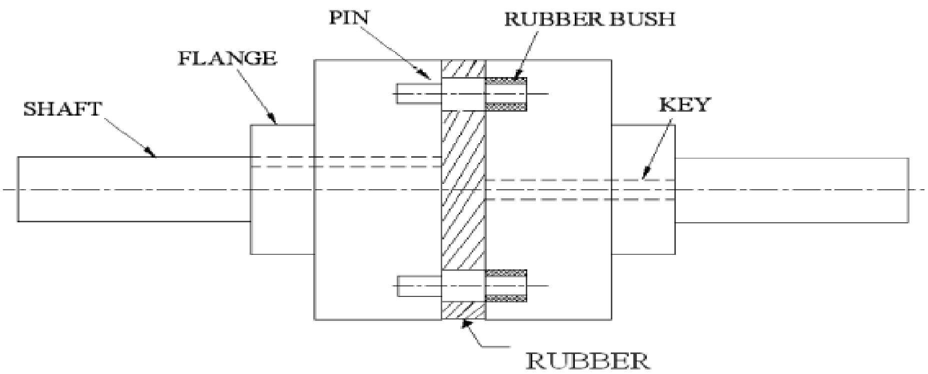

Fig. 1 Four Jaw flexible coupling with key and shaft

Misaligned response investigations of two stage turbine rotor systems were carried out by Arumugam et al. [1]. Dwell and Mitchel [2] detected misalignment of a disk coupling using spectrum analysis. Further, the truth behind misalignment vibration spectra of rotating machinery was analysed by Ganeriwala et al. [3]. Coupling misalignment forces were studied by Gibbons [4]. Then, dynamics of rotor bearing systems were analysed by Goodman [5]. Afterwards a shaft alignment handbook was presented by Piotrowski [6]. Further, Sekhar and Prabhu [7] studied the effects of coupling misalignment on vibrations of rotating machinery. After some time a study on the prediction of vibration behaviour of large turbo machinery on elastic foundation due to unbalance and coupling misalignment was presented by Simon [8]. A book on rotor dynamics of turbo machinery was presented by Vance [9]. Not stopping here, Xu and Marangoni [10] analysed theoretically the vibrations of a motor flexible coupling rotor system subjected to misalignment and unbalance. Same authors, Xu and Marangoni [11] did the experimental analysis of the vibrations of a motor flexible coupling rotor system subjected to misalignment and unbalance. Patel and Darpe [12], did the experimental investigations on vibration response of misaligned rotors. Dynamic response of two rotors connected by rigid mechanical coupling with parallel misalignment was studied by Hussain and Redmond [13]. Further, failure analysis of a misaligned and unbalanced flexible rotor was studied by Hili et al. [14]. Next to it, Saavedra and Ramirez [15] did experimental vibration analysis of rotors for the identification of shaft misalignment. In this paper a general method is presented for obtaining the misaligned response orbit based on the experimentation of a gear-coupled two rotor shaft bearing system, where the shaft may rotate at different speeds. In this paper aligned and misaligned system of 2 intermediate rotors and 2 overhung rotors are considered for the study. Experiments were conducted for two different positions of rotors (2 rotors intermediate) & (2 rotors overhung) for misalignment at rotor speed of 30 Hz (1800 rpm) and results are plotted. However the set up can also be made to take results at different rotor speeds. The rotor misalignment can be detected by FFT analysis. The vibration frequency of rotor misalignment is synchronous that is one times the shaft rotational speed, since the misalignment can be reduced significantly by aligning the rotors.

2. DESCRIPTION OF 4 JAW FLEXIBLE COUPLING

A 4 Jaw coupling is as shown in Figure 1. It has two flanges. One flange has a pin hole of required number at pitch circle diameter. The other flange will have a number of pins projected outside at pitch circle diameter to accommodate into the first flange holes with rubber bush. The driver and driven shafts are connected to their respective flanges i.e. output and input flanges by means of parallel square key. Mild-steel is considered for the input and output shafts, pins and keys. Over these pins, a circular natural rubber bush is provided and its length is equal to the length of the hole. The diameter of the flange holes is equal to the diameter of pin plus the thickness of rubber bush. The cast-iron material is chosen for both left and right flanges and the natural rubber is for bush. There is no nut and bolt to clamp the both input and output flanges. The Figure 1 represent the two dimensional model of 4 Jaw flexible coupling. In between flanges a rubber material is introduced to give the flexibility. Dimensions of the coupling and materials used are given in Table 1 and Table 2.

3. DESCRIPTION OF THE EXPERIMENTAL SETUP

The Experimental apparatus is shown in Figure 2 (two rotors in overhung position in two planes) and in Figure 3 (two rotors in intermediate position in two planes). It consists of a D.C. motor, a flexible coupling and a double disk rotor. The rotor shaft is supported by two identical ball bearings and has a length of 660 mm with a bearing span of 460 mm. The diameter of the rotor shaft is 20 mm. Two disks of 128 mm in diameter and 10 mm in thickness are mounted on the rotor shaft non drive end. The bearing pedestals are adjustable in vertical direction so that different misalignment conditions can be created. The rotor shaft is driven by 0.75 hp D.C. motor. The D.C. voltage controller known as control panel is used to adjust the power supply so that motor speed can be continuously increased or decreased in the range from 0 to 3000 rpm. The baseline signal has been measured at a rotor speed of 30 Hz. to check the concentricity.

The instruments used in the experiments include accelerometers and a vibration analyser (DCVA). The accelerometer directly measures the velocity of bearing housing vibrations and displays in the vibration analyser.

4. EXPERIMENTAL PROCEDURE

Experimental facility as shown in Figure 2 (with two rotors in overhung position in two planes) is used for both baseline and unbalance tests. First the setup is run for few minutes to settle down all minor vibrations. Before creating unbalancing, the shaft is checked for any misalignment and unbalance. After that an unbalance has been created by placing a mass of 18 gram in the overhung rotor at a radius of 54 mm. Accelerometer along with the vibration analyzer is used to acquire the vibration signals. The accelerometer is attached with the help of wires to take readings at three positions (Horizontal, Vertical and Axial) at NDE (Non Drive End) and DE (Drive End) for both motor and rotor. Following are the three positions for motor and rotor:

<u>MOTOR</u>

[NDE (H)]$_M$ – Horizontal Non Drive End of motor

Vibration signals are measured for baseline and unbalanced conditions at two different rotor locations at a frequency of 30 Hz or at a rotor speed of 1800 rpm both at non drive end (NDE) and drive end (DE) and stored in the vibration analyser for comparison later on.

[NDE (V)]$_M$ – Vertical Non Drive End of motor

[NDE (A)]$_M$ – Axial Non Drive End of motor

[DE (H)]$_M$ – Horizontal Drive End of motor

International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 02 Issue: 01 | Mar-2015 www.irjet.net p-ISSN: 2395-0072

$[DE (V)]_M$ – Vertical Drive End of motor
$[DE (A)]_M$ – Axial Drive End of motor

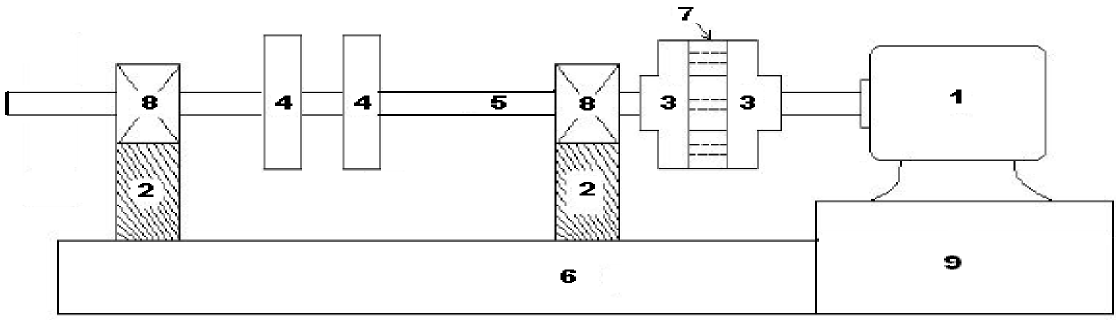

Fig. 4. Experimental set with two rotors at intermediate position in 2 planes

1- D.C Motor
2- Bearings supported in Plummer Blocks
3- 4 Jaw Pin type flexible coupling
4- Rotors (5-8 kg each, adjustable)
5- Rotor shaft
6- Base
7- Rubber
8- Ball bearings
9- Control Panel

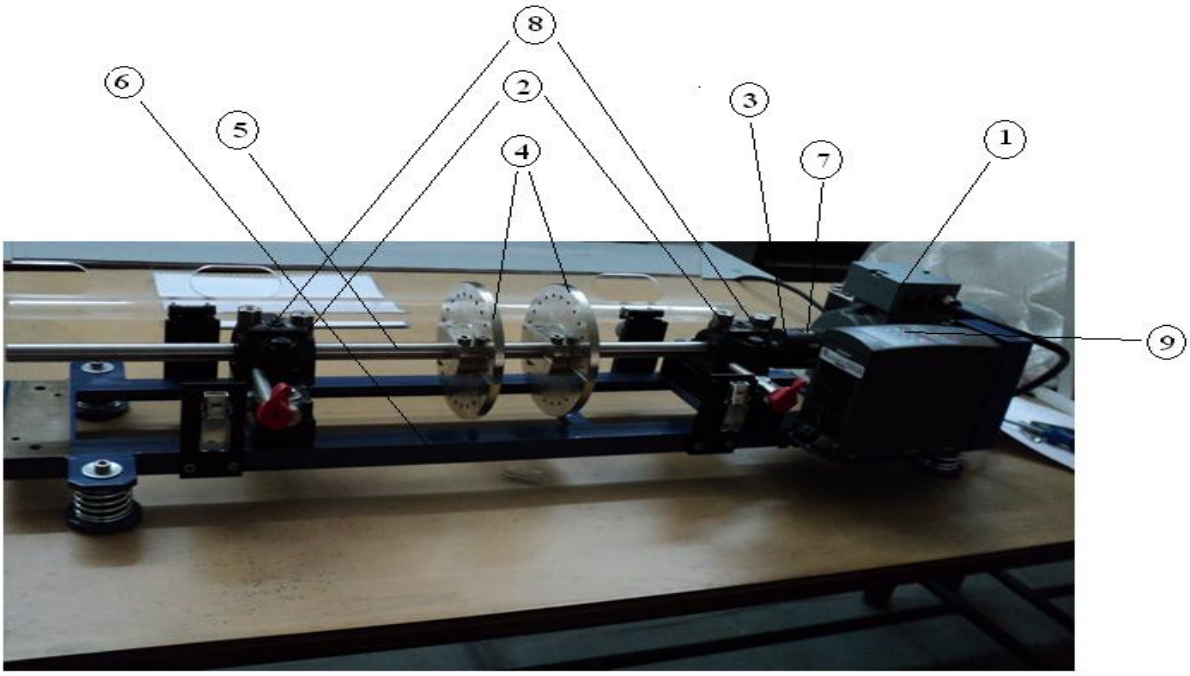

Fig. 3. Photograph of Experimental set up with two rotors at intermediate position in 2 planes

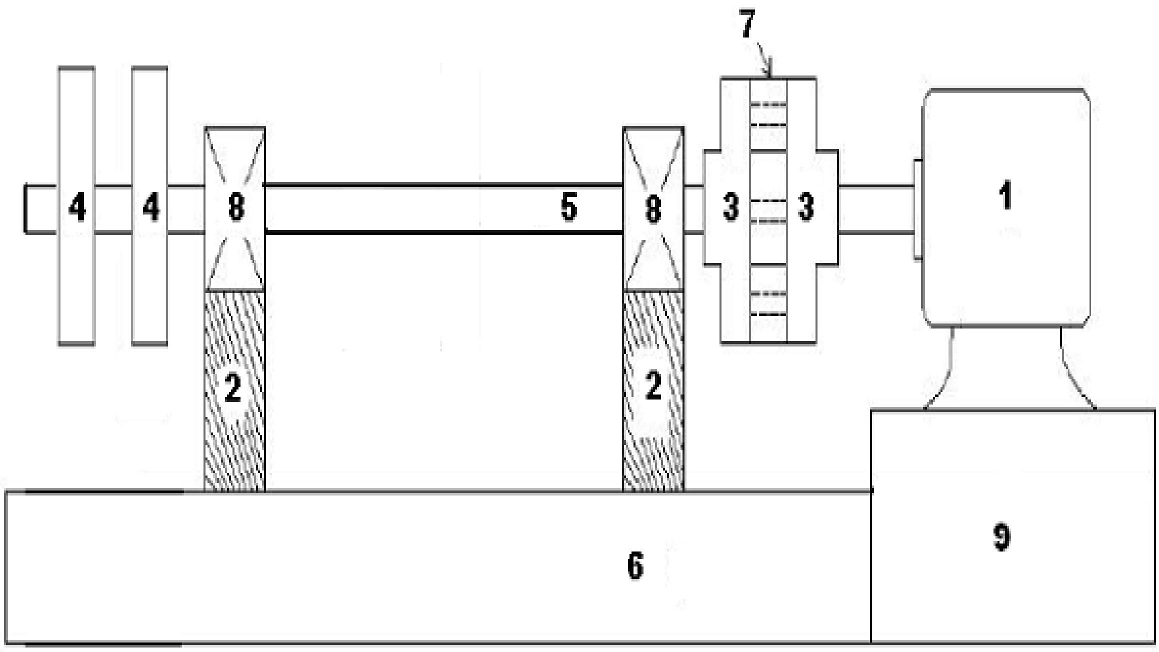

Fig. 4. Experimental set with two rotors at overhung position in 2 planes

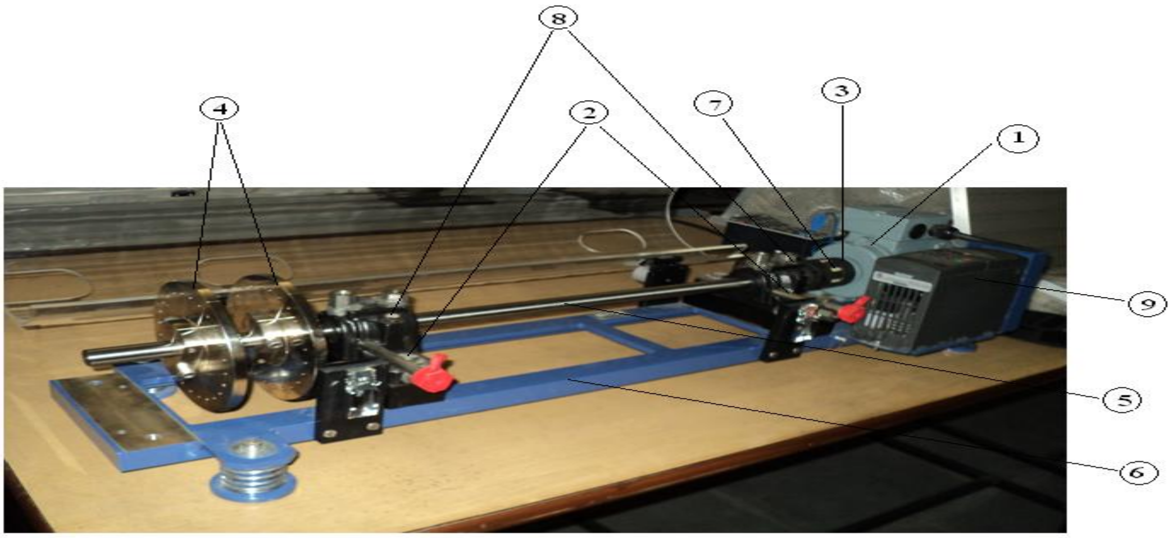

Fig. 5. Photograph of Experimental set up with two rotors at overhung position in 2 planes

Table1: Dimensions of Shaft and Coupling

SR. NO.	DESCRIPTION	UNIT
1.	Shaft Diameter	20 mm
2.	Length of the Shaft	630 mm
3.	Hub Diameter	40 mm
4.	Length of the Hub	30 mm
5.	Outside Diameter of Flange Coupling and Rubber Pad	80 mm
6.	No. of Holes for Pin	04
7.	Diameter of Pin Hole	4.2 mm
8.	Diameter of Pin	4.0 mm
9.	i) Rubber Bush Outside Diameter	11 mm
	ii) Rubber Bush Inside Diameter	06 mm
10.	**KEYWAY DEPTH**	
	i) In Shaft	04 mm
	ii) In Hub	03 mm
11.	**KEYWAY CROSS-SECTION**	
	i) Height	06 mm
	ii) Width	06 mm

Table2: Material Properties

PROPERTIES	CAST IRON	MILD STEEL	RUBBER
Young's Modulus, (MPa)	1×10^5	2×10^5	30
Poisson's Ratio	0.23	0.3	0.49
Density, (kg/mm^3)	7250×10^9	7850×10^9	1140×10^9

5. RESULTS AND DISCUSSION

4.1 Amplitude Trend for Two Rotors with Intermediate Position at Baseline and misaligned Conditions:

The experimental frequency spectra were obtained to the baseline condition. The perfect alignment and balancing cannot be achieved in practice. Thus, a baseline (well aligned) case is presented first to show the residual misalignment. The graphical comparison measured amplitude trend of vibrations in the form of acceleration of a baseline AND misaligned system at Drive End (DE) and Non Drive End (NDE) with 4 Jaw flexible coupling at a frequency of 30Hz (1800 RPM) is shown in figure 6. The baseline spectrum is measured experimentally using dual channel vibration analyser. Table 3 & 4 shows the comparison of experimental vibration amplitudes when two rotors are in intermediate position for both baseline and misaligned signals on MOTOR SIDE and ROTOR SIDE respectively. Figure 6 shows that the max. amplitude 3.681 m/sec^2 is observed in red curve at **[NDE (V)]$_M$** in misaligned condition but when the system is aligned a lower amplitude of 0.317 m/sec^2 is observed in blue curve at same location. This shows the dangerous level of misalignment. In this figure it can be seen that the amplitude trend changes from 1.534 m/s^2 to 3.044 m/s^2 with peak amplitude of 3.681 m/sec^2 in case of misaligned rotors at all different locations whereas for aligned or baseline case the amplitude changes from 0.235 m/s^2 to 0.274 m/s^2 with peak amplitude of 1.196 m/s^2 only. Similarly, for Figure 7 at the same location but along rotor side **[NDE (V)]$_R$** the maximum amplitude reaches 6.399 m/s^2 for misaligned rotors but for aligned rotors this value is just 0.344 m/s^2. In this case the amplitude for misaligned rotors changes from 1.233 m/s^2 to 4.853 m/s^2 with peak amplitude of 6.399 m/s^2 whereas for aligned or baseline case the amplitude changes from 0.155 m/s^2 to 0.134 m/s^2 with peak amplitude of 0.344 m/s^2 only.

Table 3:

Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (balanced) and misaligned signals along MOTOR SIDE

	Two rotors intermediate baseline signals (Motor) (Exp. Values in m/s^2)			Two rotors intermediate misaligned signals (Motor) (Exp. Values in m/s^2)					
							Increase in amplitude		
	H	**V**	**A**	**H**	**V**	**A**	**H**	V	**A**
Non Drive End $(NDE)_M$	**0.235**	**0.317**	**1.196**	**1.534**	**3.681**	**2.891**	1.299	3.364	1.695
Drive End $(DE)_M$	**0.170**	**0.310**	**0.274**	**3.197**	**1.418**	**3.044**	3.027	1.108	2.77

- **Graphical Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (aligned) and misaligned signals along MOTOR SIDE**

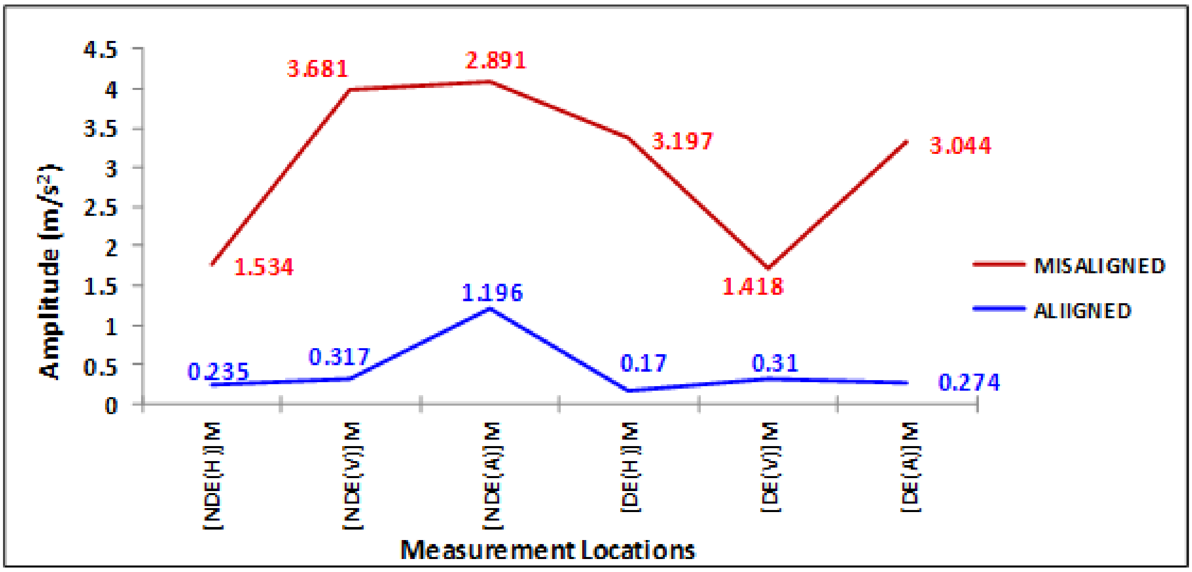

Fig. 6 Amplitude trends of NDE & DE of both motor and rotor in baseline and misaligned conditions with two rotors in INTERMEDIATE POSITION along MOTOR SIDE

Table 4:
Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (balanced) and misaligned signals along ROTOR SIDE

	Two rotors intermediate baseline signals (Rotor) (Exp. Values in m/s²)			Two rotors intermediate misaligned signals (Rotor) (Exp. Values in m/s²)					
							Increase in amplitude		
	H	**V**	**A**	**H**	**V**	**A**	**H**	**V**	**A**
Non Drive End $(NDE)_R$	**0.155**	**0.344**	**0.194**	**1.233**	**6.399**	**4.967**	1.078	6.055	4.773
Drive End $(DE)_R$	**0.286**	**0.144**	**0.134**	**5.146**	**4.819**	**4.853**	4.860	4.675	4.719

- **Graphical Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (aligned) and misaligned signals along ROTOR SIDE**

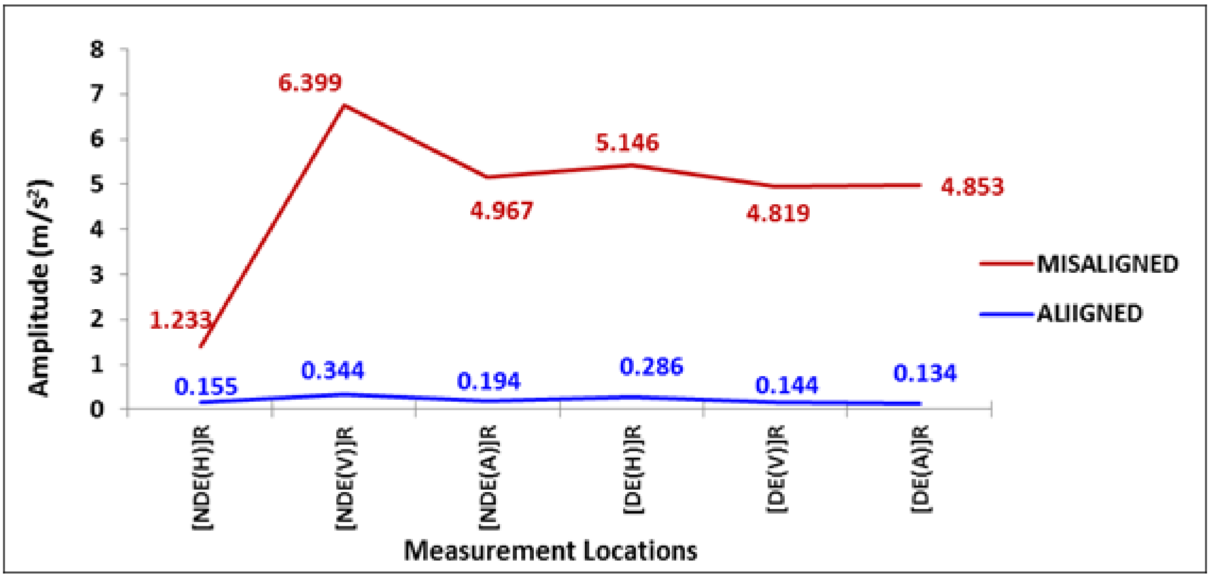

Fig. 7 Amplitude trends of NDE & DE of both motor and rotor in baseline and misaligned conditions with two rotors in INTERMEDIATE POSITION along ROTOR SIDE

This highest amplitude at this location is due to the residual forces acting on the system. Small amount of peaks at harmonics of shaft speed are the indications of manufacturing errors of coupling and other elements like rubber pad or rubber bush that cannot be eliminated. From the vibration spectrum of both DE and NDE for motor and rotor it is also observed that the 4 Jaw flexible coupling produce different amount of vibrations at different rotor locations discussed in this paper (2 overhung & 2 intermediate). So it is concluded that, 4 Jaw flexible coupling is to withstand much vibrations at different rotor locations. Experimental amplitudes are very accurately measured.

5.2 Amplitude Trend for Two Rotors with Overhung Position at Baseline and Misaligned Conditions

The experimental frequency spectra were obtained to the baseline condition. The measured amplitude trend of vibrations in the form of acceleration of a baseline AND misaligned system at Drive End (DE) and Non Drive End (NDE) with 4 Jaw flexible coupling at a frequency of 30Hz (1800 RPM) is shown in Figure 8 & 9. Table 5 & 6 shows the comparison of experimental vibration amplitudes when two rotors are in overhung position for both baseline and misaligned signals on MOTOR SIDE and ROTOR SIDES respectively. Figure 8 shows the maximum amplitude 13.788 m/s^2 is observed in red curve at **[NDE (A)]$_M$** in misaligned condition but when the system is aligned a lower amplitude of 0.297 m/s^2 is observed in blue curve at the same location. This shows the dangerous level of misalignment. In this figure it can be seen that the amplitude trend changes from 2.76 m/s^2 to 5.02 m/s^2 with peak amplitude of 13.788 m/sec^2 in case of misaligned rotors at all different locations whereas for aligned or baseline case the amplitude changes from 0.65 m/s^2 to 0.13 m/s^2 with peak amplitude of 0.86 m/s^2 only. Similarly, for Figure 9 at a different location of **[NDE (H)]$_R$** along rotor side the maximum amplitude reaches 9.25 m/s^2 for misaligned rotors but for aligned rotors this value is just 0.204 m/s^2. In this case the amplitude for misaligned rotors changes from 9.25 m/s^2 to 1.45 m/s^2 with peak amplitude of 9.25 m/s^2 whereas for aligned or baseline case the amplitude changes from 0.204 m/s^2 to 0.105 m/s^2 with peak amplitude of 0.635 m/s^2 only.

Table 5:
Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (balanced) and misaligned signals along ROTOR SIDE

	Two rotors overhung baseline signals (Motor) (Exp. Values in m/s^2)			Two rotors overhung misaligned signals (Motor) (Exp. Values in m/s^2)					
							Increase in amplitude		
	H	V	A	H	V	A	H	V	A
Non Drive End (NDE)$_M$	**0.646**	**0.860**	**0.297**	**2.760**	**2.210**	**13.788**	2.114	1.35	13.491
Drive End (DE)$_M$	**0.206**	**0.634**	**0.13**	**7.467**	**3.945**	**5.019**	7.261	3.311	4.889

Table 6:
Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (balanced) and misaligned signals along ROTOR SIDE

	Two rotors overhung baseline signals (Rotor) (Exp. Values in m/s^2)			Two rotors overhung misaligned signals (Rotor) (Exp. Values in m/s^2)					
							Increase in amplitude		
	H	V	A	H	V	A	H	V	A
Non Drive End (NDE)$_R$	**0.204**	**0.612**	**0.123**	**9.248**	**5.220**	**6.769**	9.044	4.608	6.646
Drive End (DE)$_R$	**0.635**	**0.333**	**0.105**	**2.420**	**4.348**	**1.454**	1.785	4.015	1.349

- **Graphical Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (aligned) and misaligned signals along MOTOR SIDE**

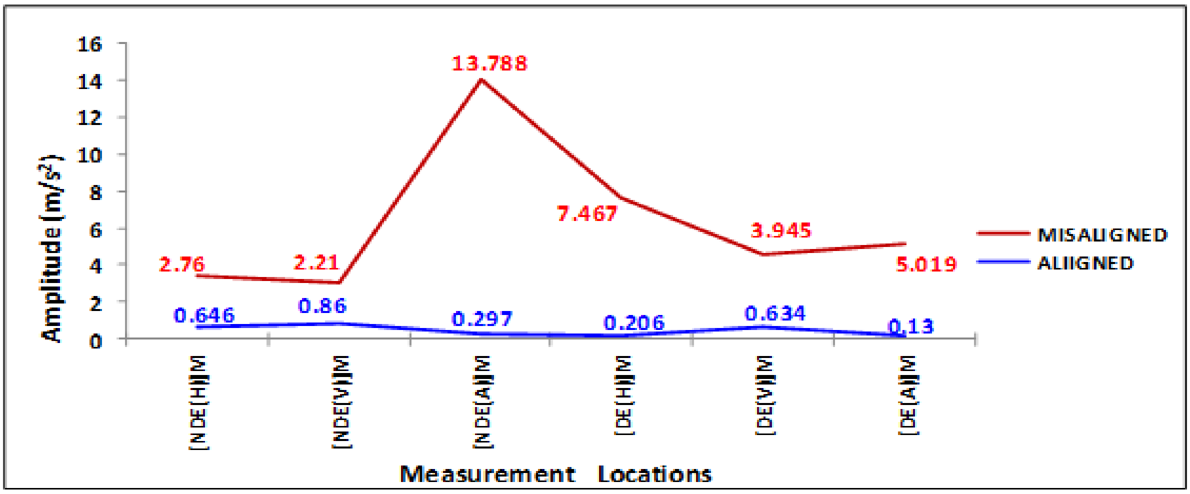

Fig. 8 Amplitude trends of NDE & DE of both motor and rotor in baseline and misaligned conditions with two rotors in INTERMEDIATE POSITION along MOTOR SIDE

- **Graphical Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (aligned) and misaligned signals along MOTOR SIDE**

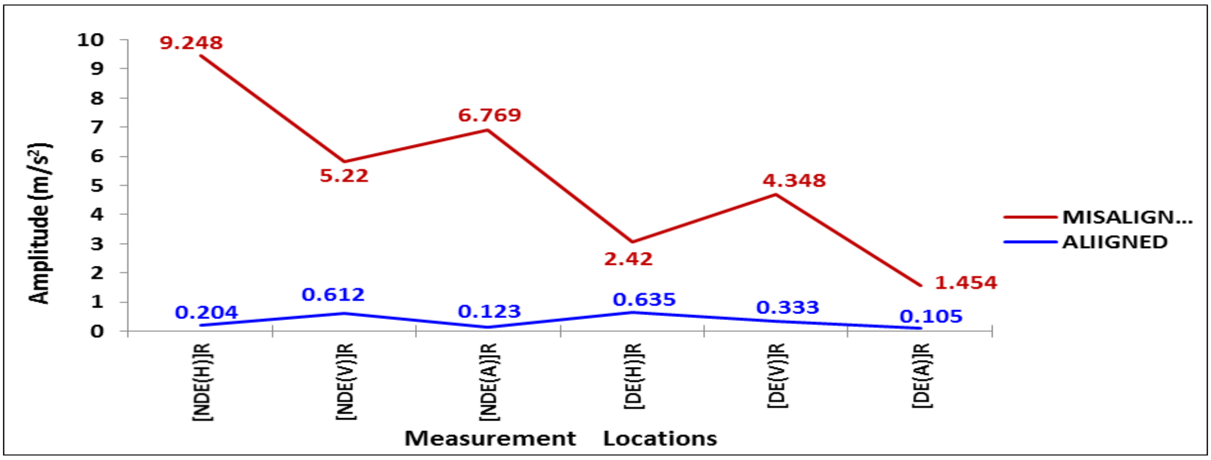

Fig. 9 Amplitude trends of NDE & DE of both motor and rotor in baseline and misaligned conditions with two rotors in INTERMEDIATE POSITION along ROTOR SIDE

6. CONCLUSION

The model of a 4 Jaw flexible coupling- ball bearing system with misalignment was studied experimentally. Throughout the experimental work, the validity of the model was successfully verified and the rotor dynamic characteristics related to misalignment were investigated. The experimental spectra were obtained. The experimental predictions are in good agreement with the condition monitoring techniques. The experimental results show that how the misalignment enhances the vibrations of a system in a drastic manner. It has been noticed in the experiment that the peak amplitudes occur at a frequency of 30Hz. At other frequencies there is an ignorable increase in peaks of amplitudes. Based on this experiment a condition monitoring of industrial machines prone to vibrations can be done with the help of spectrum obtained. If the peak amplitudes are occurring at 30Hz frequency then automatically it is to be sure that there is some unbalancing in the machine which has to be balanced his spectrum can also be done It has also been seen that the maximum peaks of amplitudes are occurring in the second case (2 rotors in intermediate position) as compared to the first case (2 rotors in intermediate position). This means that the system in the second case is more unstable as compared to the first case. However, at some positions the unbalance effect is not close enough to system natural frequencies to excite the system appreciably. Therefore, at these positions the unbalance response is hidden and does not show up in the vibration spectrum. On the other hand if the unbalancing frequency is at or close to one of the system's natural frequencies, the unbalanced effect can be amplified and high amplitude in the form of velocity level is pronounced in the spectrum. As further unbalancing is created by adding more weights the amplitude also increases for the same rotor positions. This increase in amplitude is because of the more centrifugal forces acting on the system due to unbalancing. There are many other locations of rotors at which vibrations spectra can be studied experimentally.

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