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A General Paper Review on Effect of Misalignment on Vibration Response of Coupling

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ABSTRACT

This paper presents the view of different author regarding vibration response of coupling due to misalignment in machinery. The effect of misalignment had been explained by performing controlled experiment on experimental setup fabricated by the respective author. The results and conclusion obtained from these experimentations shows the need for continuing the study on effect of misalignment on coupling performance. Hence, an attempt is being made to study the dynamics of coupling and the effect of load variation on coupling performance.

Keywords— Misalignment, Coupling, Vibration analysis.

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I. EXPERIMENTAL INVESTIGATION OF EFFECT OF PARALLEL MISALIGNMENT IN ROTATING MACHINERY[1]

In this paper vibration response of a jaw coupling under parallel misalignment is shown. The Fig.1 shows the experimental setup used for studying the effect of misalignment on vibration response of a jaw coupling.

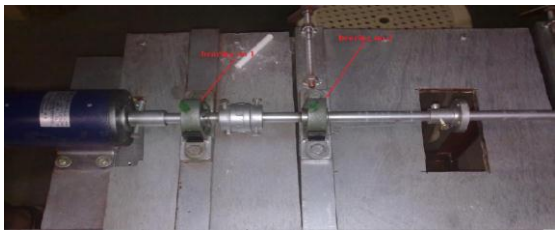


Figure1. Experimental setup

Vibration spectrum for aligned system and parallel misaligned system is obtained and then the spectra are compared.

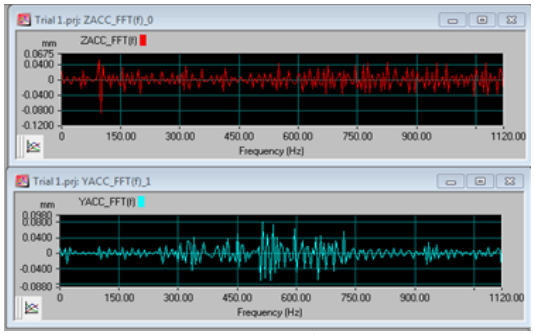


Figure 2. Vibration spectra for aligned system

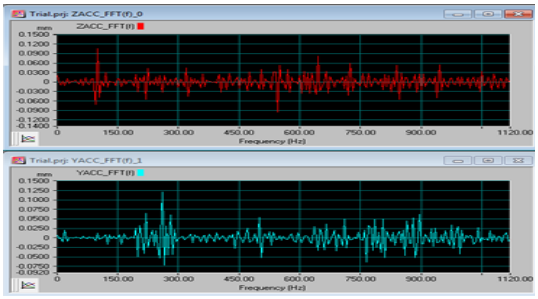


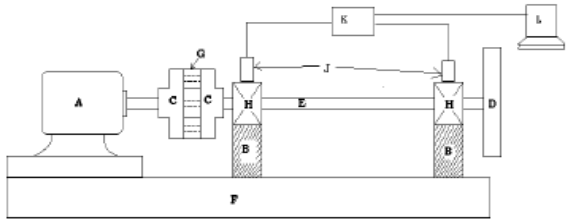
Figure 3. Vibration spectra for misaligned system

A comparative study of vibration spectra obtained from aligned system and misaligned system reveals that presence

of parallel misalignment can be identified by observing peak in vibration spectrum of misaligned system. Vibration spectra show increase in amplitude in axial direction is more than radial direction.

II. VIBRATION ANALYSIS OF MISALIGNED SHAFT –BALL BEARING SYSTEM[2]

The author has shown that 2X shaft running speed is the predominant frequency indicating shaft misalignment. In this experiment the model of the pin type coupling-ball bearing system with misalignment was simulated. The experimental and numerical (ANSYS) frequency spectra were obtained. The experimental predictions are in agreement with the ANSYS results. Both the measured and ANSYS results spectra shows that misalignment can be characterized primarily by 2X (two times) shaft running speed. The experimental apparatus consists of a D.C motor, a flexible coupling and a single disk rotor. 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 is used to adjust the power supply so that motor speed can be continuously increased or decreased in the range from 0 to 2000 rpm. The misalignment case of 0.2 mm between driver and driven shaft results are shown in table 1.



A: D.C Motor; B: Bearing Support; C: Coupling; D: Disk; E: Rotor Shaft; F: Base; G: Rubber; J: Piezo electric accelerometer: K: Vibration Analyzer; L: Computer

Figure 4: Experimental setup

TABLE I

ACCELERATION (m/s^2) VALUES AT DIFFERENT SPEEDS

RPM	Experimental Results						ANSYS Results					
	Drive End			Non Drive End			Drive End			Non Drive End		
	1x	2x	3x	1x	2x	3x	1x	2x	3x	1x	2x	3x
500	0.0125	0.015	0.016	0.02	.025	.021	0.012	0.015	0.021	0.014	0.02	0.024
1000	0.035	0.036	0.031	0.048	0.045	0.043	0.024	0.03	0.026	0.03	0.04	0.044
1500	0.03	0.039	0.049	0.039	0.049	0.055	0.039	0.049	0.042	0.039	0.045	0.055
2000	0.05	0.052	0.062	0.065	0.069	0.079	0.049	0.052	0.063	0.059	0.062	0.077

III. AN ANALYSIS OF THE IMPACT OF FLEXIBLE COUPLING MISALIGNMENT ON ROTOR DYNAMICS.

The author investigated the source of the 2N vibration response seen in misaligned vibrating machinery by simulating misalignment through a coupling. Three flexible disc-pack couplings (4-bolt, 6-bolt, and 8-bolt coupling) were modeled and parallel and angular misalignments were simulated using a finite element program. The stiffness terms obtained from the coupling simulations had 1N, 2N, and 3N harmonic components. The 4-bolt coupling had large 1N reaction components under angular and parallel misalignment. The 6-bolt coupling model only had a 1N reaction component under angular misalignment, and both cases of parallel misalignment showed a strong 2N reaction component, larger than both the 1N and 3N components. The 8-bolt coupling model under angular misalignment produced large 1N reaction components. Under parallel misalignment, it produced 1N, 2N, and 3N components that were similar in magnitude. All the couplings behaved linearly in the range studied. A 5-tilting pad journal bearing was also tested to better understand its behavior under misalignment. The response of the 5-tilting pad bearing did not produce any 2N components while the bearing was subjected to unit loads of up to 34.5 bars.

The experimental results are shown with the help of water fall plot of the rotor response in y and x directions with no load, unit load of 17.4 bars and unit load of 34.5 bars.

Fig. 5 and Fig. 6 shows the waterfall plot of the rotor response in the Y direction and X direction respectively with no unit load. The synchronous (1N) component dominates the response while a relatively small 2N and 3N components are also present .This figure was used as a baseline to compare against the other tests where the bearing was loaded.

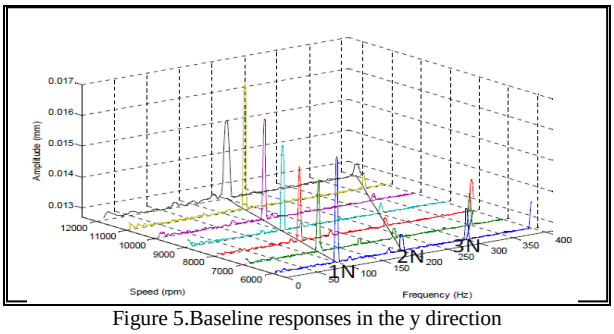


Figure 5.Baseline responses in the y direction

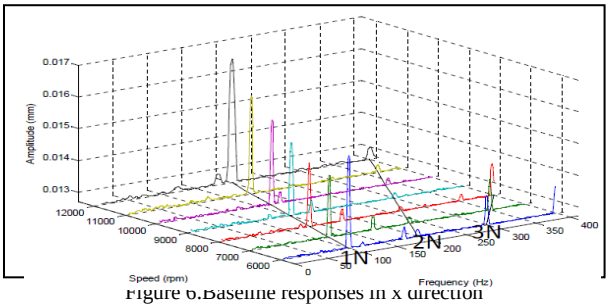


Figure 6.Baseline responses in x direction

Fig. 7 shows the response in the Y direction for a unit load of 17.2 bars. Except for the 12000 rpm case, there is no apparent growth in the 2N or 3N frequency components. Figure 8. shows the response in the X direction. In the X direction, the 12000 rpm case shows less growth in the 2N component than in the load direction. Even though the results for the 12000 rpm case with 17.2 bars unit load shows some level of 2N excitation, the results for the next unit load of 34.5 bars will show that this trend does not continue. Also, the amplitude of the synchronous response decreased when the load was applied compared to the baseline response. This last fact shows that the bearing was being loaded properly and that the proximity probes were working correctly.

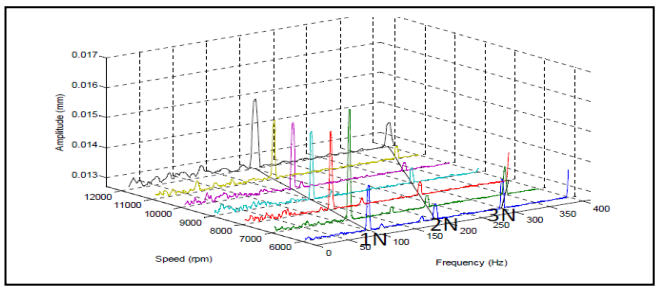


Figure 7. responses in y direction with unit load of 17.2 bars

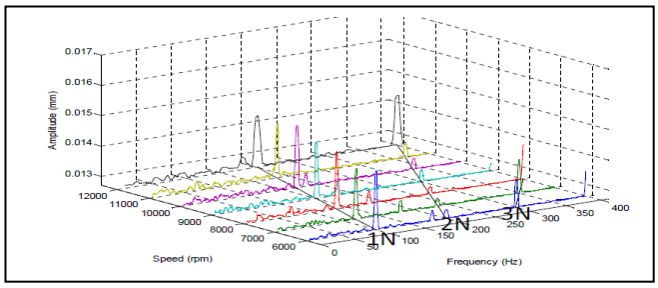


Figure 8 responses in x direction with unit load of 17.2 bars

Fig. 9 shows the response in the *Y* direction for a unit load of 34.5 bars. The 2N and 3N components had approximately the same amplitude as the previous 17.2 bars
unit load cases. The synchronous response also remained approximately constant as compared to the previous load. Fig. 10 shows the response in the *X* direction. It has the same characteristics as Figure 9.1. Note that doubling the unit load to 34.5 bars seemed to even reduce the amplitude of the 2N component in some of the cases. This trend can be seen in the 12000 rpm case where the amplitude was significantly reduced. Throughout all the tests, there was no indication that having a high load, such as a unit load of 34.5 bars, could create or increase the 2N or 3N vibration frequency components of the response.

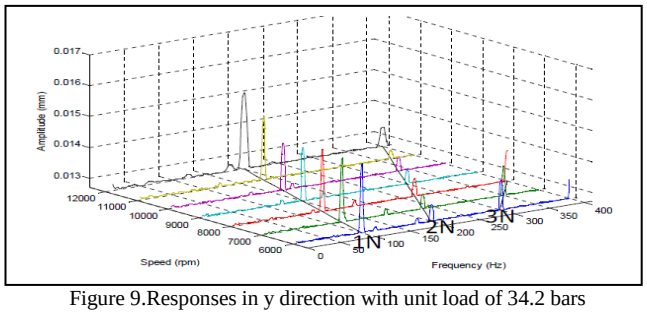


Figure 9.Responses in y direction with unit load of 34.2 bars

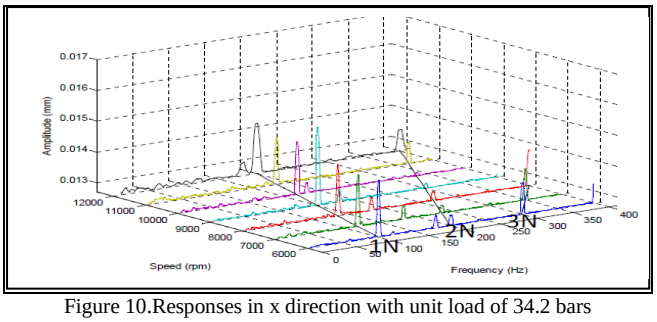


Figure 10.Responses in x direction with unit load of 34.2 bars

IV. VIBRATION ANALYSIS OF 4 JAW FLEXIBLE COUPLING CONSIDERING UNBALANCE IN TWO PLANES

In this paper, experimental studies were performed on a 2 rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance. A 4 Jaw flexible coupling was used in the experiments. The rotor shaft velocities were measured at rotor speed of 30 Hz using accelerometer and a Dual Channel Vibration Analyzer (DCVA) under the balanced (baseline) and unbalanced conditions. The experimental frequency spectrum was also obtained for both baseline and unbalanced condition under different unbalanced forces. The experimental results of balanced and unbalanced rotors are compared at two different rotor locations.

The Fig 11 shows the set up for two rotors in overhung position in two planes and Fig. 12shows the set up for two rotors in intermediate position in two planes.

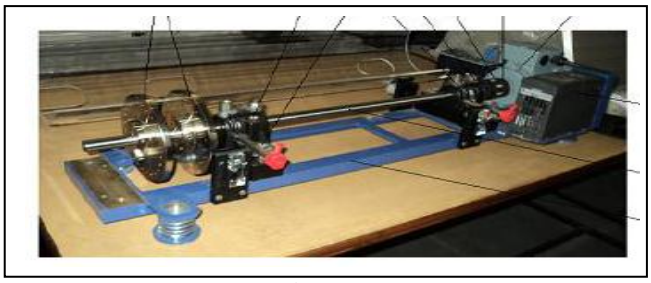


Figure 11. Setup of rotors with overhung position

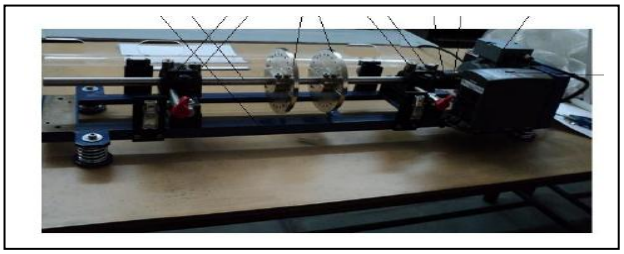


Figure 12.Setup of rotors with intermediate position

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. It was found that by creating an unbalance the amplitude of the system increases. This

amplitude increase was noticed in all the positions (Horizontal, Vertical and Axial) of the unbalanced system but severe boosting of the amplitudes was observed in the vertical position because of increase in centrifugal forces acting on the system.

V. CONCLUSION

There are three general statements that can be made form a number of controlled tests where rotating machinery was purposely misaligned and also, from many filed observations that have been made on equipment that was operating under misalignment conditions.

1. You cannot detect the severity of misalignment using vibration analysis. In other words, there is no relation between the amount of misalignment and the level / amplitude of vibration.
2. The vibration signature of misaligned rotating machinery will be different with different flexible coupling designs, For example, a misaligned gear coupling will not show the same vibration pattern as a misaligned 'rubber tire' type coupling.
3. Misalignment vibration characteristics of machinery rotors supported in sliding type bearings are typically different than the vibration characteristics of machinery rotors supported in antifriction type bearings.equations are an exception to the prescribed specifications of this template

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