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CONDITION MONITORING OF SHAFT FOR MISALIGNMENT DETECTION USING VIBRATION SIGNATURES

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ABSTRACT

Misalignment is the most common cause of machine vibrations. Vibration monitoring is a useful technique for providing useful information regarding symptoms of rotating machinery failures. This paper explains the investigation of vibration signatures at bearing away from motor of test rig due to effect of simulated faults. The faults simulated are parallel and angular misalignment of different magnitudes. The vibrations occurring in horizontal and vertical directions were measured by accelerometer combined with Fast Fourier Transform (FFT) analyzer. The study was carried out for three elastomers namely Nitrile Butadiene Rubber, Hytrel and Polyurethene. The Root Mean Square (R.M.S.) values of acceleration due to vibration were obtained for both parallel and angular misalignment conditions. The data obtained is compared on common basis of R.M.S. values of acceleration and magnitude of misalignment for different speeds. The MATLAB software is used to generalize the obtained data by curve fitting tool. These curves can be used to select the elastomers in advance according to operating conditions. The experimental data is simulated using ANSYS software. The experimental and simulated results were compared and found to be close.

KEYWORDS: Condition Monitoring, Elastomer, FFT Analyzer, Misalignment, Root Mean Square (R.M.S.) Value of Acceleration, Vibration Analysis

INTRODUCTION

The shaft alignment is an important task when installing and maintaining machinery. The coupling wear, bearing failures, bent rotors, bearing housing damage are all common results of poor alignment. The load on mechanical parts such as bearings, seals and couplings decreases with improved alignment. There is an increased tendency to get exact information about the predictive parameters based on system analysis through various diagnostic techniques to judge the health of a machine. The vibration analysis is one of the techniques that can be used to determine the mechanical condition of a machine. The advantage of vibration analysis is that it can identify the fault before it becomes serious and cause failure. A vibration signature measured at the external surface of machine can provide information regarding running condition of a machine. In this paper study on effect of machinery faults on the rotor vibrations has been carried out using vibration monitoring techniques.

Literature Review

Nakhaeinejad M. and Ganeriwala S. [1] in their technical paper have investigated dynamic effects of angular and parallel misalignment in rotating machinery. The authors showed that misalignment can generate excessive vibrations. Misalignment can excite vibration harmonics from 2X to 10X.The bearing housing forces are very sensitive to type and level of misalignment. The coupling stiffness can change the forces and vibrations significantly. The vibration signals in three directions can help to correlate results with stiffness of coupling.

Hariharan V. and Srinivasan PSS. [2] compared vibration signatures of rigid and self designed 3-pin type flexible coupling using experimentation and simulation. According to the researchers, misalignment can be characterized primarily by second harmonics (2X) of the shaft speed.

Patel T. H. and Darpe A. K.[3] proposed an experimental approach for determination of magnitude and harmonic nature of misalignment excitation. According to the authors, the 2X harmonics is not sufficient for misalignment diagnosis since other faults such as fatigue crack exhibit strong 2X harmonics. The full spectrum analysis and orbit plot techniques are effectively used in the order to reveal the distinguishing nature of misalignment.

Saavedra P.N. and Ramirez D.E. [4] describe theoretical and experimental work on the vibration behavior of coupled rotor with flexible couplings. A theoretical model capable of describing the mechanical vibration resulting from shaft misalignment is developed. The authors concluded that the vibrations generated by shaft misalignment is caused by a variation in coupling stiffness as the shaft rotates and the forcing frequencies generated by shaft misalignment are composed of shaft harmonics of the shaft speed.

Sekhar A.S. and Prabhu B.S. [5] modeled a rotor coupling bearing system using higher order finite elements. The effect of misalignment on vibration response in two harmonics at different speeds has been evaluated. The 2X vibration response shows the characteristics signature of misaligned shafts. The vibration response in different harmonics can be evaluated by using the present model.

Saavedra P.N. and Ramirez D.E. [6] performed experimental tests to validate the theoretical model for misaligned shaft, coupling, bearing system. The authors showed that vibratory response on misalignment is caused by the variation in coupling stiffness. The authors have studied effect of the load on spectral amplitudes. The authors showed that the load has a great influence on the amplitude of the spectral components and it can be a determinant factor when diagnosing the condition of a machine. The waveform analysis is used to distinguish between misalignment and mechanical looseness.

Pandey S. and Nakra B. C. [7] designed and fabricated test rig to investigate the vibrations at bearing due to effect of simulated faults. The faults simulated in the test rig are parallel misalignment, angular misalignment, combined parallel and angular misalignment, unbalance. Root mean square vibration accelerations in vertical, horizontal and axial directions of rolling element bearing were monitored using piezoelectric accelerometer to study the effect of simulated faults. The authors concluded that for all types of misalignment; the amplitudes of vibration are higher at bearing away from motor than at bearing near the motor.

The literature review has been carried out in the area of vibration monitoring of rotating machinery. The various techniques of fault diagnosis and analysis of resulting vibration signatures have been reviewed. This paper explains behaviour of misaligned shaft using different elastomers. It can be useful for condition monitoring of rotating machinery.

EXPERIMENTAL SETUP AND INSTRUMENTATION

The test rig consists of input and output shaft connected by Love-joy type flexible coupling. The elastomers used for experimentation namely Nitrile Butadiene Rubber, Hytrel and Polyurethene are inserted between two hubs of coupling. A D.C. drive is used to avoid fluctuations in speed. The whole setup is mounted on a base plate that rests on “C” channel.

The "C" channel is bolted to concrete foundation. The instrumentation used in experimental setup consists of an accelerometer mounted at housing of bearing away from motor. The output of accelerometer is connected to input of Fast Fourier Transform (FFT) analyzer. The NVGate software is used for vibration signature acquisition, recording and analysis. The photograph of the test rig used for experimentation is shown in Figure1.

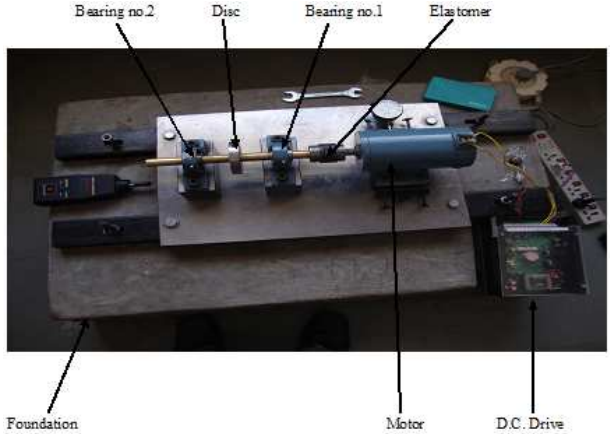

Figure 1: Photograph of Test Rig

The specifications of instruments and devices are described as follows

Table 1

Output shaft	Material- brass	Diameter -19 mm	Length- 275 mm
Disc	Material- M.S.	Diameter -75 mm	Thickness- 15 mm
D.C. Motor	0.25 h.p.	Maximum Speed -2000 r.p.m.	

Experimentation

The experimentation consists of balancing the shaft, coupling and disc assembly, to ensure base alignment by face and rim indicator method and creating the faults to obtain vibration signatures. The vibration signatures obtained should be exclusively of misaligned shaft and to ensure this, the shaft is dynamically balanced. The initial misalignment between driving and driven shaft should be as minimum as possible before inducing misalignment of known magnitude. This is achieved by using face and rim indicator method.

Creation of Faults

The test rig used for experimentation is shown in Figure1. The two type of faults created are parallel and angular misalignment.

Parallel Misalignment

The parallel misalignment is created by moving the motor in horizontal plane. A provision has been made in test rig to create misalignment of different magnitudes. The amount of misalignment created is measured by dial gauge having a least count 0.001 mm. A parallel misalignment of magnitudes 0.1, 0.2, 0.3, 0.4, 0.5, 0.6 and 0.7 mm is created. The shaft is rotated at selected speed of 600, 900, 1200, 1800 r.p.m. The accelerometer is mounted in horizontal and vertical position at bearing no. 2. The vibration spectrum for 0.2 mm misalignment with elastomer as Nitrile Butadiene Rubber, speed 600 r.p.m. in horizontal direction is shown in Figure 2. The R.M.S. values of acceleration obtained for Nitrile Butadiene Rubber having parallel misalignment in horizontal and vertical direction are shown in Table 1and Table 2.

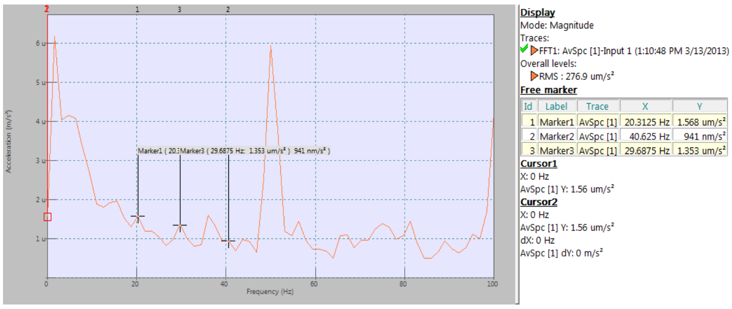

Figure 2: Vibration Spectrum of 0.2 mm – Nitrile Butadiene Rubber- 600 r.p.m. - Horizontal Direction

Table 2: R.M.S. Values of Acceleration (μm/s^2) for Nitrile Butadiene Rubber in Horizontal Direction at Different Speeds for Parallel Misalignment

Speed (R.P.M.)	Parallel Misalignment (mm)						
	0.1	0.2	0.3	0.4	0.5	0.6	0.7
600	268.4	276.9	292	315.7	325.7	364.4	386.9
900	474.4	507	514.7	519	542	546	553.1
1200	631.6	707.3	760	765	785	833.8	928
1800	797.7	931.1	1510	1602	1664	1897	2164

Table 3: R.M.S. Values of Acceleration (μm/s^2) for Nitrile Butadiene Rubber in Vertical Direction at Different Speeds for Parallel Misalignment

Speed (R.P.M.)	Parallel Misalignment (mm)						
	0.1	0.2	0.3	0.4	0.5	0.6	0.7
600	68	83.1	106.4	139.7	169.4	210.1	227.7
900	97.7	114.4	136.6	164.5	186	228.7	253.1
1200	131.3	146.2	166.6	186.5	204.4	255	267.7
1800	154	169.3	181.4	222.8	258.2	289.7	320.9

The same set of experimentation is carried out for other elastomers Hytrel and Polyurethene.

Angular Misalignment

The angular misalignment is created by moving motor in angular plane. The angular misalignment created is 0.045°, 0.09°, 0.135°, 0.18°, 0.225°, 0.27° and 0.315°. The R.M.S.values of acceleration obtained for Nitrile Butadiene Rubber having angular misalignment are shown in Table 3and Table 4.

Table 4: R.M.S. Values of Acceleration (μm/s^2) for Nitrile Butadiene Rubber in Horizontal Direction for Different Angular Misalignment

Speed (R.P.M.)	Angular Misalignment (Degrees)						
	0.045°	0.09°	0.135°	0.18°	0.225°	0.27°	0.315°
600	161.2	179.7	266.2	270.9	275.9	292	320.9
900	220.9	310.3	333.5	359.1	378.5	404.6	419.5
1200	374.3	376.1	440	474.4	682	742	894.9
1800	442.1	496.8	518.3	545.9	1194	1318	1465

Table 5: R.M.S. Values of Acceleration (μm/s^2) for Nitrile Butadiene Rubber in Vertical Direction for Different Angular Misalignment

Speed (R.P.M.)	Angular Misalignment (Degrees)						
	0.045°	0.09°	0.135°	0.18°	0.225°	0.27°	0.315°
600	98.6	114.4	186	219.2	240.2	268.3	386.7
900	154	177.3	209.5	222.8	400.1	440.9	504.6
1200	219.6	230.6	284.1	291.7	453.3	583	593.3
1800	291.7	328	355.7	425.7	1111	1257	1291

The same set of experimentation is carried out for other elastomers Hytrel and Polyurethene.

Generalization of Misalignment Data for Condition Monitoring

The data collected for parallel and angular misalignment is compared based on R.M.S. value of acceleration and magnitude of misalignment to generalize the relationship between them. The curve drawn for parallel misalignment with elastomer Polyurethene at speed 600 r.p.m by polynomial curve fitting technique using MATLAB software is shown in Figure 3. The equation of this curve is shown in Figure 3.

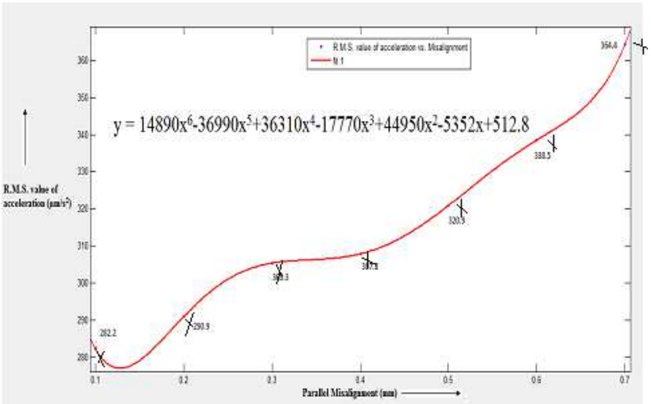

Figure 3: Curve Fitting for 600 R.P.M., Parallel Misalignment, Polyurethene Elastomer in Horizontal Direction

The equation shown in Figure 3 can be used to establish the relationship between magnitude of misalignment present and R.M.S. value of acceleration due to vibration. It is possible to find the unknown magnitude of misalignment if operating condition like speed is defined. This can be used for condition monitoring of machine which can avoid machine failure.

Simulation

The experimental conditions are simulated using ANSYS software. The shaft, coupling and disc are modeled using CAD software Creo 2.0 with the exact dimensions as used in experimental test rig. The different misalignment conditions are created. The element SOLID 186 is used for meshing. The meshed model is shown in Figure 4.

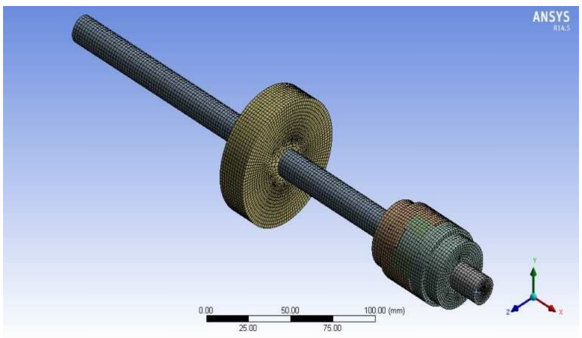

Figure 4: Meshed Model of Shaft, Disc and Coupling Assembly

The boundary conditions used are acceleration due to gravity, rotational velocity and remote displacements. The rotational velocity is 300-1800 r.p.m. as the shaft rotates in this speed range. The first and second natural frequencies are found to be 327.04 Hz and 973.19 Hz which are used to avoid resonance. The maximum operating frequency during experimentation is 30 Hz. The results obtained by simulation for parallel misalignment with elastomer as Nitrile Butadiene Rubber in horizontal direction are shown in Table 5.

Table 6: R.M.S. Values of Acceleration (μm/s^2) for Nitrile Butadiene Rubber in Horizontal Direction for Different Parallel Misalignment at Different Speeds by Simulation

Speed (R.P.M.)	Parallel Misalignment (mm)						
	0.1	0.2	0.3	0.4	0.5	0.6	0.7
600	281.43	297.49	318.04	35.59	341.07	344.46	365.08
900	524.01	540.18	551.47	563.67	569.04	579.32	594.11
1200	672.39	712.41	728.08	745.28	769.00	782.08	829.24
1800	849.42	970.76	1449.03	1800.60	1826.14	1886.41	1925.90

The result of simulation with elastomer as Hytrel and 0.2 mm parallel misalignment at speed 600 r.p.m.in horizontal direction is shown in Figure 5.

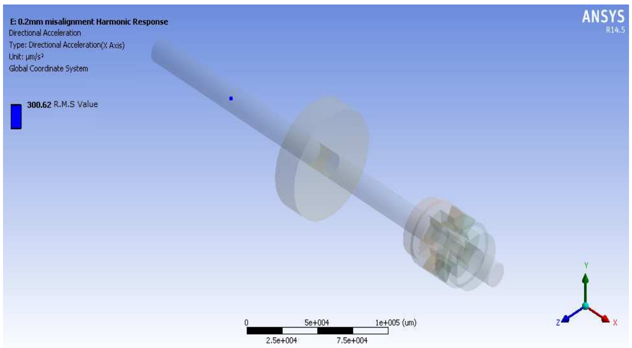

Figure 5: R.M.S Value of Acceleration for 0.2 mm Parallel Misalignment with Elastomer as Hytrel and Speed 600 r.p.m

Comparison of Table 1 and Table 5 show that the results obtained by experimentation and simulation are found to be close.

CONCLUSIONS

A test rig is fabricated for simulation of faults in coupled shafts. The faults simulated are parallel and angular misalignment. The R.M.S. values of acceleration are measured at bearing no. 2 in horizontal and vertical directions. The comparison of R.M.S. values of acceleration with speed shows that there is an increase in R.M.S. value of acceleration with speed for same amount of misalignment. Similarly there is increase in R.M.S. value of acceleration with magnitude of misalignment for same speed. MATLAB software is used to generalize the curve by curve fitting technique.

The equation of the curve shows the relationship between misalignment present and R.M.S. value of acceleration. In case of parallel misalignment, the R.M.S. value of acceleration is more dominant in horizontal direction than in vertical direction. This is because the parallel misalignment is created in horizontal plane. The experimental conditions are simulated by using ANSYS software and the experimental predictions are in good agreement with ANSYS results.

This will be useful for simulating the condition of machine without experimentation. The test rig designed in this work can be used in speed range of 300-1200 r.p.m. as the forces due to misalignment become more dominant in speed range of 1200 – 1800 r.p.m. causing more vibrations.

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