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Vibrational Response Diagnosis of Rotor System

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Abstract: Unbalance and misalignment, both are most common causes of machine vibration which reduces life of the machine. In this paper, experimental studies were performed on a rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance and misalignment. A self-designed simplified flexible coupling was used in the experiments. The rotor shaft accelerations were measured at three different speed using FFT four channel vibration analyzer (ADASH) under the unbalance and misalignment conditions. The experimental frequency spectra were also obtained for unbalanced and misalignment condition.

Keywords: Unbalance, Misalignment, Rotor system, FFT.

I. INTRODUCTION

Even in good condition, machines generate vibrations. Many such vibrations are directly linked to periodic events in the machine's operation, such as rotating shafts, meshing gear teeth, rotating electric fields, and so on.
Increased complexities of rotating machinery and demands for higher speeds and greater power have created complex vibration problems. Unbalance and misalignment is the most common cause of machine vibration.

An unbalanced rotor always cause more vibration and generates excessive force in the bearing area and reduces the life of the machine. Rotor unbalance is the most common reason in machine vibrations. Rotor unbalance is a condition in which the centre of mass of a rotating assembly, typically the shaft and its fixed components is not coincident with the centre of rotation. After unbalance, misalignment is accepted as the second most commonly observed disturbance source in rotor systems. Misalignment is categorized in two types, (1) angular misalignment and (2) parallel misalignment. Misalignment may be present because of improper machine assembly, thermal distortion and asymmetry in the applied load.
Engineering judgments based on understanding of physical phenomena are needed to provide the diagnosis and methods for correcting the rotating machinery faults. Vibration monitoring helps in identifying the early failure and hence in reducing the machine down time. The objective of vibration monitoring is to provide valuable information for the diagnosis of symptoms that help in maintenance planning.

II. LITERATURE REVIEW

The literature review has been carried-out in the areas of vibration monitoring of rotating machinery. Techniques of faults diagnosis and analysis of resulting vibration signatures have been reviewed [1]. Vibration monitoring aims to define the current condition of machine and compare it with previously measured condition. Some element of prediction is inferred by noting trends in the observed parameter. Vibration signal analysis has become an established method for monitoring the condition of rotating machines. Traditionally, the vibration monitoring of rotating machinery is heavily dependent on the spectral analysis or the Fourier transform of the vibration signals. The Simple equation of motion were studied for rotor system [5].In this paper, experimental studies were performed on a rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance[2]. In this study a model based technique for fault diagnosis of rotor–bearing system is described. It has been found that the misalignment couples vibrations in bending, longitudinal and torsional modes. Some diagnostic features in the fast Fourier transform (FFT) of torsional and longitudinal response related to parallel and angular misalignment have been revealed [3][4].

III. THEORY

A. System Model

For a symmetrical single-disk flexible rotor-bearing system as shown in Fig. 1

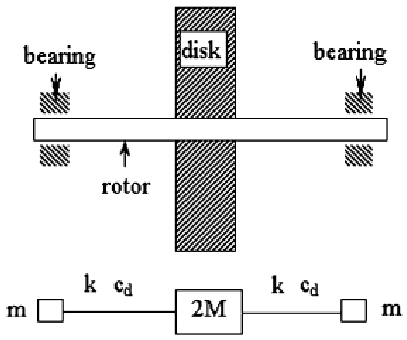

Fig 1 Rotor Bearing System

System motion equations can be written as follows if the lump mass method is adopted

$$
\begin{aligned}
m\ddot{x}_1 &= k(x_2 - x_1) + c_d(\dot{x}_2 - \dot{x}_1)m\ddot{y}_1 \\
&= k(y_2 - y_1) + c_d(\dot{y}_2 - \dot{y}_1) + mgM\ddot{x}_2 \\
&= k(x_1 - x_2) + c_d(\dot{x}_1 - \dot{x}_2) \\
&+ Me\omega^2 \sin(\omega t + \varphi_0)\, M\ddot{y}_2 \\
&= k(y_1 - y_2) + c_d(\dot{y}_1 - \dot{y}_2) + Mg \\
&+ Me\omega^2 \cos(\omega t + \varphi_0)
\end{aligned}
$$

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B. Line Diagram of Experimental Setup
The line diagram of experimental test rig is shown in fig 2. The test rig consists of a shaft with central rotor disc, which is supported on two ball bearings. A dc motor coupled by a flexible coupling drives the shaft. The parameters used in studies are given in table 1.

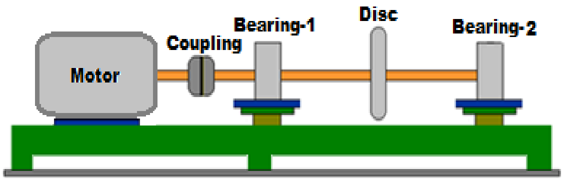

Fig 2 Line Diagram of Rotor Rig

Table No.1 List of Designed Parts of System

Shaft	
Diameter ,d	20 mm
Length L	500 mm
Material	Mild steel
Density of Material	7860 Kg/m^3
Young Modulus	$2.1\ X10^{11}\ N/m^2$
Key	5 X 5 X 30
Hub	
Outside diameter of hub, D	40 mm
Length of hub, l_h	30 mm
Pitch circle diameter for pin	60 mm
Bearing	6004
Disc	
Diameter	128mm
Thickness	12mm
Material	Mild steel
Density of Material	7860 Kg/m^3

C. Experimental Setup
The actual rotor rig is also shown in fig 3 this rig consist of a shaft of 20 mm diameter and 500 mm length mounted in between the two SKF single row groove ball bearing. A disc of 128 mm diameter and 12 mm thick is fixed at the mid span of the shaft. The disc consists of 4 drilled holes on the periphery at a radius of 40 mm to fix the balancing mass. The rotor rig is driven by 0.5 HP motor running at a speed of 2800 RPM. A coupling connects the driver and driven shaft.

Fig 3 Actual Experimental Set Up

The measuring instruments used in the experimental set-up. An accelerometer is mounted with a stud on the ball bearing housing. The output of the accelerometer is connected to the FFT.

D. Experiment Procedure
Experimental facility as shown in Figure 4 is used for unbalance test. First the setup is run for few minutes to settle down all minor vibrations. Before creating the unbalanced, the shaft is checked for any misalignment and unbalance. Two dial gauge method is used to check the proper alignment and balancing. After this an unbalance has been created by placing a mass of 13 gram in the rotor at a radius of 40mm. Accelerometer along with the FFT analyzer is used to acquire the vibration signals.Ch no. 1 of FFT is connected to vertical direction on drive end motor ch no 2 of FFT is connected to horizontal direction on the same drive end. B&K 4384 charge-type accelerometers are used. Rotor speed is also measured with a B&K photoelectric tachometer probe. The Calibrated accelerometer is fitted over the motor and connected to the FFT. Vibration signals are measured at three different speeds 1000, 1500 and 2000 rpm with the well.

Fig 4 Experimental test rig with FFT connection

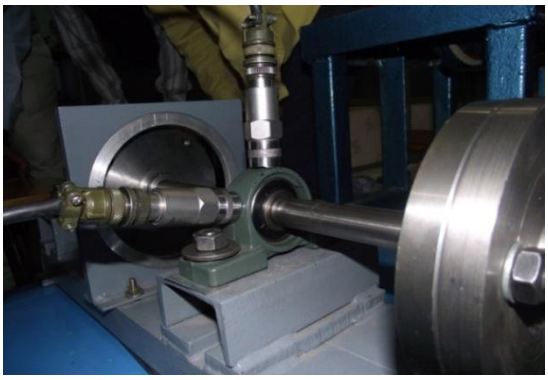

Fig 5 Accelerometer Placed on Bearing, FFT is connected to vertical and horizontal direction

IV. RESULT AND DISCUSSION

A. Frequency Spectrum of Base Condition
The experimental frequency spectra were obtained to baseline condition. The perfect aligned and balance cannot be achieved in practice.

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Thus, a baseline case is presented first to show the residual unbalance. The measured amplitude of a well balanced system at drive end (DE) with the self designed 3 pin type coupling at three speeds 16.67 Hz, 25 Hz and 33.333 Hz (1000, 1500, 2000 rpm) shown in Figure 3,4and 5,6 on CH.NO.1 and CH.NO.2 respectively. The base line spectrum is measured experimentally using four channel FFT analyser.

Table No.2 Values of overall Acceleration vs. Frequency

Speed RPM	Acceleration vs. Frequency					
	Vertical direction (Ch No.1)			Axial direction (Ch No.2)		
	1X	2X	3X	1X	2X	3X
1000	0.01	0.03	0.04	0.01	0.02	0.04
1500	0.05	0.07	0.08	0.02	0.04	0.06
2000	0.05	0.08	0.10	0.04	0.05	0.08

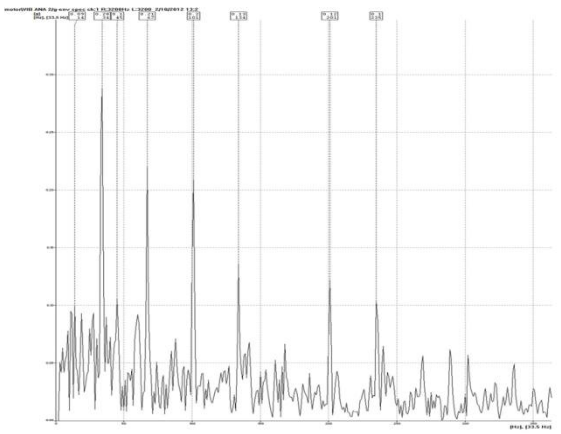

Fig 6 Graph of overall acceleration vs. Frequency of healthy system for Vertical Direction

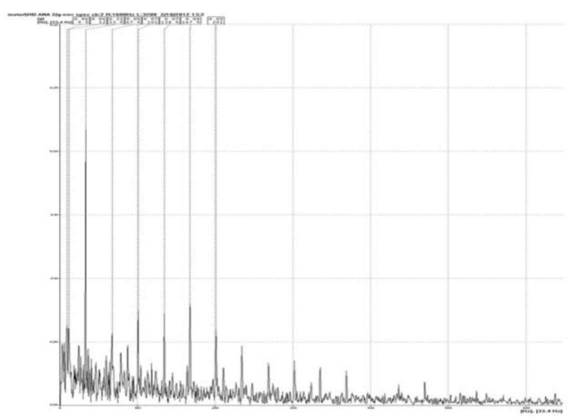

Fig 7 Graph of overall acceleration vs. Frequency of healthy system for Horizontal Direction

In a healthy system the FFT spectrum displays a single 1X frequency peak. As the speed increases the acceleration at 1X is also increases.

B. Shaft alignment

Shaft alignment is the process to align two or more shafts with each other to within a tolerated margin. Any misalignment between the two increases the stress on the shafts and will almost certainly result in excessive wear and premature breakdown of the equipment. This can be very costly. When the equipment is down, production might be down.

1. Parallel Misalignment

Parallel misalignment with, the center lines of both shafts are parallel but they are offset as shown in fig 8 The parallel misalignment can be further divided up in horizontal and vertical misalignment. Horizontal misalignment is misalignment of the shafts in the horizontal plane and vertical misalignment is misalignment of the shafts in the vertical plane.

Parallel horizontal misalignment is where the motor shaft is moved horizontally away from the pump shaft, but both shafts are still in the same horizontal plane and parallel.

Parallel vertical misalignment is where the motor shaft is moved vertically away from the pump shaft, but both shafts are still in the same vertical plane and parallel.

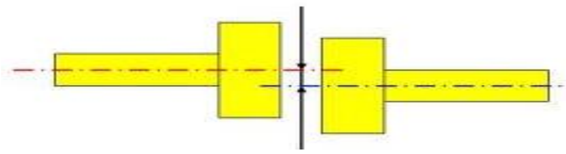

Fig 8 Parallel Misalignment

The shaft is running at speed 1000 RPM and the parallel misalignment at 0.2, 0.4 and 0.6 mm.

Table No.3 Values of overall amplitude in vertical and axial directions of different values of different parallel misalignments

Sr. No	Parallel misalign ment (mm)	Speed (RPM)	Experimental value (mm/s)					
			Vertical direction (Ch.No.1)			Axial direction (Ch.No.2)		
			1X	2X	3X	1X	2X	3X
1	0.2	1000	0.95	0.34	0.45	1.6	0.35	0.55
2	0.4	1000	0.31	0.37	0.63	0.19	0.49	0.36
3	0.6	1000	0.8	0.6	11	5	0.8	0.9

Experimental analysis can be used in predictive maintenance to monitor the parallel misalignment conditions. FFT gives more predictable results for parallel misalignment fault.

2. Angular Misalignment

The angular misalignments are created moving the bearing 2 and keeping the bearing 1 fixed. The angular misalignments are measured with the following formula and with help of Fig 9. This gives angular misalignment in degrees.

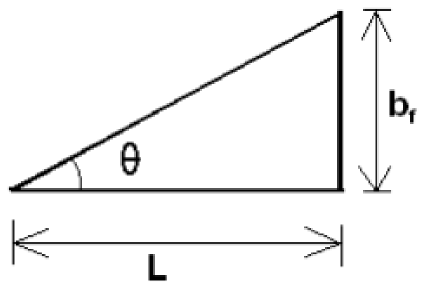

Fig 9 Measurement of Angular Misalignment

$$\theta = tan\frac{b_f}{L}$$

Where b_f =Distance move by bearing 2
L=Length of the shaft
θ=Angular misalignment in degree

The different values of angular misalignment ranging from 0 degree to 1 degree were created. The rotor was run at different speed. In angular misalignment the overall accelerations have used for classifying the faults which is recorded with FFT.The shaft is running at speed 1000 RPMand the Angular misalignment at 0.26^0, 0.34^0 and 0.43^0

Table No.4 Values of overall acceleration in vertical and axial directions of different values of different angular misalignments

Angular Misalignment in degree	Speed(RPM)	Acceleration mm/s2					
		Vertical direction (Ch No.1)			Axial direction (Ch No.2)		
		1X	2X	3X	1X	2X	3X
0.26	1000	0.02	0.05	0.03	0.07	0.12	0.06
0.34	1000	0.02	0.03	0.06	0.02	0.05	0.04
0.43	1000	0.02	0.05	0.04	0.03	0.05	0.025

C. Unbalance

First the setup is run for few minutes to settle down all minor vibrations. Before creating the unbalanced, the shaft is checked for any misalignment and unbalance. Two dial gauge method is used to check the proper alignment and balancing.

Table No.5 Values of overall amplitude in vertical and horizontal directions of unbalanced mass 12 and 24 grm at a Radius of 40 mm in Rotor at different speed

Unbalances mass grams	Speed (RPM)	Amplitude mm/s	
		Vertical 1X (Ch No.1)	Horizontal 1X (Ch No.2)
12	1000	0.30	0.8
12	1500	0.20	10
12	2000	0.80	35
24	800	2.8	0.6
24	1000	10	10.5
24	1500	11.5	22
24	2000	14	33

In an unbalance fault the FFT spectrum displays a single 1×rpm frequency peak. This defect may cause high horizontal direction than vertical direction. From that table if increase the unbalance mass the amplitude at 1X is also increases FFT gives more predictable results for unbalance fault.

V. CONCLUSION

In case of parallel and angular misalignments the amplitudes are measured at drive end in vertical and Axial Directions. The highest value of overall amplitude is produced in axial direction. It is observed that the amplitudes are higher in axial direction in the drive end and increased when misalignments are increased.

In case of unbalance, the amplitudes are measured at drive end in vertical and horizontal Directions. The highest value of overall amplitude is produced in horizontal direction. It has been found that the amplitudes are higher in horizontal direction in the drive end and increased when unbalance increased.

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