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Experimental Study of Effect of Eccentricity on Vibration of Shafts

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Abstract

Mechanical systems such as motors, pumps, engines, and turbines operate on shafts which are rotating at different speed. Due to unexpected operating conditions, various faults such as cross-sectional cracks, looseness and misalignment may occur during their service life. Eccentricity plays a vital role in predicting the operating life of the shafts. Eccentricity is one of the factors that is likely to cause whirling of the shafts thereby subjecting them to cyclic bending stresses resulting in premature failure of the shafts. In this paper an attempt has been made to experimentally study the effect of eccentricity in a rotary system and predict the vibration spectrum for shaft with concentric and eccentric rotation using Lab VIEW software.

Key Words: vibration analysis; eccentricity, condition monitoring, Lab VIEW

I. Introduction

Misalignment is the second most common malfunction after unbalance. Misalignment may be present because of improper machine assembly [1], thermal distortion [2] and asymmetry in the applied load [3]. Shaft misalignment is a condition in which the shafts of the driving and driven machines are eccentric because of improper machine assembly. Perfect alignment of the driving and driven shafts cannot be achieved in practical applications. Even if perfect alignment is achieved initially, it would not be possible to maintain it during operation of the machine due to various factors. Hence, the misalignment condition is always observed in machines. LabVIEW is a virtual instrument software platform developed by National Instrument (NI); a development tool for creating applications using icon code instead of text programming language. The data acquisition system designed with LabVIEW can simulate various signal acquisitions, and brings great convenience to the development of the measurement and control system [7]. Shaft eccentricity promotes mixing by creating a pair of counter-rotating vortices, enabling a more effective interaction between the inner and near-wall particles of the shaft material [8,10].A larger magnitude of one gear's eccentricity can excite a larger magnitude of the standard vibration and dynamic coupling of that gear and thus on shaft. [9]. Vibration monitoring is a useful technique applicable to rotating machines and provides valuable information regarding symptoms of machinery failures which in practice may avoid costly breakdowns. The present work focuses on the study of vibrations produced by parallel misalignment of rotating shaft in horizontal direction. It also involves fabrication of a rotor bearing system to investigate effect of misalignments on rotating shafts. The objective of this work is to construct a small rotor test rig supported in conventional ball bearings that could be used to simulate and determine effect of misalignments of the vibration of rotor bearing system. Specifically parallel misalignment in horizontal direction is the targeted fault under investigation at the support. RMS acceleration has been carried out for the vibration signatures due to parallel horizontal misalignment condition. The overall magnitude of vibration in RMS acceleration for various misalignments conditions such as parallel and angular or the combination of both is higher at the housing away from the motor and lesser towards the housing near the motor. It is also found that the values of acceleration increases with the increase in misalignment [4]. The temperature monitoring technique can be used to monitor machine condition monitoring. This non-contact method will aid fault detection at critical points. . Through the creation of misalignment, a temperature rise at the coupling mid-point was successfully detected using the technique. This analysis shows that misalignment is a direct cause of temperature change. A maximum temperature increase of approximately 5% was found from the varying experiments and above 10% in the increasing experiment. . These affect the couplings directly through fatigue and indirectly as a result of a loss of output. These are key focuses in terms of improving maintenance and reliability in a rapidly developing industry. Both the technique and investigation require further development and research [5]. Following design cues for designing a robust rotor shaft configuration which includes options to address fundamental geometry driven stress concentration factors, fretting driven fatigue stress concentration effects and vibration natural frequency: [6]

- Eliminating/removing the stress concentration area core pack location shoulder.
- Increasing the interference fit between the core pack and the rotor shaft in order to reduce the relative local movements and hence, reducing/eliminating fretting fatigue
- Selection of an optimum bearing pre-load to reduce the vibration g-load or the order of the Q factor.
- Shot Peen the shaft at the area of the interference fit with the corepack in order to introduce a surface compression stress. This will lead into an improvement of the fatigue life of the shaft as well as reduce the relative movements of the corepack.

II. Methodology

The test rig consists of a metallic shaft, which is supported by a ball bearings at driving and driven ends of the shaft and is driven by an AC motor. An accelerometer is mounted on the ball bearing housing at the driven end. The output of the accelerometer is connected to a data acquisition system to acquire the signal and to carry out the frequency analysis using LabVIEW software. The parameters used in studies for mild steel shaft having material density and young's modulus 7860 Kg/m^3 and 2.1X10^{11} N/mm^{2}respectively are 25mm diameter and 300mm length. Similarly the parameters for SKF-6025 ball bearing are 25 mm inner diameter, 45mm outer diameter and 14mm thickness. The actual rotor rig is also shown in figure 1. This test rig consist of a shaft of 25 mm diameter and 300 mm length mounted to SKF single row groove ball bearing. The rotor rig is driven by 1 HP motor running at a speed of 1440 RPM. A collar connects the driver and driven shaft.

Fig. 1 The Test Rig to measure the vibration signature of the shaft mounted.

1. Creation of Misalignment Conditions

Two collars have been fabricated and mounted at both driven and driving end of the shaft. These collars have screw arrangement to create the eccentricity. The parallel misalignments have been created by moving both collars 1 and 2 simultaneously ranging from 0 mm to 2.5 mm in step of 0.5 mm. The dial indicator having least count 0.01 is used for measurement of eccentricity of the shaft.

2. Acquiring the Vibration Data

The vibration data is acquired using the accelerometer mounted onto the bearing housing on the driven end of the shaft. The accelerometer acquires the analogue acceleration due to vibration and sends it to DAQ (Data Acquisition) system which converts the analogue data into an electronic digital data. This electronic digital data is read by the LabVIEW software in which graphical programing for vibration data analysis is done. The output displayed by the software in the form of graphs can be used to analyse the vibration pattern.

III. Results and Discussions

The maximum and minimum vibration amplitudes for eccentricity ranging from 0 mm to 2.5 mm are measured. The following results portray the maximum and vibration amplitudes for each level of eccentricity

1. For eccentricity of e = 0 mm

Table 2 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM.

Table.2 Maximum and minimum amplitudes of vibration for eccentricity e=0mm

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.004722	-0.004574	0.004786	-0.004584
0.004725	-0.004596		
0.004645	-0.004583		
0.004758	-0.004572		
0.004793	-0.004599		

Figure 2 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

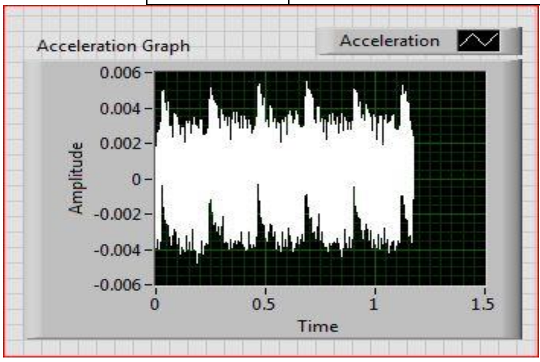
2. For eccentricity of e = 0.5 mm

Table 3 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM. It is observed that the average acceleration is slightly more than for the shaft rotating with eccentricity of 0 mm. Vibration increased for slight increase in the eccentricity.

Table 3Maximum and minimum amplitudes of vibrationfor eccentricity e=0.5mm

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.004815	-0.00454	0.0048612	-0.004613
0.004874	-0.004688		
0.004867	-0.004615		

0.00485	-0.004613		
0.00490	-0.00461		



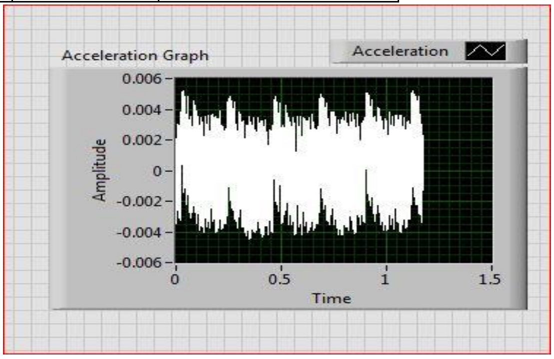


Fig 2. vibration amplitudes for e=0mm for set 1, Figure3 data in LabVIEW software for set 1 out of 5 sets of readings.

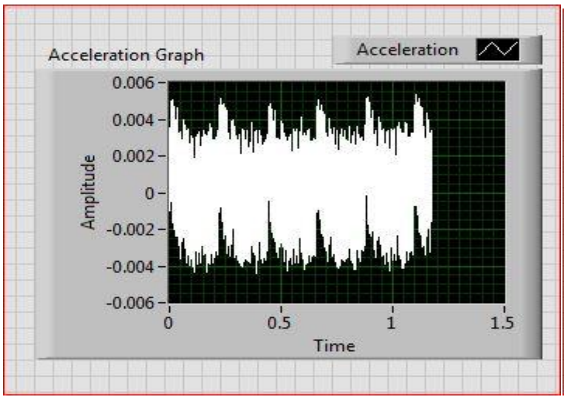
Figure3 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

3. For eccentricity of e = 1 mm

Table 4 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM.

Table 4Maximum and minimum amplitudes of vibration for eccentricity e= 1 mm

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.004926	-0.00452	0.004960	-0.004622
0.004945	-0.00453		
0.004946	-0.00466		
0.004987	-0.00467		
0.005040	-0.00473		



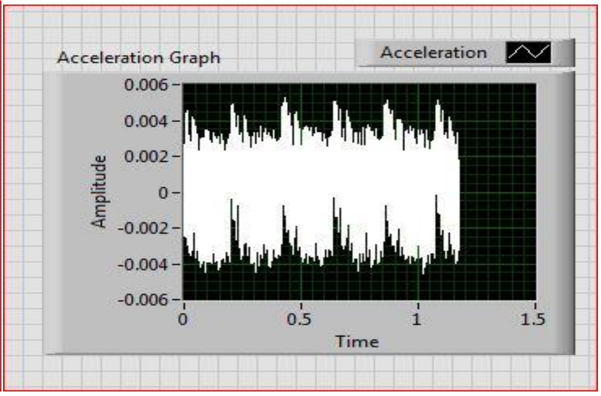


Fig 4. Vibration amplitudes for e=1 mm for set 1, Fig 5. Vibration amplitudes for e=1.5 mm for set 1

Figure 4 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

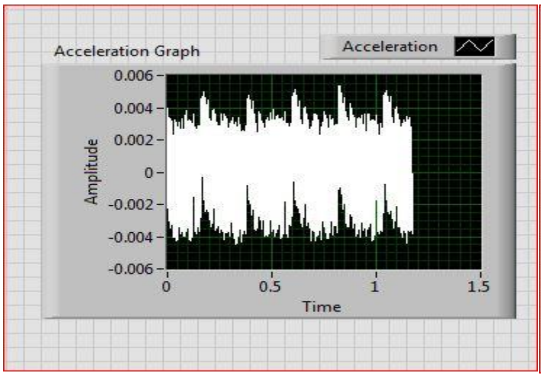
4. For eccentricity of e = 1.5 mm

Table 5 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM.

Figure 5 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

Table 5Maximum and minimum amplitudes of vibration for eccentricity e=1.5mm

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.004889	-0.00467	0.0049818	-0.004606
0.005012	-0.00463		
0.004983	-0.00462		
0.004957	-0.00452		
0.005073	-0.00459		



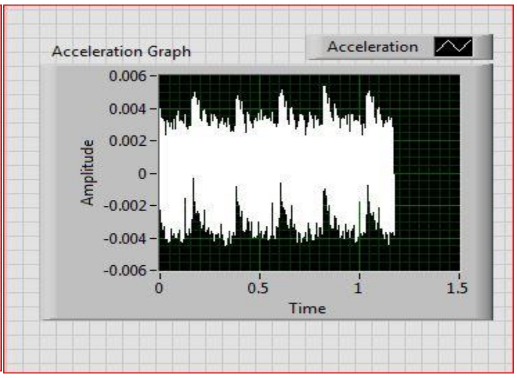


Fig 6. Vibration amplitudes for e = 2 mm for set 1, Fig 7. Vibration amplitudes for e = 2.5 mm for set 1

5. For eccentricity of e = 2 mm

Table 6 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM.

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.005043	-0.00454	0.0050784	-0.0046208
0.005096	-0.004674		
0.005005	-0.00468		
0.005186	-0.00461		
0.005062	-0.00460		

Table 6Maximum and minimum amplitudes of vibration for eccentricity e=2 mm

Figure 6 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

6. For eccentricity of e = 2.5 mm. Table 7 shows the maximum and minimum amplitudes of vibration for the rotor speed of 1440 RPM.

Table 7Maximum and minimum amplitudes of vibration for eccentricity e=2.5 mm

Acceleration in m/s^2		Average acceleration in m/s^2	
Max	Min	Max	Min
0.005010	-0.00467	0.005144	-0.004636
0.005137	-0.00462		
0.005187	-0.00455		
0.005183	-0.00469		
0.005203	-0.00465		

Figure 7 represents the output graph of the above data in LabVIEW software for set 1 out of 5 sets of readings.

IV. Conclusion

The experiment was performed for different eccentricities varying from 0 to 2.5mm and it is clearly observed that the average maximum and minimum acceleration goes on increasing with increase in the eccentricity. For every 0.5 mm of eccentricity the average amplitude of vibration was observed and it can be concluded that the vibration increases by 1.3%. This clearly emphasizes that the shaft whirls due to eccentricity. More the eccentricity more is the whirling of the shaft. Higher the shaft whirls more is the repeated bending stresses acting on the shaft and thereby causing premature failure of shaft due to fatigue.

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