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Simulation and Experimental Study to Investigate the Effect of Bent Shaft on Vibration Spectrum

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Abstract:
Rotating system is the backbone of machinery. With the high speed demand of today's machinery, it becomes more important than ever. There are many causes of bending of shaft. Shafts may bend during transportation, installation and also due to excessive heat. Bending of shaft results in high vibration which creates stress on bearings and couplings. If such fault is ignored, catastrophic failure will result. Therefore it is important to be able to diagnose this fault. Bent in the shaft can be identified by studying the vibration spectrum on the bearings. Researchers have reported that the dominant peak is observed normally at 1X if the bend is near the center of the shaft and at 2X if the bend closer to the coupling end. This data can be misinterpreted for misalignment but bent shaft frequency component is 1800 out of phase. The present study deals with the experimental investigation of detecting unique vibration signature for bent shaft having bent at coupling end. Experimental studies are performed on a rotor system test apparatus to predict the vibration spectrum for shaft bending. The rotor shaft vibrations are measured by using FFT analyzer. These acquired spectrums are compared with results obtained from simulation study by using ANSYS for confirmation of experiment results and these results are verified as reported in the literature.

Keywords: *ANSYS©, Bent Shaf, FFT analyzer, flexible flange coupling, vibration spectrum*

1. Introduction

Rotating machines are used to transform electrical energy to mechanical energy and vice versa in case of turbines. As the bearings are integrated part of rotating machinery, research conclusions show that the faults related to bearing are the commonly tackled machine faults, leading to 40% of total machine failures. Considering the importance of the bending in the shaft, detecting and diagnosing the bent is still elusive. The main causes of mechanical vibration are bent rotor shaft, unbalance, misalignment, looseness and distortion, defective bearings and so on. These are some of the most common faults that can be detected using vibration analysis.

Rotor shaft bending is a common problem in the operation of rotating machinery. Sometimes a shaft with high length to diameter ratio may develop a bent by itself at rest. Due to current trends in the design of rotating machinery towards higher speeds, manufacturers are tending to produce machines which operate closer to lateral critical speeds than has previously been necessary. The effect of bent shaft on the critical speeds and its vibration amplitudes have become an important consideration in diagnostic maintenance of a rotor bearing systems. Perfect alignment of the driving and driven shafts cannot be achieved in practical applications, thus a bent and misalignment condition is virtually always present in the machine. Identification of all above fault in machine is very important to avoid catastrophic failure. Vibration analysis is an important tool for identification of these defects. These aspects motivated in the present study to explore and confirm the effect of bent shaft on the vibration spectrum.

As per Mobius institute, The shaft may bend due to excessive heat, due its length, or it may be physically bent. A bent shaft predominantly causes high 1X axial vibration. The dominant vibration is normally at 1X and if the bend is near the center of the shaft, however 2 X vibrations occur if the bend is closer to the coupling. Vertical and horizontal axis measurements will also often reveal peaks at 1X and 2X, however the key is the axial measurement. Phase is also a good test used to diagnose a bent shaft. The phase at 1X measured in the axial directions at opposite ends of the component will be 180° out of phase [1].Suri Ganeriwala studied bearing vibration and reaction force signatures caused by bent shafts were studied experimentally. The vibration and force spectrum signatures between baseline and bent shafts (center bent and coupling end bent) under two shaft speeds, 1000-rpm and 5000-rpm, were compared. The experiment results indicate that a center bent shaft will increase the bearing vibration and the vibration amplitude shows dominant peak at 1X of the shaft speed. However, a coupling end bent shaft gives peaks at 1X and 2X of the shaft speed. 2X amplitude is more than amplitude at 1X in case of coupling end bent [2].

V. Hariharan, P.S.S.Srinivasan studied the effect of parallel misalignment on a rotor shaft with a rigid as well as flexible coupling. An experimental setup with rigid and flexible coupling is designed and manufactured. The setup is modeled in ANSYS and simulated. A parallel misalignment of 0.2mm is created in the set up with rigid coupling and the frequency spectrum is acquired. A rigid coupling is changed by a flexible coupling and the same tests are carried out. These tests are simulated in ANSYS and results are compared which found to be in good agreement. They concluded that by using flexible coupling vibrations are reduced by 85-89% [3]. It is revealed from the literature review that it is important to diagnose the fault at an earlier stage, so that the machine life can be enhanced with less cost. It is clearly proven that bent produces a high vibration level in bearings. Therefore, it is proposed to investigate experimentally the effect of bent shaft on frequency spectrum using flexible flange coupling and validate the typical spectrum reported in literature.

2. Experimental Setup

The experimental setup is shown in Figure 1. The experimental setup consists of a 0.25 HP D.C. motor with extended shaft, a flexible flange coupling, and three identical self aligned ball bearing. The driven shaft is supported by two bearing and has a length of 430 mm with a bearing span of 300 mm. The diameter of shaft is 15.875mm.

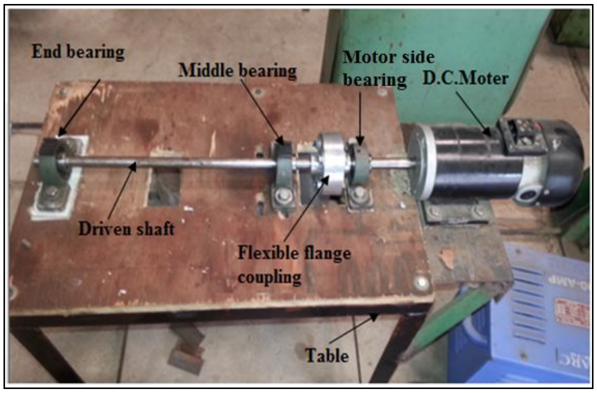

Figure1: The experimental setup

Pedestal bearings are used in setup for supporting motor shaft. A four pin type flexible flange coupling is used to connect motor shaft and driven shaft. The bearing which is near to the coupling on driven side is called as middle bearing. Bearing fitted at the end of driven shaft is called as end bearing. Two shafts were tested in the experiment. Initially a straight shaft without bend is used for baseline condition. Later it is replaced by a shaft which has a bent at coupling end. A piezoelectric, Triaxial, shear type accelerometer (Type AC102-A, Sl. No 66760) is used along with the Photon+ (Bruel and Kajer make) ultra portable dynamic signal analyzer. This is a multichannel data recorder. The data acquired by Photon+ data recorder is processed with FFT software and then the measured vibration data are collected at a computer terminal through RT-PHOTON+ interface. Vibrations are measured in frequency domain. Therefore output display in terms of frequency vs amplitude graph is selected in RT-PHOTON software. Vibration amplitude is measured in terms of displacement. For measuring 1600 spectral lines, frequency band of 0-40 Hz. The vibration sensor is mounted on top of bearings for acquiring signal. A speed regulator is used to vary the motor speed.

3. Experimental Procedure

Initially the rotor system is checked for alignment. To do this, the dial gauge method is used to make perfect alignment. Also, the surface level is checked by using spirit level. The setup is allowed run using shaft without bent for a few minutes to allow all minor vibrations to settle. Then, the accelerometer is fitted on the bearing housing and connected with the FFT analyzer. Next, the vibration data measured by using FFT analyzer and saved in a computer. A typical vibration spectrum is acquired on the middle bearings for three different speeds viz. 300rpm, 600rpm and 900rpm to study the behavior of the vibration frequency spectrum at different speeds and to check whether the unique signature of vibration on frequency spectrum is dependant of speed or not. Later the driven shaft is replaced by a bent shaft and the same experimental procedure is followed and vibration spectrum is acquired on middle bearing for bent shaft.

4. Simulation Study

Rotor shaft, Virtual bearings and coupling are modeled using Autodesk Inventor 2014 with the exact dimensions as used in the experimental setup. Two different models are prepared for baseline condition and bent driven shaft having bent near coupling end. Figure 2 shows solid model of the rotor shaft assembly for bent shaft. First of all, the parts are created using part drawing files and materials are specified for each part. As the two flanges are made up of aluminum, the same material is selected for flanges. Both the shafts are created of mild steel. The driven shaft is created with a bend at one end where the coupling is attached which is 0.8^0as the actual shaft has. Rubber bushes are given material as natural rubber. Shafts are coupled with coupling by using rectangular key of mild steel. Nuts and bolts are standard components and are available in the content center library. Therefore those are called from the library in the assembly file. Hexagonal metric Nuts and Bolts of size M8 are selected from the content center. The shaft and coupling assembly is constrained in such way that there is only rotational motion possible about z axis for shafts and coupling. All the translation motions are constrained.

Figure 2: Solid model of flexible flange coupling and rotor shaft assembly for bent driven shaft at coupling end

The models created in Autodesk Inventor are saved in IGES format and then imported to ANSYS-12 software. Using ANSYS meshing, analysis is carried out. The material properties used are listed in Table 1. The material used for shaft is mild steel and nut bolts is mild steel. For flange coupling aluminum material is used.

Material	**Aluminium**	**Mild steel**	**Rubber**
Young's Modulus (Mpa)	$0.675 * 10^5$	$2 * 10^5$	30
Poison's ratio	0.334	0.3	0.49
Density (kg/mm^3)	$2700*10^{-9}$	$7850*10^{-9}$	$1140*10^{-9}$

Table 1: Material Properties

The material property of rubber is initially defined as an isotropic material with Young's modulus and Poisson's ratio values. In this stage, the rubber acts as a linear material. To convert it into non-linear material, hyper elastic property with Mooney Rivlin constants are introduced [3]. Maximum nine Mooney Rivlin constants are available. In this analysis, all the nine constants are used for better accuracy. The Mooney Rivlin constants used in the present study are represented in Table 2. These constants account for non-linear property of the natural rubber[3]. The surface to surface contact is considered between the rubber and aluminum flanges.

C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_9
58.66	0.774	54.26	-117.49	52.77	3.58	-23.067	33.69	-12.486

Table 2: Mooney Rivilin Constants for Accounting Rubber Non Linearity [3]

In the present model, the element type of SOLID 95 is used. Smart element size control is used for mapped mesh. SOLID 95 is a higher order version of the 3D 8-noded solid element. It can tolerate irregular shapes without the loss of accuracy. In fact, SOLID 95 elements have compatible displacement shapes and are well suited to model curved boundaries. These elements can also be tetrahedral and can automatically transition between hexahedral and tetrahedral using pyramids. Other solid elements like SOLID 92 and SOLID 186 are higher versions with good accuracy are available but consumes more memory location therefore requires high hardware configuration computer. So compromising between accuracy and hardware requirements SOLID 95 element is chosen. The meshed model is presented in Figure 3.

Figure 3: Meshed model of shaft coupling assembly

The rotor shaft is supported between two identical ball bearings of 197 mm span on non-drive end and one bearing on the drive end. The bearing P 204 type is represented by COMBIN 40 element. COMBIN40 is a combination of a spring-slider and damper in parallel, coupled to a gap in series. A mass can be associated with one or both nodal points. The element has one degree of freedom at each node, either a nodal translation, rotation, pressure, or temperature. The mass, springs, slider, damper, and/or the gap may be removed from the element. The element may be used in any analysis. The stiffness of the bearing is 1.5 x 10^4 N/mm. The rotor shaft model rotates with respect to global Cartesian Z-axis. The angular velocity is applied with respect to Z-axis. The degree of freedoms along UY, UZ and ROTX, ROTZ are used at bearing ends. Since there is a gap between rubber bushes and bolts, it is necessary to create contact between those surfaces. Therefore CONTA 174 element is used as contact element for outer surface of bolts and TARGE170 is used as target element for concave surface of rubber bushes. Angular velocities of 5Hz, 10 Hz and 15 Hz are given as input and corresponding displacements are measured at both the drive end and the non-drive end. The static loads at bearing locations are calculated. The frequency range is give from 0 Hz to 20 Hz with a sub step of 2.5 Hz. From time history postprocessor X component displacement vs frequency at each bearing node is obtained. Figure 4 shows ANSYS spectra for bent shaft at middle bearing for 5Hz which shows dominant peak at 15 Hz which is 2X of the shaft fundamental frequency ie at 900 rpm.

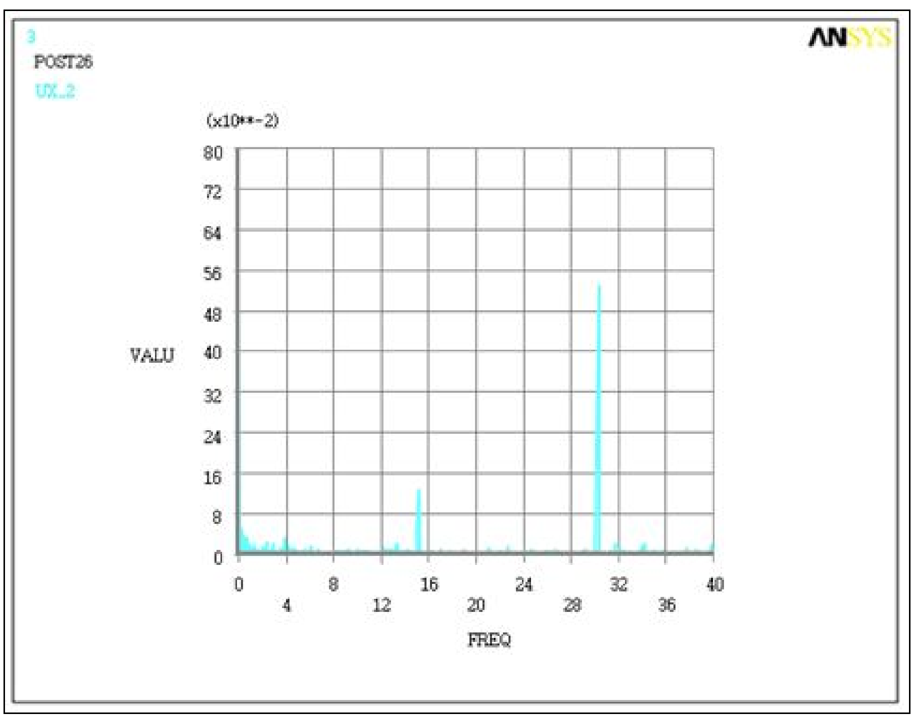

Figure 4: Simulation result for bent shaft on middle bearing at 15 Hz

5. Results and Discussions

Experimental and simulation study has been carried for investigating the effect of shaft bending on vibration spectrum. The results of the experimental study and simulation study for baseline condition and bending are also compared.

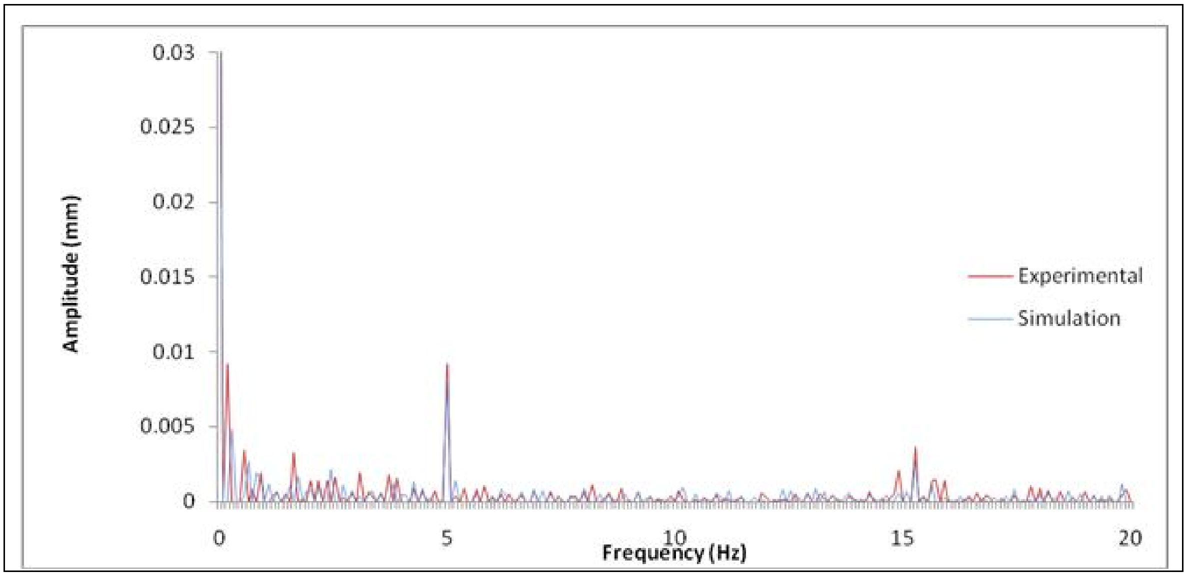

Figure 5: Comparison of simulation and experimental study results for baseline condition on middle bearing at 5 Hz

The vibration spectrum acquired on middle bearing at a frequency of 5 Hz for aligned condition is shown in figure 5. It shows the comparison of results obtained from simulation and experimental study. At 300 rpm the maximum vibration amplitudes observed are 0.015mm and 0.012mm respectively from experimental and simulation results. As observed from above graph the highest amplitude is seen at the fundamental frequency but the amplitudes are not matching. In simulation results it is seen that the amplitude is less as compared with experimental results. The simulation and experimental results are in close agreement and are as reported in literature. It confirms that for a flexible flange coupling for aligned condition peak amplitude of vibration is at 1X the shaft speed.

Figure 6 depicts comparison of bent shaft experimental as well as simulation study results on middle bearing at 5 Hz,. From figure 6 it is observed that the dominant peak for bent shaft occurs at 1X and 2X the shaft speed.

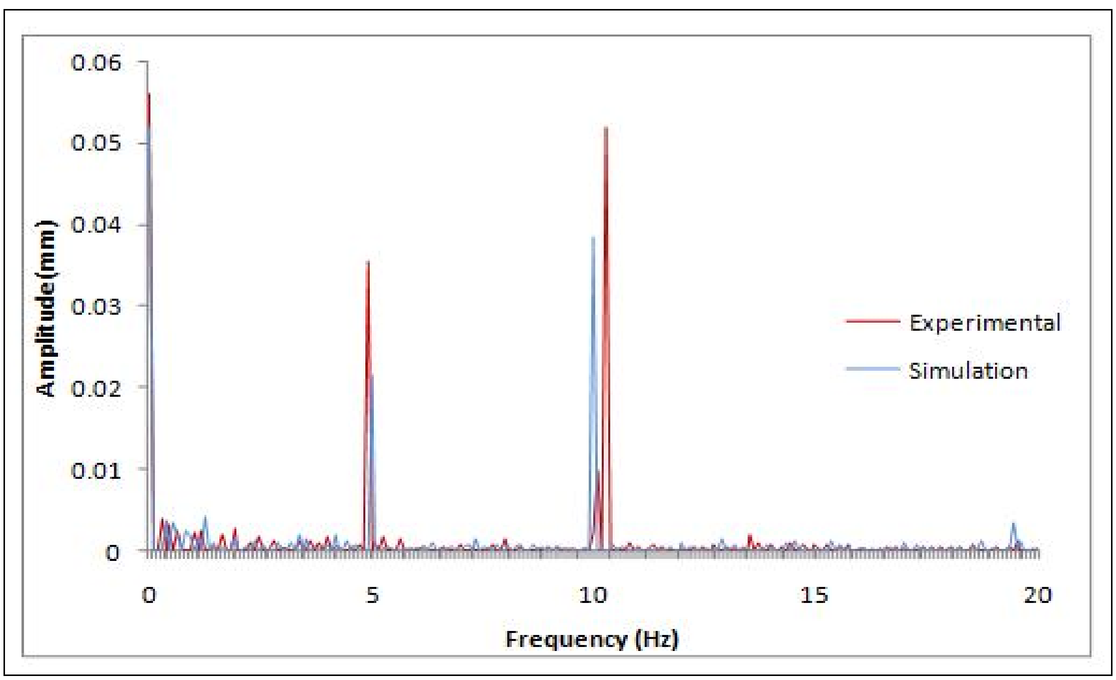

Figure 6: Comparison of baseline and bent shaft experimental study results on middle bearing at 5 Hz

6. Conclusion

The shaft with bent near coupling end is simulated and studied using the both experimental investigation and simulation. The experimental and simulated frequency spectra are obtained and found to be similar. The experimental predictions are in good agreement with the ANSYS results. Both the experiment and simulation results prove that bent can be characterized primarily by second harmonics ie. 2X of shaft running speed in case of bent at coupling end and it is independent of speed. It shows high amplitude at 1X of the shaft speed but it is less than amplitude seen at 2X of the shaft speed in both experimental as well simulation study. These results are similar as reported in literature. Also the simulated model is valid for baseline condition and bent shaft.

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