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IDENTIFICATION OF DYNAMICS CHARACTERISTICS IN A ROTOR BEARING SYSTEM

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Abstract

Vibrations are found almost everywhere in rotating machines. The most common rotating machinery fault is mass unbalance, which is caused by uneven distribution of the mass. In this study, dynamics characteristics of rotor bearing system has been investigated and verified by numerical simulation and experimental measurements. A simple rotor model is considered in the present work. The purposes of this paper are to identify the natural frequencies of rotor bearing system and study the vibration responses due to rotor unbalance. This numerical simulation is done using COMSOL Multiphysics FEM Software Package. The simulation results are verified by rotor bearing test rig for unbalance and residual unbalance conditions.

Keywords: rotor bearing system, dynamics characteristics, vibration responses, FEM, unbalance

Introduction

Rotor unbalance is the most common reason in machine vibrations. Most of the rotating machinery problem can be solved by using the rotor balancing and misalignment. Before detecting the unbalance, the most important fact is to identify the dynamic characteristics of the rotor bearing system. The dynamic characteristics are natural frequency and mode shape (whirling of shaft). A natural frequency is the frequency at which the structure would oscillate if it were disturbed from its rest position and then allowed to vibrate freely. A mode shape is a specific pattern of vibration executed by a mechanical system at a specific frequency. Akash Rajan et al. (2014) investigated dynamic unbalance detection in rotating machinery. These analyses were done without any rotation and with rotation for Jeffcott rotor model. And then, the offset disc and shaft with two rotors were performed for eigenfrequency analysis. Tamrakar and Mittal (2016) discussed about the vibration response of crack rotor with the help of Campbell diagram. Zang et al. (2017) presented the dynamic behaviour of rotor bearing system with bearing inner-race detect and explained the effect of the system stability with the change in speed. A.J.Muminovic et al. (2018) presented numerical and analytical analysis of elastic rotor natural frequency. In previous studies, experimental analysis of dynamic characteristics in rotor bearing system has been rarely investigated.

The main aim of this study is to identify natural frequency and mode shape of system. The results of numerical simulation are validated with the experimental results.

Theory

A rotor dynamics is used to analyse the behaviour of structure ranging from jet engines and steam turbines to auto engines and computer disc storage.

Whirling of Shaft

Whirling of a shaft is its movement in a direction transverse to the axis of rotation. Whirling of shafts occurs due to rotational unbalance of a shaft, even in the absence of external

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loads, which causes resonance to occur at certain speeds, known as critical speeds. Whirl plots (mode shape), which plot the mode shapes of a rotor about the rotor axis at discrete rotation intervals.

Free and Forced Vibration

Free vibration occurs when a mechanical system is set in motion with an initial input and allowed to vibrate freely. If a system is subjected to an external force, the resulting vibration is known as forced vibration.

Resonance

Resonance occurs when the applied force or base excitation frequency coincides with a structural natural frequency. During resonant vibration, the response displacement may increase until the structure experiences buckling, fatigue, or some other failure mechanism.

Balancing Methods

The most common balancing methods are as follows.

(1) Static balancing or one plane balancing

(2) Dynamic balancing or two plane balancing

To determine whether a disc is balanced or not, mount the shaft on two low friction bearings. Rotate the disc and permit it to come to rest. Mark the lowest point on the circumference of the disc with chalk. Repeat the process several times, each time marking the lowest point on the disc with chalk. If the disc is balanced, the chalk marks will be scattered randomly all over the circumference. If the disc is unbalanced, all the chalk marks will coincide. The unbalance detected by this procedure is known as *static unbalance.* The static unbalance can be corrected by removing (drilling) metal at the chalk mark or by adding a weight at 180° from the chalk mark. Since the magnitude of unbalance is not known, the amount of material to be removed or added must be determined by trial and error.

Mathematical Modelling

The Jeffcott rotor model is considered for model analysis. It consists of a shaft, at the center of which, a fixed rigid circular disc is mounted, which is supported a pair of bearings. Fig. 1 shows general position of disc in a simple rotor.

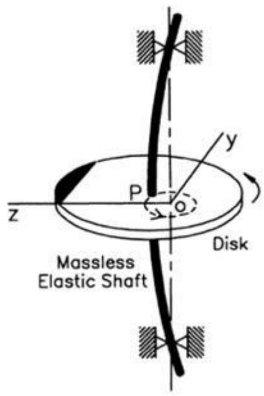

Figure 1 General position of disc in a simple rotor

Equation of Motion

The equations of motion for the rotor may be written, in the coordinates (y,z), as in following equation (3).

$$m\ddot{y}+c\dot{y}+ky=me\omega^2 e^{i\omega t} \quad \text{Eq (1)}$$

$$m\ddot{z}+c\dot{z}+kz=me\omega^2 e^{i\omega t} \quad \text{Eq (2)}$$

$$m\ddot{r}+c\dot{r}+kr=me\omega^2 e^{i\omega t} \quad \text{Eq (3)}$$

Where, e = mass eccentricity, m

ω = shaft rotational speed, rpm

k = shaft lateral stiffness, N/m

y, z = coordinates of the shaft

r = whirl radius, m

$$I_p = m\left(\frac{d^2 - D_0^2}{8}\right) \quad \text{Eq (4)}$$

$$I_p = m\left(\frac{d^2 + D_0^2}{16} + \frac{h^2}{12}\right) \quad \text{Eq (5)}$$

Where, I_P = Polar mass moment of inertia, kg-m^2

I_d = Diametral mass moment of inertia, kg-m^2

d = diameter of shaft, m

D_o = diameter of rotor disc, m

h = thickness of disc, m

The natural frequency of the Jeffcott rotor model can be calculated by the following equation.

$$\omega_n = \frac{\pi}{2}\times\left(n+\frac{1}{2}\right)^2\times\sqrt{\frac{gEI}{ml^4}} \quad \text{Eq (6)}$$

Simulation Model

In this paper, the numerical simulation is done for simple rotor model with residual unbalance condition. Simple rotor model with residual unbalance, shown in Fig. 2, is simulated for eigenfrequency analysis by using COMSOL Multiphysics software. The rotor geometry is drawn by using Bezier polygon as shown in Fig. 3. From this simulation, the natural frequencies and mode shapes (whirling of shaft) are investigated. And then, the comparison of natural frequencies from numerical simulation and experiment are discussed.

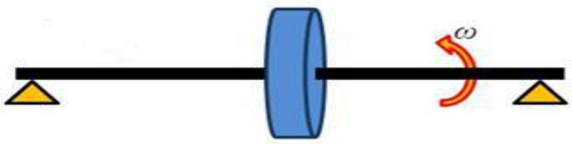

Figure 2 Simple rotor model

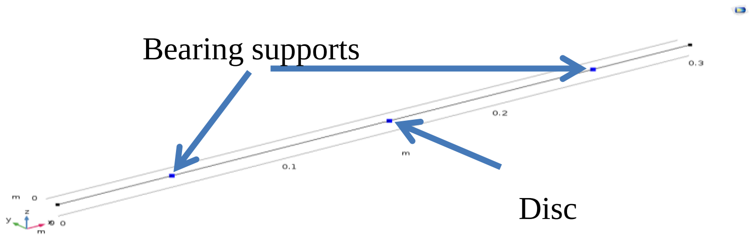

Figure 3 Geometry of rotor model

Table1 Specifications of shaft and disc

Shaft	Disc
Diameter = 8 mm	Diameter = 100 mm
Length = 300 mm	Thickness = 5 mm
ρ = 7850 kg/m3	ρ = 1250 kg/m3
$\nu = 0.3$	$\nu = 0.36$
E = 210 GPa	E = 1280 MPa

Table 1 shows the specifications of shaft and disc. Shaft is made with steel and disc and bearing housing are made with polymer (PLA).

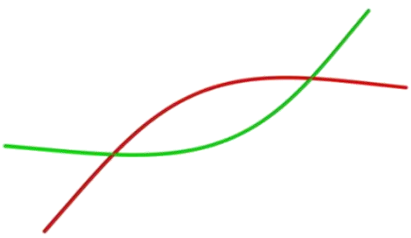

Figure 4 First mode at eigenfrequency 195.79 Hz

Figure 4 Second mode at eigenfrequency 457.85 Hz

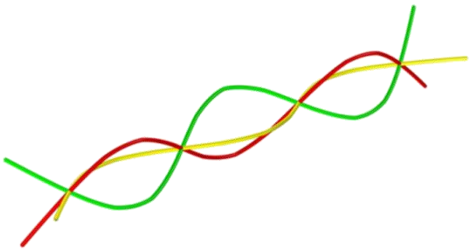

Figure 4 Third mode at eigenfrequency 855 Hz

From numerical simulation, mode shapes and natural frequencies are obtained. First mode occurs at 195.79 Hz as shown in Fig. 4. Second and third mode occurs at 457.85 Hz and 855 Hz.

Experiment

The shaft, 8 mm in diameter and 300 mm in length, is supported by two bearings. A disc, 100 mm in diameter and 5mm in thickness, is mounted at the mid span of the shaft. The accelerometer 1 (Fujikura ARF – 500 A) is placed on right side bearing block to measure the acceleration amplitude in vertical (Z-axis) directions. The accelerometer 2 is placed on left side bearing block to measure the acceleration amplitude in vertical (Z-axis) directions. The data from the sensor is collected by the computer which is connected to the data logger. The natural frequencies are identified from free vibration experiment. Impact hammer (Bruel & Kjaer 8204) is applied to get the initial disturbance in the rotor system. The responses are collected from accelerometer and time domain responses are converted to frequency domain responses by using data logger (Fujikura DC – 7004P).

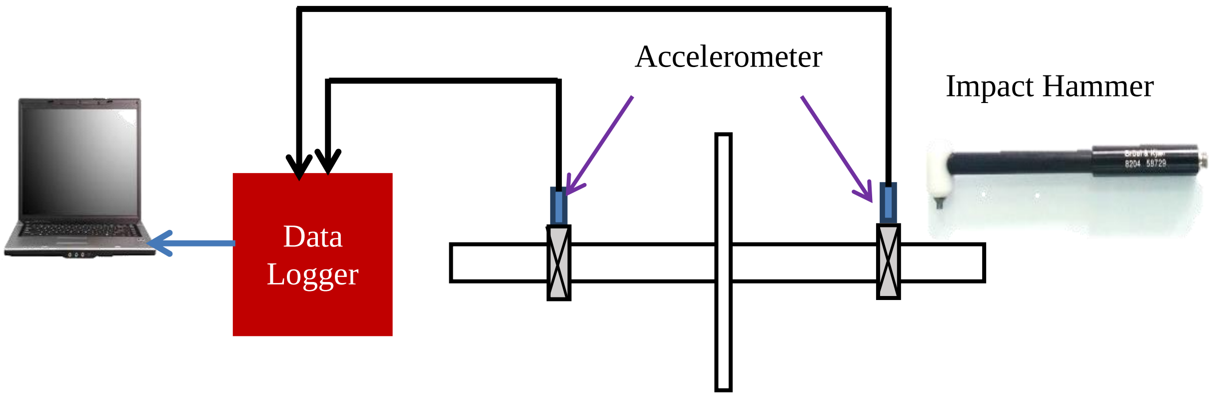

Figure 5 Experimental set up diagram

The experimental set up diagram is shown in Fig. 5. The actual experimental set up figure can be seen in Fig. 6. From free vibration experiment, the natural frequencies are observed as shown in Fig. 7. The results are discussed in next section.

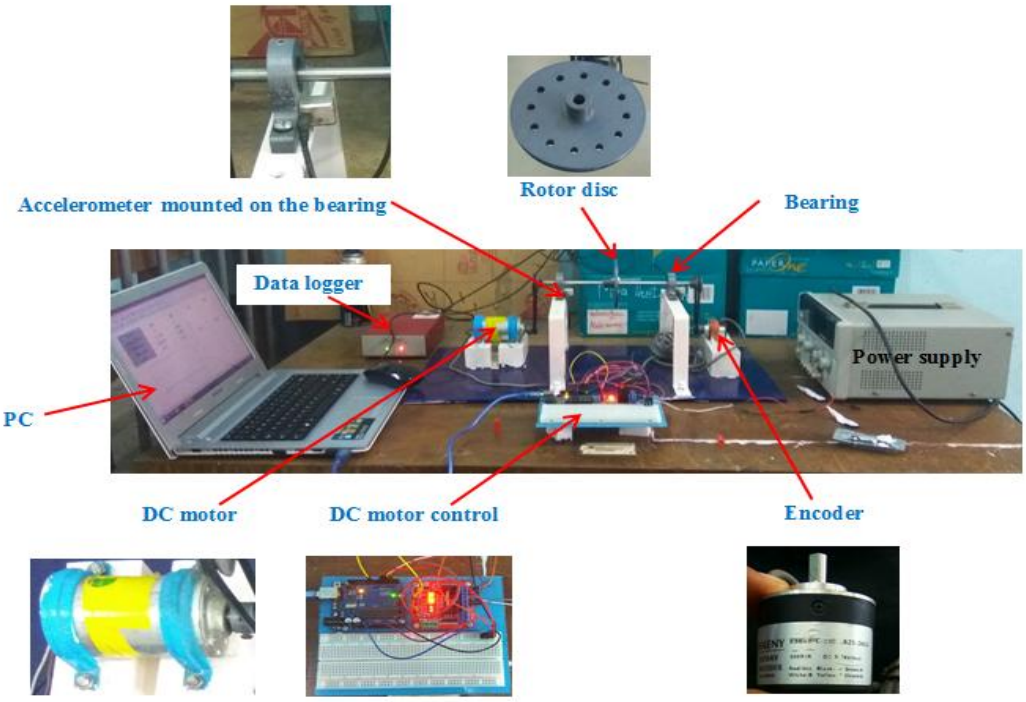

Figure 6 Actual experimental set up figure

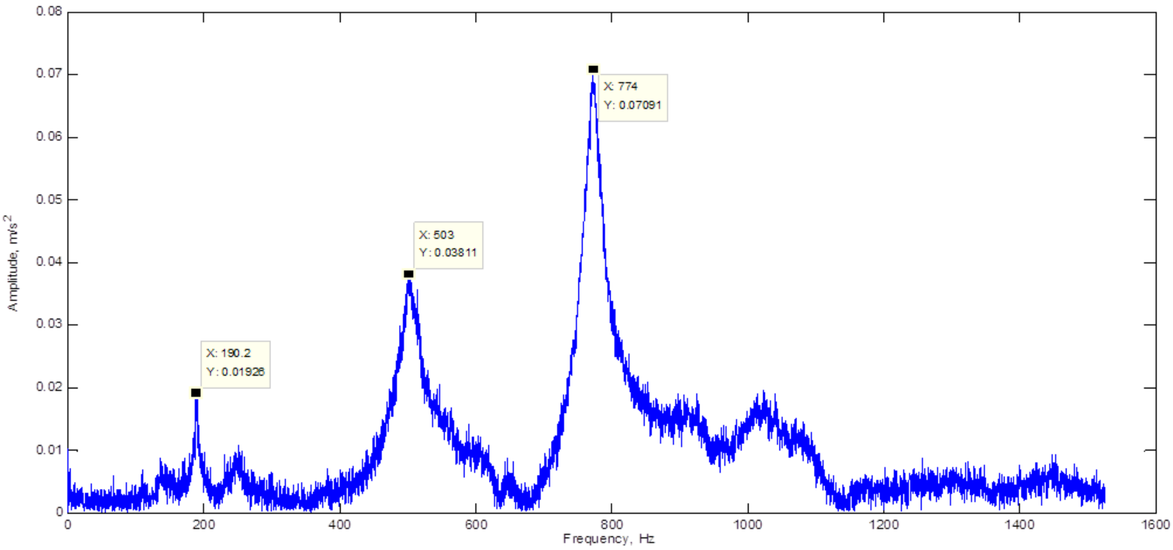

Figure 7 Natural frequency from experiment

Results and Discussions

The comparative results of eigenfrequencies between simulation and experiment for simple rotor system are shown in Table II. It can be seen that the natural frequencies for first three modes are investigated. Although there is a little difference between theoretical and simulation results, these results are nearly the same. The maximum discrepancy is 9.47%which occurs at third mode. There is a good agreement in natural frequencies with maximum difference of less than 10%.

Table 2 Comparison of Natural Frequency

Mode no.	**Natural frequency (Hz)**		**Discrepancy (%)**
	Simulation	**Experiment**	
1	195.79	190.2	2.86
2	457.85	503	8.97
3	855	774	9.47

Conclusion

This paper presented analysis of natural frequencies for simple rotor system. In this study, the dynamic characteristics of a simple rotor system were identified under numerical simulation and experiment. From the simulation and experiment, it can be seen that the comparison of dynamic characteristics was a good agreement although a little discrepancies. It is important to know the natural frequency of the system because of the resonance.

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