

# Study of effect of imbalance in rotor-bearing system due change in radius of gyration

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**Abstract**— Disk imbalance is a condition in which the center of mass of a rotating disk is not coincident with the center of rotation. Imbalance in a rotor system is unavoidable and it cannot be completely eliminated. Due to some reasons, such as porosity in casting, non-uniform density of material, manufacturing tolerances, and gain or loss of material during the operation, there is imbalance in rotors. In this study effect of radius of gyration of imbalanced mass on rotor-bearing vibration has been made. The imbalance increases with increase in inertia. A LabVIEW based SCXI system with accelerometer as a sensing element has been used in this study. Vibration signals collected from rotating machinery are often very complex which makes them very difficult to interpret in the time domain. Frequency spectrum analysis using Fourier transform and Continuous Wavelet transform has been made for detecting imbalance and its severity.

**Index Terms**— Imbalance disks, Rotor-bearing system, Frequency spectrum analysis, Accelerometer, Fourier transform and Continuous Wavelet transform.

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## 1 INTRODUCTION

Rotors play an important role in many industries and various engineering fields. They are used in large machines such as turbines or pumps in power generation plants and large vehicles such as airplanes or ships, as well as in small machines in workshops, automobiles, and computer hard drives. Disk imbalance can occur on the shaft from a variety of causes, such as fluctuating bending stresses. Consequently, machine failures from an imbalance shaft can compromise the safety of operators and result in increased operating and maintenance costs. Model based methods are used to identify location and magnitude of the fault. Model based methods are of different types. Many researchers have successfully identified rotor faults using model based methods. Markert et al. [1] and Platz et al. [2, 3] identified rotor faults using model based methods that use virtual loads generated in the system due to the faults. Nelson and McVaugh [4] have utilized the Raleigh beam method to formulate the rotor-bearing systems for purposes of determining critical speeds, stability, unbalance response, transient response etc. Greenhill and Cornejo [5] presented the rotor dynamic analyses typically ignore the potential for critical speeds to be created by traversing a backward precessional

crack in a structural member introduced a local flexibility that used to affect its vibration response. In the area of crack detection and diagnosis, the work presented by Seibold and Weinert [7] uses a modal Kalman filter to localize the position of a crack, however the observer is time invariant and the effects from the crack are transformed and approximated as another external force. Emmanouilidis et al. [8] have proposed that among reliable diagnostic methodologies for automatic fault diagnosis. Including statistical, polynomial, neural network, fuzzy and neurofuzzy techniques offer a framework for analysis. Sekhar [9] has investigated vibration characteristics of a cracked rotor with two opens and applied FEM analysis on rotor system for flexural vibration considering two transverse open crack. Suh et al. [10] have established that a crack has an important effect on the dynamic behavior of a structure. This effect depends mainly on the location and depth of the crack. Vyas and Kumar [11] have carried out experimental studies to generate data and discussed the development of neural network simulator for prediction of faults like mass unbalance, bearing cap loose, play in spider coupling and rotor with both mass unbalance and misalignment. Vania and Pennacchi [12] developed methods to measure accuracy of the results obtained with model based techniques to identify faults in rotating machines. Free and Forced response measurement methods have been proposed by Karthikeyan et al. [13] which gave crack flexibility coefficients as a by-product. Timoshenko beam theory was used in the beam modeling for transverse vibrations. A new concept of nonlinear output frequency response functions (NOFRFs) has been introduced by Peng et al. [14] to detect cracks in beams using frequency domain information and the results showed that the NOFRFs were a sensitive indicator of the presence of cracks providing

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whirl mode. Different crack identification techniques have been discussed briefly by Dimarogonas [6]. The presence of a

the excitation is of an appropriate intensity. Sinou [15] reported that the areas of instability increase considerably as the crack deepens, and that the crack's position and depth are the main factors affecting not only the nonlinear behavior of the rotor system but also the various zones of dynamic instability in the periodic solution of the cracked rotor. Patel and Darpe [16] have studied the influence of crack breathing model on nonlinear vibration characteristics of a cracked rotor and presented equations for two model switching crack model and response dependent breathing crack model. Different studies on multi-cracks has been summarized by Sekhar [17] and the respective influences, identification methods in vibration structures as beams, rotors, pipes etc. were discussed. Jalan and Mohanty [18] identified unbalance and misalignment experimentally in a rotor bearing system using model based method. These are taken in the form of disk weight.

In the present study aimed at understanding the dynamics of imbalanced rotors (in form of inertia) and reducing the ambiguity so as to improve the reliability of the imbalance fault diagnosis in a rotor-bearing system with experimental based technique. In this study effect of radius of gyration of imbalanced mass on rotor-bearing vibration has been made. A LabVIEW based SCXI system uses with accelerometer as a sensing element has been used. Frequency spectrum analysis using Fourier transform and Continuous Wavelet transform has been made for detecting imbalance of disk on shaft. MATLAB is used for processing the acquired signal to find the responses due to imbalance.

## 2 MODEL BASED FAULT IDENTIFICATION METHOD

### 2.1 Mathematical description

The amplitude of vibrations of disk represented by the vector  $x_0(t)$  at N DOF (degree of freedom) of the healthy rotor system due to the operating load  $F(t)$  during normal operation is described by the linear equation of motion

$$M\ddot{x}_0(t) + C\dot{x}_0(t) + Kx_0(t) = F(t) \quad (1)$$

where  $M$ ,  $C$  and  $K$  are mass, damping, and stiffness respectively, of any complex rotor system which includes the effect of bearings, foundations, gyroscopic effects etc, and  $\ddot{x}_0(t)$ ,  $\dot{x}_0(t)$ , and  $x_0(t)$  is the vibrations represented by the vector in form of acceleration, velocity, and displacement respectively of healthy rotor. The occurrence of a fault in the system changes its dynamic behavior. The extent of the change depends on the vector  $b$ , which describes the fault parameters such as type location, magnitude etc. The fault-induced change in the vibrational behavior is represented by the additional load acting on the healthy system;

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = F(t) + \Delta F(\beta, t) \quad (2)$$

where  $\ddot{x}(t)$ ,  $\dot{x}(t)$ ,  $x(t)$  is the vibrations represented by the vector in form of acceleration, velocity, and displacement respectively of faulty rotor. The vibrations induced represent the difference of previously measured normal vibrations  $x_0(t)$  of healthy system from currently measured vibrations  $x(t)$  of faulty system. The vibration may thus be written as;

$$\begin{aligned} \Delta x(t) &= x(t) - x_0(t); \\ \Delta \dot{x}(t) &= \dot{x}(t) - \dot{x}_0(t); \\ \Delta \ddot{x}(t) &= \ddot{x}(t) - \ddot{x}_0(t) \end{aligned} \quad (3)$$

Subtraction of the equations of motion for the healthy system Eq. (1) from that of faulty system Eq. (2), and using Eq. (3), the equation of motion for vibration can be represented as;

$$M\Delta \ddot{x}(t) + C\Delta \dot{x}(t) + K\Delta x(t) = \Delta F(\beta, t) \quad (4)$$

The system matrices remain unchanged and the rotor model remains linear. Only the equivalent loads induce the change in the dynamic behavior of the healthy linear rotor model. To identify the fault parameters, the difference of the theoretical fault model and the measured equivalent loads will be minimized by some statistical algorithm like least square fitting.

### 2.2 Scheme of imbalance diagnosis

To identify the imbalance in any system a model based scheme is established with the help of amplitude of vibration. The prerequisite for this kind of imbalance diagnosis is the measured vibration signal data for the healthy rotor system in the form of displacement which can be written as " $x_0(t)$ ". Now in order to find out the imbalance in the system, measured vibration signal data for the imbalance system are stored in the form of displacement which can be written as " $x(t)$ ". The vibration is calculated from the measured vibration displacement data for both the healthy system and the imbalance system at same operating conditions. The displacement (vibration) is given by

$$\begin{aligned} \Delta x_0(t) &= x(t) - x_0(t) \end{aligned} \quad (5)$$

which is a measure of imbalance in the system.

## 3 EXPERIMENTAL SET UP

A test rig for experimental validation of the model based identification technique has been built at is located in the Metrology Laboratory at Sant Longowal Institute of Engineering and Technology, Longowal, Sangrur, Punjab (India). The Photograph of test rig, shown in Figures 2, consists of a 28.9 mm diameter and 405.5 mm long shaft supported on two identical ball bearings having no. 6205 and is driven by a motor. Four disks of masses 640, 635, 635 and 635 grams are mounted on the shaft. Out of four available disks; one disk is mounted at a time. The disk is fixed on the rotor shaft by nut and bolt. One additional nut of weight 3.5 gram is to create unbalance. Specification of test rig is shown in Table 1.

TABLE 1:  
 ROTOR-BEARING DATA

<b>Motor specification:</b>	
Power supply	220/230 V, 4.2 A, AC power supply
Speed range (using pulley)	1150 rpm, 2050 rpm, 3080 rpm
<b>Rotor shaft specification</b>	
Diameter of the shaft	28.5 mm.
Length of the shaft	570 mm
Distance between bearings	405.5 mm
<b>Disks specification:</b>	
No. of disks	4
Diameter of the disks	5 cm, 8 cm, 9 cm, 10 cm.
Weight of disks	0.640 kg, 0.635 kg, 0.635 kg, 0.635 kg.
Material of the shaft and disks	
Mild steel	
Sampling rate of data acquisition	
65535 sample/second	

Fig. 2: Photograph of experimental setup



Fig. 3: Rotor disks, (a) 100 mm outer diameter & thickness 12.5 mm, (b) 90 mm outer diameter & thickness 15 mm, (c) 80 mm outer diameter & thickness 20 mm, (d) 50 mm outer diameter & thickness 60 mm

Figure 4 and 5 show the acquired vibrational signal with LabVIEW based SCXI system

**3.1 Description of experimental set-up**

Schematic of experimental setup on which experiments were conducted is shown in Figure 1.

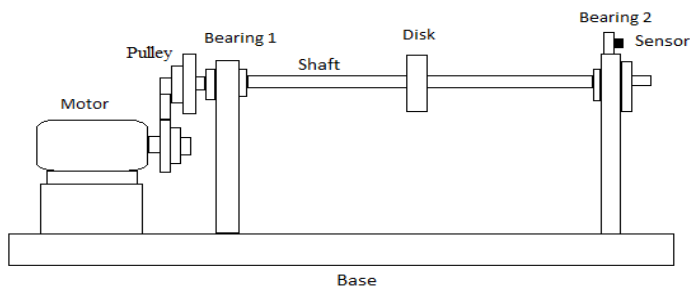


Fig. 1: Schematic diagram of experimental setup of Rotor-bearing system

Photograph of experimental setup and rotor disks are shown in Figure 2 and 3.

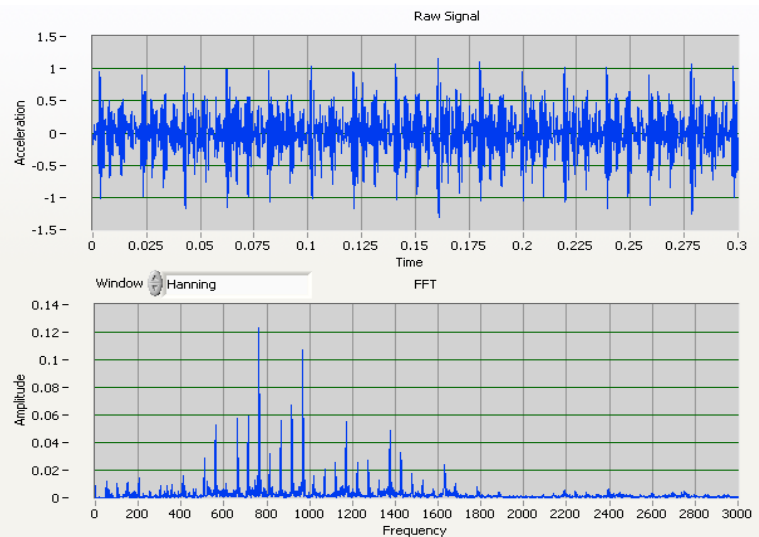
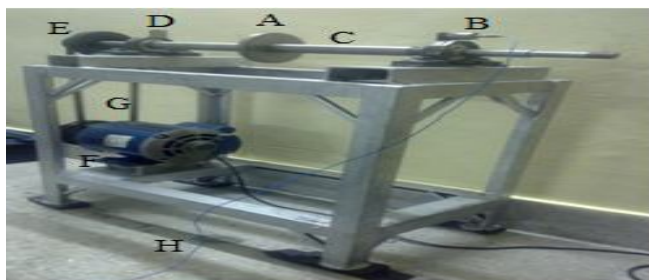


Fig. 4: Disk (having outer diameter 5 cm) mounted at centre position of shaft rotating at 3080 rpm

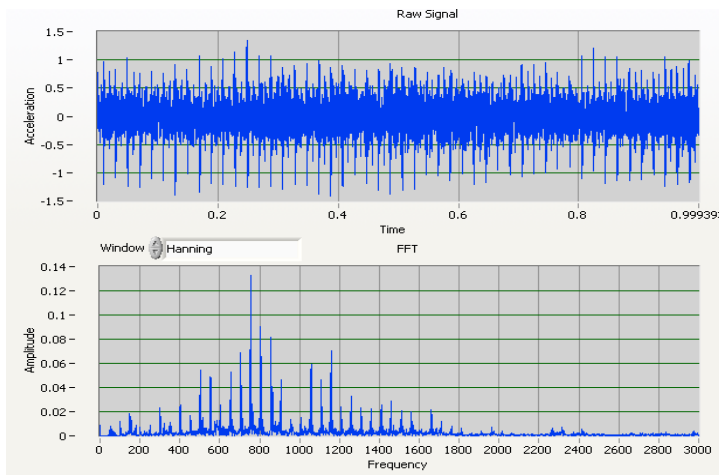


Fig. 5: Disk (having outer diameter 8 cm) mounted at centre position of shaft rotating at 3080 rpm

## 5 RESULTS AND DISCUSSION

### 5.1 Effect of radius of gyration (ROG) of imbalance mass on shaft vibration

From the recorded signal, value of amplitude of vibration for different disk and speed combination when disk is placed at centre of the shaft is tabulated and shown in Table 2. The graph of this Table is shown in Figure 6.

TABLE 2:  
THE AMPLITUDE OF VIBRATION FOR DIFFERENT DISKS (HAVING OUTER DIAMETER 5, 8, 9, AND 10 CM) AT DIFFERENT RPMS (1150, 2050, AND 3080 RPM) WHEN POSITIONED AT CENTRE OF SHAFT

Rotor shaft speeds (rpm)	Amplitude (mV) of vibration at centre position on shaft of 5 cm disk diameter ROG (2.0347 cm)	Amplitude (mV) of vibration at centre position on shaft of 8 cm disk diameter ROG (3.0025 cm)	Amplitude (mV) of vibration at centre position on shaft of 9 cm disk diameter ROG (3.337 cm)	Amplitude (mV) of vibration at centre position on shaft of 10 cm disk diameter ROG (3.676 cm)
1150	0.3715	0.4021	0.4440	0.4502
2050	0.6776	0.6906	0.8262	0.8497
3080	2.5520	2.7850	2.9050	3.3980

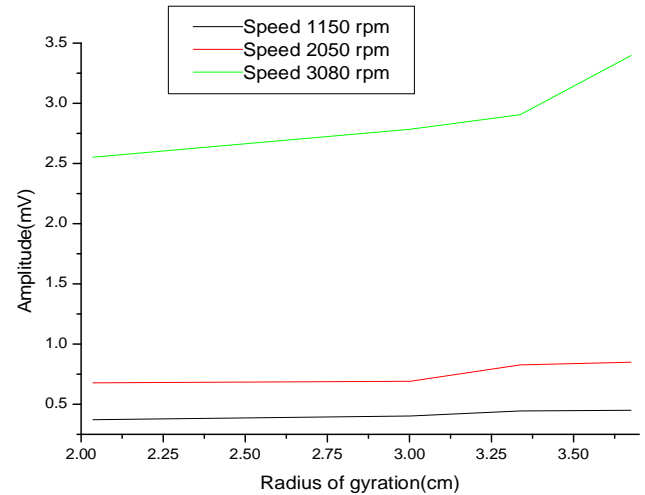


Fig. 6: Effect of radius of gyration on linearity of responses at the rotor mid-span

The above plot is shown at the speed 1150 rpm, the amplitude of vibration of first disk (having radius of gyration 2.0347 cm), second disk (having radius of gyration 3.0025 cm), third disk (having radius of gyration 3.337 cm), and fourth disk (having radius of gyration 3.676 cm) are observed as 0.3715, 0.4021, 0.4440, and 0.4502 mV respectively at the centre position of disk on shaft.

Imbalance increase between first and second disk, second and third disk, and third and fourth disk are 0.0306, 0.0419 and 0.0062 mV respectively. It indicates that imbalance increases with increase the radius of gyration.

Similar is the trend at 2050 and 3080 rpm. With increases in rpm rate of increase in imbalance is more.

## 6 CONCLUSIONS

In this research paper, the results of radius of gyration of imbalance mass on rotor bearing shaft vibration, we can see that the imbalance increases with inertia increase. This method has thus demonstrated the model based fault detection system for a simple rotor-bearing system. This method may be useful for large systems like in turbine shafts, and gearboxes etc. Such a method has enormous potential are automated diagnostics process whereby the measurement of the responses, the fault condition and location can be detected. Then by simple measurement of the axial vibration alone and applying the proposed model based technique, faults of rotor-bearing system can easily be identified.

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