Roller Bearing Fault Detection Using Vibration Data Analysis and Non-linear Dynamic Model Approaches

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ABSTRACT

Rolling bearings are very critical components of rotating machines, and the presence of defects in the bearing may lead to failure of the machine. Hence, early detection of such defects under operating condition of the bearing may avoid malfunctioning and breakdown of the machine. Defective bearings are source of vibration and these vibration signals can be used to evaluate the faulty bearings. This research paper seeks to detect and diagnose roller bearing early faults using vibration data analysis techniques. Bearings of three different conditions, healthy bearing (baseline), outer race fault and inner race fault were considered. Time domain, frequency domain and the envelope spectrum techniques were utilised to analyze the experimental collected data. The results of the investigation revealed that, the envelope spectrum technique proved to be the most reliable technique to detect rolling bearings faults, where the noise and interference sources are suppressed significantly.

Furthermore, one degree of freedom non-linear dynamic model was also developed using Matlab Toolbox to study the bearing vibration response of healthy and faulty bearings. There was a good correlations between the numerically simulated and experimental results, and it is proved that this model is reasonable and can be implemented for the study and prediction of roller bearings early defects.

Keywords: Rotating machinery; Rolling bearing; Vibration data analysis; Envelope spectrum.

1. Introduction

Rolling bearings are basic and key components in rotary machine systems. Any slight fault will lead to abnormal working conditions of the rotary machine structures, and even affect the normal running of other components [1]. Therefore, it is important to have an effective bearing condition monitoring (CM) and fault diagnosis system in place so that incipient bearing faults can be detected and correctly diagnosed on time, to prevent them from deteriorating further to cause damage to the machine. This research paper seeks to detect and diagnose roller bearing early faults using vibration data analysis techniques, and it is an extension to the research work conducted at the University of Huddersfield [2]. The idea of condition monitoring has grown exponentially since its extensive use in the sixties [3]. The analysis of vibration signals is major technique for condition monitoring of bearing in machine components. A slight transformation in the working environment could alters this signal. Therefore, when this signal is different from the normal signal, an impending failure in the process can be expected. The process of machine condition monitoring outlines a certain parameter of the machine to ensure that it does not go over the recommended operational limits. These parameters include but not limited to oil debris monitoring, temperature monitoring, and vibration monitoring [4]. Among these, vibration monitoring is found to be the most widely used technique [5]. Therefore, machine CM ensures a proper timetable of maintenance and solutions to avoid catastrophic failure and predicting the unseen faults, and also to be employed in case the parameter exceeds its limits when the signal generated is different from its standard value [6]. By detecting and analysing the machine vibration, it is possible to determine and predict the machine failure. This paper provides a relative of various techniques used for finding the fault in the roller bearings based on vibration analysis method and non-linear vibration dynamic model.

2. Condition Monitoring Using Vibration Analysis

Condition monitoring is the process of monitoring a parameter of condition in machinery, such that a significant change is indicative of a developing failure. Machine condition monitoring can be realized by monitoring following characteristics: vibration, aural, visual, operational variables (state of the system), temperature and wear debris (e.g. oil analysis), [7]. Several methods are used to diagnose faults within machines as no single method is enough to cater the diagnosis needs of the entire system. However, the vibration measurement is the most dependable one among the CM techniques. It covers almost 90 % of the failures mechanisms. When a machine is about to develop fault, it starts producing particular sounds and sends signals which vary in vibrations. These variations determine the exact nature of the faults which may cause a breakdown. With these series of variations in the vibrations level, the maintenance staff can detect the faults at an early stage; so they not only save the machine from total breakdown but also guarantee safety of the machine [8].

2.1 Time Domain Analysis Method

The most common method is the time domain parameter for determining machinery faults, which known as the probability density distribution. This technique is not reliable and dependable method as it does not necessarily determine the exact nature of the particular defects, and can be misleading. The time-domain features are extracted from the raw vibration signal through statistical parameters. The statistical parameters are used: Peak value (Pv), Root Mean Square value (RMS), Crest factor (Cf), Kurtosis value (Kv), Clearance factor (Clf), and Impulse factor (If). Where, Pv and RMS values is related to the energy of the signal. Kv and Cf are measurements of the signal spikiness [9]. These techniques also lack in representing where the fault lies, and which component of the machine is going to likely develop the fault [10].

2.2 Frequency Domain Analysis Method

For evaluating the mechanical vibrations in the hardware, a process known as Fast Fourier Transform (FFT) is used. This process changes the time domain signals into frequency domain signals that can then be employed for the analysis of the machine conditions [11]. If there is a local fault in the bearing or other mechanical components are not properly fitted; vibration amplitude occurs. This vibration is sizable and its subsequent frequencies are large [12]. However, there is a considerable difference in locating the part harmonic due to the residual sound of the machine and the vast frequency range. Thus, it is difficult to control the numerous frequencies of rolling elements as well as the rest of the structure [13].

2.3 Envelope Analysis Method

Hidden faults in the rolling element bearings of the machines can be found by the employment of envelope analysis [14]. The basic concept of this technique is that whenever a fault develops in a machine, the bearing comes in contact under load with other parts, and an impulse vibration is produced for very short interval as compared to time and interval between other impulses. Consequently, the energy obtained is widely distributed at a greater frequency range. Also, there is a high signal to noise ratio around the resonance which can be easily recognised from others [15].

3. Types of Rolling Elements

Machines are essentially based on rolling bearings. There are two types of rolling bearings; rolling element and sliding bearings. Depending upon the shape of the rolling elements, the various sizes and designs of rolling elements bearings are further divided. The manner in which the load is supported by the bearing also leads to further classifications of the bearings. Table 1 indicates the classifications of rolling bearing.

Table 1:*Types of Rolling Element Bearing* [16]

Rolling Bearing			
Ball Bearings		Roller Bearings	
Radial Ball Bearings	Thrust Ball Bearings	Radial Roller Bearings	Axial Thrust Roller Bearings

3.1 Bearing Description and Components

There are four basic parts of the rolling bearing; inner race, outer race, rolling elements and cage or separator. Guide race and seals may be added in some particular bearings. The bearing load is supported by the inner race, outer race and rolling elements. The adjacent rolling elements are separated by the cage, which prevents friction among the two moving rolling elements [17]. The bearing components are illustrated in figure 1.

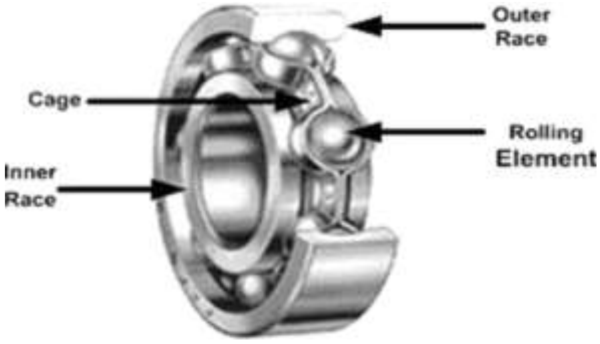

Figure 1. Components of Rolling Element Bearing [17]

4. Dynamic Vibration Model of Deep Groove Ball Bearing

The raceway contact of rolling element can be taken as a spring mass system in order to study and observe the rolling element bearing structural vibration features. In the system the inner race is secured to a motor shaft, while the outer race is secured with a firm support. The non-linear phenomenon among deformation and force due to elastic deformation is acquired by the Hertzian theory. Hence Hertzian contact deformation theory is used for determining the contact force [18]. Figure 2 illustrates the bearing as non-linear contact spring mass system.

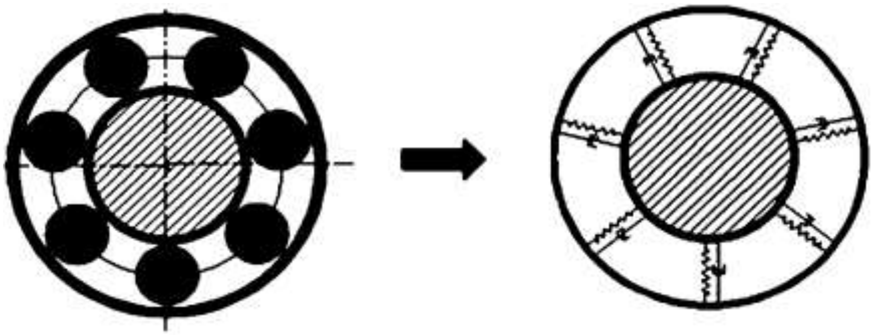

Figure 2. Rolling Elements Substituted by Dash-Pot and Spring [18]

4.1 Contact Force Calculation

The non-linear relation load deformation in accordance with Hertzian contact deformation theory [18] is stated as:

$$F = K\delta_r{}^n \tag{1}$$

Where, K represents the load deflection factor, n represents the load deflection exponent, F represents the contact force, and δ_r the radial deflection; n=10/9 for roller bearing and n=3/2 for ball bearing [19]. The outer k_o and inner k_i raceways to ball contact stiffness can be determined as:

$$k_{o,i} = 2.15x10^5 \sum \rho^{-1/2} (\delta^*)^{-3/2}$$

(2)

Where, δ^* is the dimensionless contact deformation gained utilising curvature variation, and $\sum \rho$ is the sum of curvature which is determined utilising the curvature radii in a pair of principal planes passing by the point contact. The summation of the approaches within each raceway and rolling element give the total deflection among two raceways [19]. Considering this the following equation can be obtained:

$$K = \left[\frac{1}{(\frac{1}{k_i})^{1/n} + (\frac{1}{k_o})^{1/n}} \right]^n \tag{3}$$

A circumferential half sinusoidal wave is used to depict the defect. The inner and the outer raceway defects can be observed respectively in figure 3.

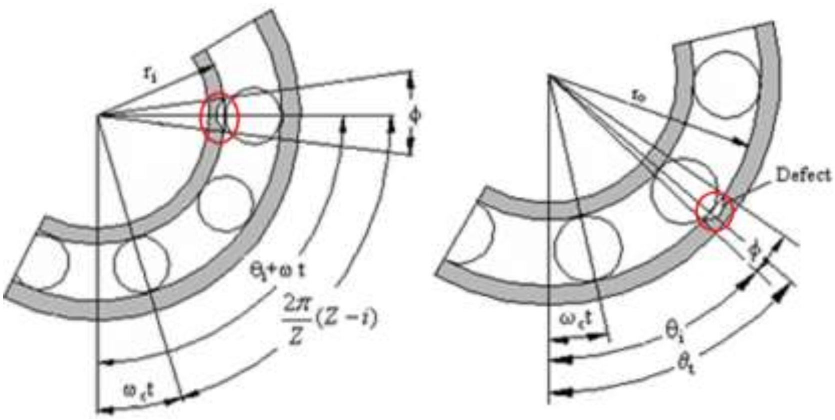

Figure 3. Defect on Inner Race and Outer Race [17].

The radial Displacement is calculated from figure 3 by taking into consideration the restoring force and resulting distortion due to a defect on the race along both axis by:

$$F_{XD} = \sum_{i=1}^{Z} k\left[(x\cos\theta_i + y\sin\theta_i) - (c_r + \Delta)\right]^{3/2}\cos\theta_i \tag{4}$$

Where, Δ is the additional defect deflection and is specified as:

$$\Delta = \frac{D_b}{2} - \frac{D_b}{2}\cos(\frac{\theta_i}{2})$$, D_b is the ball diameter.

$$F_{YD} = \sum_{i=1}^{Z} k\left[(x\cos\theta_i + y\sin\theta_i) - (c_r + \Delta)\right]^{3/2}\sin\theta_i \tag{5}$$

For a 2-DOF system the equations of motion are given as [16]:

$$Mx + cx + F_{xD} = W \tag{6}$$

$$My + cy + F_{yD} = 0 \tag{7}$$

Where, W is the radial load (*N*).

5. Experimental Procedure

A research bearing test rig was used to extract the bearing vibration data as shown in figure 4. A 3-phase electric induction motor (1500 RPM) is used with a dynamic brake. 3 shafts are used to link the motor to the brake through two pairs of matched flexible couplings. The 3 shafts are positioned in two bearing housings.

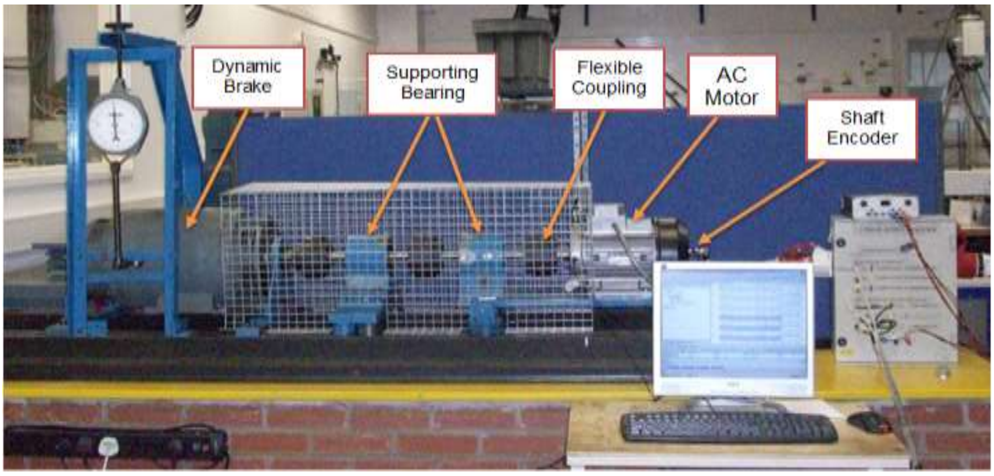

Figure 4. Motor Bearing Test Rig

The dynamic of the bearing elements is explained through five basic motions, where each movement has a unique corresponding frequency [20]. Table 3 shows the corresponding character frequencies.

Table 3:*Calculations of Bearing Frequency*

Defect position	Frequency (Hz)
Inner race	135.6
Outer race	89.39
Rolling elements	58.29
Cage	9.93

The faults of outer race and inner race can be observed in figure 5-1 and 5-2 respectively. The defect size of the inner race is bigger than the outer race.

Figure. 5-1 Outer Race Defect

Figure. 5-2 Inner Race Defect

6. Discussion of the Results

An extensive comparative analysis of the different techniques which were used in this study for detecting the bearing faults has been carried out. The study of the bearings faults were conducted under three different conditions which are a healthy bearing that considered as a baseline, outer race fault and inner race fault. This will help to indicate the difference amongst the two bearing faults conditions.

6.1 Frequency Domain Results

To observe the raw data spectrum signal the FFT is applied on the signal to convert the time domain signal into frequency domain signal. Considering the bearing's geometric structure and the shaft frequency acquired from the vibration signal's spectrum, the fault feature frequencies can be determined for each of the bearing. The spectrum does not show the fault feature frequencies and their harmonics but the shaft rotational frequency along with its multiples can be seen.

Table 4: *Fault Characteristic Frequencies*

Fault Type	Defect Frequency (Hz)
Inner race	135.6
Outer race	89.39

The vibration spectra of the outer and inner race fault in comparison to the baseline have been shown in figure 6. The initial peaks in the spectrum can be seen by the variance among the faulty spectra and baseline.

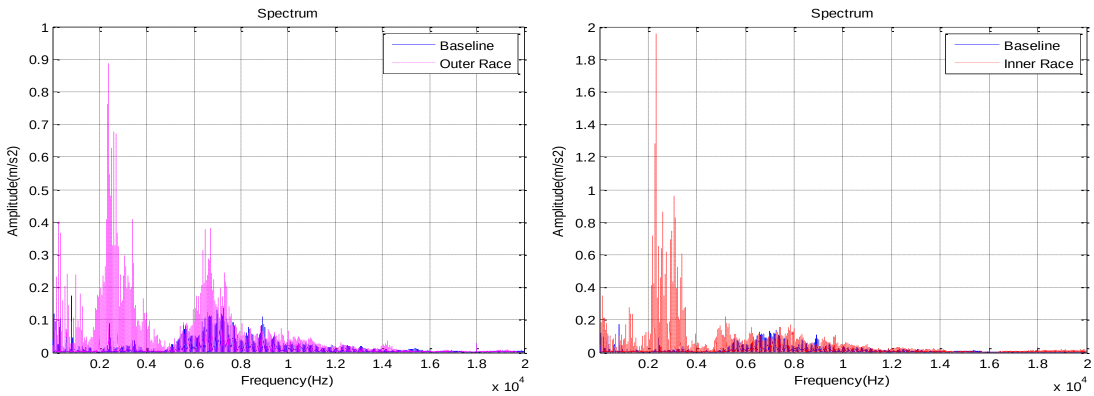

Figure 6. Outer and inner race spectrum

6.2 Envelope Spectrum Results

The envelope spectrum of the outer race and inner race faults and the baseline are illustrated in figure 7. The baseline case has no indication of spectral lines while the characteristic fault frequency and its harmonics for the outer race fault are clearly visible, it can be observed that these faults have an obvious indication. In addition, this fault can be seen clearly in terms of feature frequency and its harmonics.

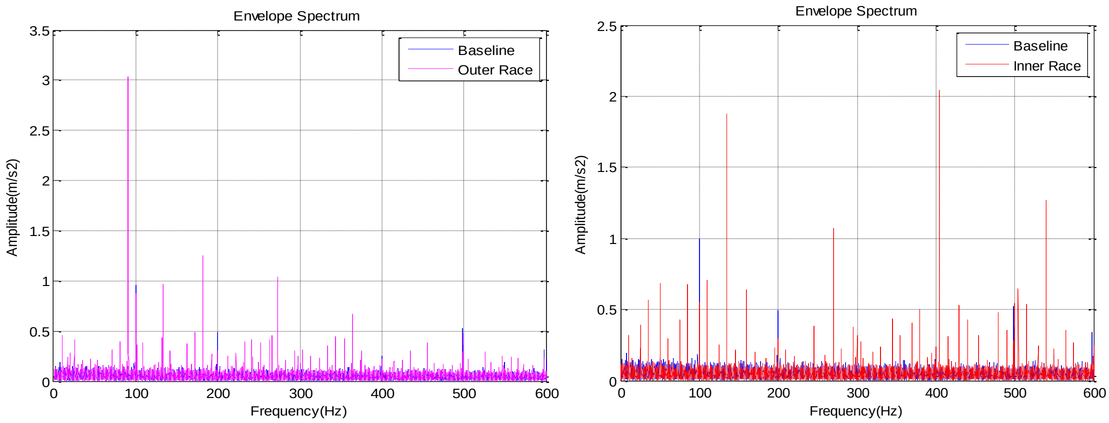

Figure 7. Outer and Inner Race Envelope Spectrum

6.3 Non-linear Dynamic Model Results

Figure 8 indicates the vibration displacement of the baseline and inner race defect on the left side and the outer race defects in the right side. It can be observed that the baseline has a small indication of vibration displacement due to diametric clearance and no defect on raceways. For the inner race case the vibration displacement can be seen clearly and it has different level of vibration amplitudes. For the outer race defect, it can be noted that a clear indication of vibration displacement in addition to the amplitude level of vibration is maximum due to the defect is at zero degree (load zone).

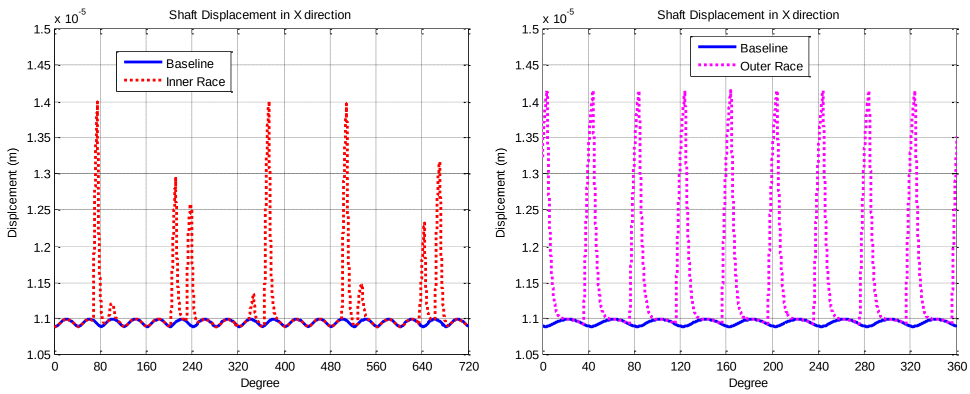

Figure 8. Vibration Displacement of Inner Race and Outer race with Baseline

Figure 9 displays the vibration spectra of baseline with inner and outer race defect. For the baseline the vibration spectra shows a small sign of loading frequency and its harmonics, which is the cage frequency multiply by the number of balls. From Figure 9, it can be noticed that the outer race defect has a clear indication of vibration displacement in addition to the amplitude level of vibration is maximum due to the defect is at zero degree (load zone).

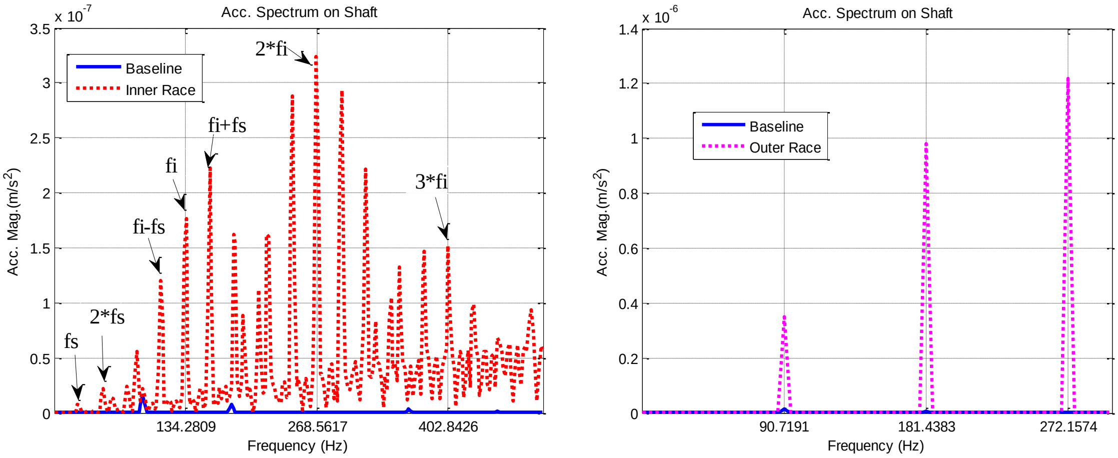

Figure 9. Vibration Spectra of Baseline with Inner and Outer Race

7. Conclusion

The results of the investigation have revealed that the bearing faults can be identified reliably through the envelope analysis under the application of Fast Fourier transform(FFT). The bearing fault frequencies and harmonic are clearly indicated through this technique. The effect of the inner race and the outer race defects were simulated through the dynamic model, the characteristic defect frequencies are denoted by the frequency spectrum. The results of the non-linear dynamic model present a clear correlation with the experimental results for the healthy and faulty cases. Therefore, the developed dynamic model can be used to investigate the faults of any geometric structure bearing. Additionally, it can be performed for bearings faults investigation in a short period of time and less costs, whereas the experimental case takes longer time and higher expenses to perform the diagnosis for a specific bearing fault.

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