FAULT DIAGNOSIS IN ROTATING MACHINES USING VIBRATION ANALYSIS

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ABSTRACT

Rotor systems are used in many different engineering fields. To prevent the failure of the parts and, ultimately, the system, it is vital to monitor the operation of the various rotating system components given the high-speed requirements of today's machinery. Some of the typical issues with rotating systems are bearing problems. One method for periodically assessing the performance of the individual components of a machine is condition monitoring. Monitoring vibration, noise, temperature, wear debris, oil contamination, and other factors can help determine how well the system is working. Vibration signatures are the modifications to these parameters. The signatures reveal differences in machine's circumstances. The probable reasons of machine failure can be predicted by examining the vibration signatures created in machine parts. One of the crucial components of any rotating machinery is bearing, and vibrations created due to various defects are transmitted to the foundation through bearing. The failure of bearing could result in collapse of the entire system; so it is crucial to investigate the defect in bearings. Early mechanical failure is indicated by the vibration signal. In the current work, frequency spectrum for bearing defects was experimentally investigated. It gives a clear understanding of how the frequency spectrum changed when the ball bearing's inner race has defects. In this investigation, wire-cut EDM was used to induce defects in the inner race of the ball bearing. In the current work, an experimental investigation into this topic was conducted. A piezoelectric, tri-axial accelerometer was employed in an experimental setting in conjunction with the photon+ (Brüel & Kjaer) ultra-portable dynamic signal analyzer for data collection to acquire frequency spectra for healthy and faulty bearing. Rise in amplitude at the ball pass frequency inner race (fbpfi) will be observed, if the bearing has a fault at the inner race. At operating frequencies of 10 Hz, 15 Hz, and 20 Hz, the frequency spectrum for bearing with and without has been acquired.

Keywords: *Bearing defect, Frequency spectrum, Inner race etc.*

1. INTRODUCTION

Typical components of rotating machines include shafts, bearings, motor drives, and couplings. These parts' functionality significantly influences the entire machine's performance [K.M. Al-Hussain and I. Redmond]. The performance of rotating machines depends greatly on rolling element bearings, and any bearing defects might result in machine failure [Deepak P et al.]. Therefore, by identifying bearing abnormalities at the proper time, malfunctions and machine breakdowns can be avoided. There has been a critical evaluation of literature in the domains of vibration monitoring techniques for fault identification and analysis of the resulting vibration signature. Crack identification in a rotor system can be accomplished by fault diagnosis using vibration analysis. A fractured rotor [A.S. Sekhar and B.S. Prabhu] can be investigated using crack detection.

Fault diagnosis using vibration analysis can be used for crack detection in a rotor system. Crack detection of cracked rotor [A.S. Sekhar and B.S. Prabhu] can be analyzed using vibration analysis. The effect of parallel misalignment on the lateral and torsional responses of two rotating shafts [[K.M. Al-Hussain and I. Redmond] is examined with mathematical model. Misalignment of multi bearing rotor system can be characterized by presence of harmonics in the vibration signal [A.W. Lees]. Presence of harmonics is due to the nonlinearities in fluid film journal bearings.

Arun Kr et al. [8] studied the effect of shaft misalignment and rotor unbalance in rotating machines, in which a model based technique for fault diagnosis of rotor–bearing system is described. Fault diagnosis of a rotor with transverse shaft cracks is studied by BSN Murthy et al. Nonlinear second-order system of equations are solved to obtain unbalance response of the rotor under different parameters like disk asymmetries and crack depth ratio.

Coupling misalignment in the rotor system can produce nonlinear forces and moments [Kun Wu] , which will result in failures of the machines. Experimental study can be carried out to identify the type bearing defect by acquiring vibration spectrum (Pratesh Jayswal et al.). Author developed a fault detection model in the bearings. Inner race, outer race &ball defects were examined using vibration analysis.

Theoretical mathematical model for outer race defects in rolling element bearings can be developed (Karthik Kappaganthu et al.) for faults detection in bearings.

Misalignments in couplings will constitute forces and moments, which will alter the vibration patterns, which are of interest for investigations focusing on malfunction signatures and, consequently, on fault identification and diagnosis [Felipe Wenzel da Silva].

M. Xu and Marangoni have carried out fault identification on rotor dynamic test system to verify the results of theoretical model of their earlier work. A helical and flexible coupling were used in the experiment. A shaft misalignment was created in the test rig and rotor shaft displacement was measured under different misalignment and unbalance conditions. The measured and predicted spectra for unbalance and misalignment reported as 1x and 2x shaft running speed respectively.

Therefore, it's crucial to identify defects in rotating systems to prevent machine failure. A common method for finding faults in rotating systems is condition monitoring based on vibration analysis. The present research employs an FFT analyzer to experimentally identify the frequency spectrum of various defects in rotating system. Using this approach, vibration response of the rotating system can be easily acquired in frequency domain and various types of fault in the system can be predicted more accurately and precisely. Even though the current study focus on bearing fault, the same technique can be adopted for other faults also. From the vibration response, rise in amplitude clearly gives the severity of faults present in the system. The novelty of the present study is that, the current experimental model is developed in such way to accommodate different types of faults and diagnosis the same.

2. DESIGN AND FABRICATION OF EXPERIMENTAL SETUP

A laboratory model is being designed and developed to capture the frequency spectrum using an FFT analyzer for both healthy and damaged bearings operating at various speeds. The experimental setup, which includes three deep groove ball bearings TR SB 202-10, a flexible coupling, and a 0.25 HP dc drive, is depicted in Figure 1. One of the three bearing housings is provided to support the extended segment of the motor shaft. Driven shaft is supported with the help of two ball bearings. The shaft has a 350 mm length and a 225 mm bearing span. The estimated the shaft's diameter to be 15.875 mm since the bearing used in present setup available in market. Four anti-vibration levelling pads support the assembly's table. These pads offer effective vibration absorption and precise readings.

The current investigation collects vibration signals for both healthy and damaged bearings. Wire cut EDM was used to consciously cause fault in the bearing at inner race, as seen in figure 2. The fault is a 0.35 mm wide slit. A flat surface was created on top of the test bearing housing, where an accelerometer with the help of clip was mounted.

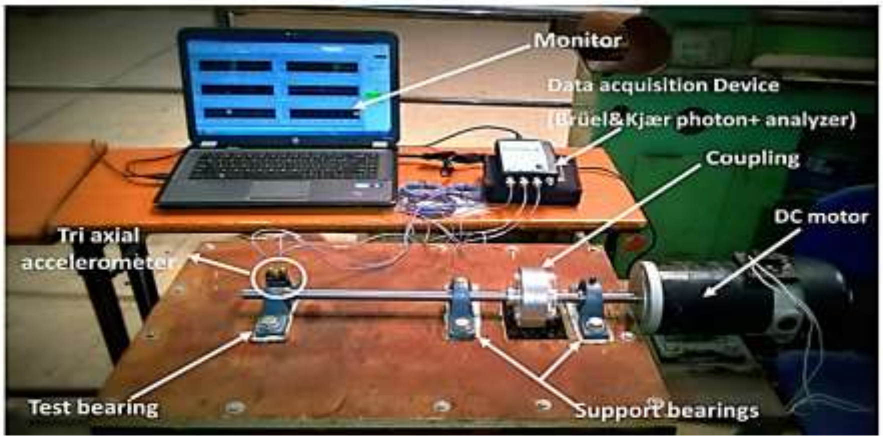

Figure 1. Experimental Set up

Figure 2. Defect made on inner race of test bearing using EDM method

3. DATA MEASUREMENT AND ACQUISITION

For data acquisition, signal conditioning, and modification, a piezoelectric sensor for acceleration measurement (Type AC 102-A, SI. No. 66760, Brüel & Kjaer, Denmark) was utilized in conjunction with the photon+ ultra-portable dynamic signal analyzer (Brüel & Kjaer, dynamic signal analyzer, Denmark). Since a tri-axial accelerometers used for data acquisition, it could collect signals from all 3 directions. FFT software was used to process the data. The tri-axial sensor was installed on top of the bearing block and used to collect vibration signals from all three axes, including axial and radial. In RT photon+ software, the output displayed as a frequency vs. amplitude graph is chosen. For measuring the 1600 spectral line's vibration amplitude in terms of displacement, a frequency band of 0–50 Hz is utilized, and the maximum displacement range is set to 0–1 mm. Vibration spectra for three different speeds such as 10Hz, 15Hz, and 20Hz bearings with and without defects were collected. On end bearing, the frequency spectrum is recorded. The method was used to collect vibration spectrum for shaft misalignment. The Vibration signals for 10Hz, 15Hz, and 20Hz, for bearings with and without defects are collected. On end bearing, the frequency spectrum is recorded. The method is used to collect vibration spectra for shaft misalignment. A frequency band of 0–50 Hz and a maximum displacement range of 0–1 mm were employed to measure the vibration amplitude in terms of displacement for the 1600 spectral line. Vibration spectra for 10Hz, 15Hz, and 20Hz bearings with and without defects are collected. On the end bearing, the frequency spectrum was recorded. The maximum displacement range was set to 0 to 1 mm, and Hz was employed. The vibration spectra for bearings with and without defects were recorded at 10 Hz, 15 Hz, and 20 Hz. On the end bearing, the frequency spectrum was acquired and recorded.

The experimentation was carried out for deep groove ball bearing TR SB 202, which was mounted on laboratory developed setup. The nomenclature of the bearing are as follows.
d = 7.2 mm, D = 28mm, n = 8 nos. For, N = 600 revolutions per minute.
i.e. shaft excitation frequency of 10 Hz.
D = pitch diameter (mm) and d= diameter of ball (mm) N = Shaft speed, revolutions per minute
n= No of Rolling elements.

Ball pass frequency on the inner race= $(f_{bp}f_i)_{theoretical} = \frac{\pi}{2}\left(\frac{N}{60}\right)\left(1 + \frac{d}{D}Cos\alpha\right)$ (1)

The analytical values of ball pass frequencies are calculated using equation (1).
Using equation (1), the ball pass frequencies at inner race are calculated at various frequencies such as 10Hz, 15Hz and 20Hz and analytical values are found to be 50.29Hz, 75.43Hz and 100.57HZ respectively.

4. EXPERIMENTAL INVESTIGATION OF EFFECT OF FAULT AT INNER RACE OF BALL BEARING

Figure 1 depicts how the experimental setup is designed. The system was initially left running for a few minutes to eliminate any tiny vibrations. To prevent errors brought on by misalignment in the test setup, the alignment was tested using the dial gauge indication. The surface was checked for unevenness with the help of spirit level. This was done to prevent error in measurement of vibration response of bearing housing which included an accelerometer. The bearing housing's top surface was accurately machined to ensure a smooth surface for installing the accelerometer. Using FFT software, the vibration signals were further processed to translate to the frequency domain. The flaw was purposefully introduced into the bearing for research objectives. Using the wire cut EDM technique, the fault was produced at the inner race. A typical spectrum for a bad bearing was collected. To determine whether the frequency spectrum is independent of speed or dependent on speed, the frequency spectrum was recorded for several shaft excitation frequencies, such as 10Hz, 15Hz, and 20Hz. Using an FFT Analyzer, the frequency spectrum for healthy and defective bearings was obtained, and it was compared to the frequency spectrum obtained under comparable conditions from other sources of literature.

The vibration response was obtained for bearing with defects, at 10 Hz and is given in figure 3. Figure 3 shows the rise in the level of amplitude at a shaft excitation frequency of 10 Hz. There is some high amplitude observed at harmonics of shaft excitation frequency viz. 20 Hz, 30 Hz, 40 Hz, and 50 Hz, respectively. They are observed may be because of mechanical looseness. But the amplitude of these peaks is less than the peak observed at shaft excitation frequency, i.e., 10 Hz.

The Vibration response is also obtained for healthy bearing at 900 rpm (excitation frequency is 15 Hz) as shown in figure 4, the response shows maximum amplitude at operating frequency of 15 Hz.

There are some peak amplitude observed at harmonics of shaft excitation frequency viz. 30 Hz, 45 Hz, 60 Hz and 75 Hz respectively which are multiples of shaft excitation frequency. The frequency spectrum is also obtained for healthy bearing at its excitation frequency of 20 Hz.

It is evident from the responses that the vibration spectrum is unaffected by operating speed and frequency. The Vibration response for a healthy bearing at the excitation frequency of 15 Hz, and it is displayed in fig. Figure 4 depicts the amplitude peak at a 15 Hz shaft excitation frequency.

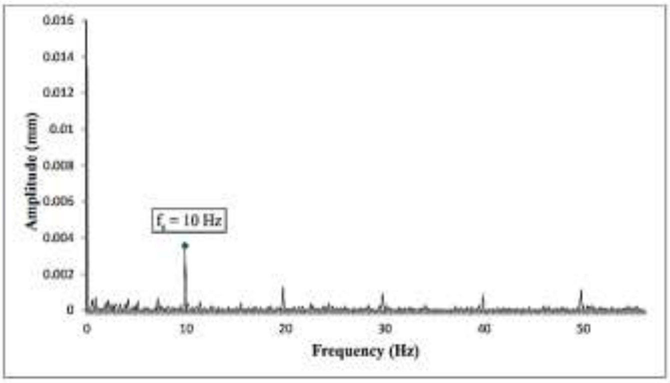

Figure 3. Vibration response for bearing without defect at 10Hz

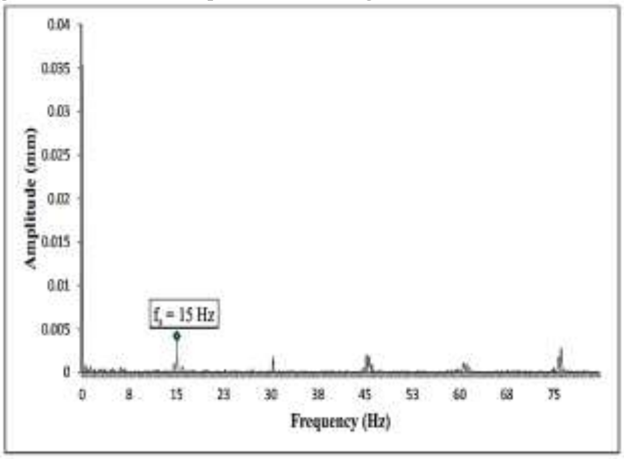

Figure 4. Vibration response for healthy bearing at 15 Hz

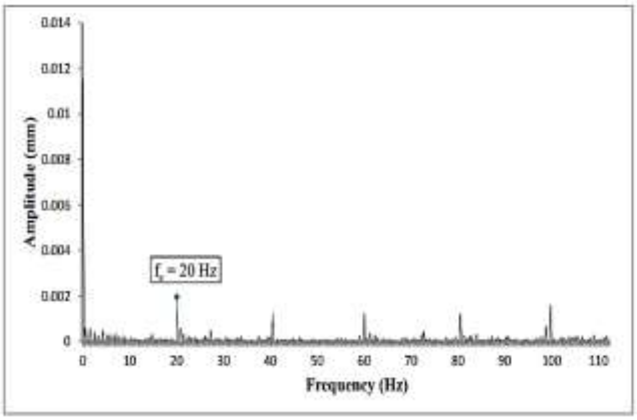

Figure 5. Vibration response for healthy hearing at 20 Hz

From the above spectrum it is clearly concluded that, the vibration spectrum is independent of operating speed and frequency.

4.1 Frequency Spectrum with Defective Bearing

Vibration response for defective bearing at 10 Hz is shown in figure 6. The peak is observed at shaft excitation frequency (fs) and its harmonics. The highest peak in amplitude is observed at 49.75 Hz which is $(f_{bp}f_i)_{experimental}$ as shown in figure 6. It is close to the value of $(f_{bp}f_i)_{theoretical}$ i.e. 50.285 Hz which is obtained from theoretical value.

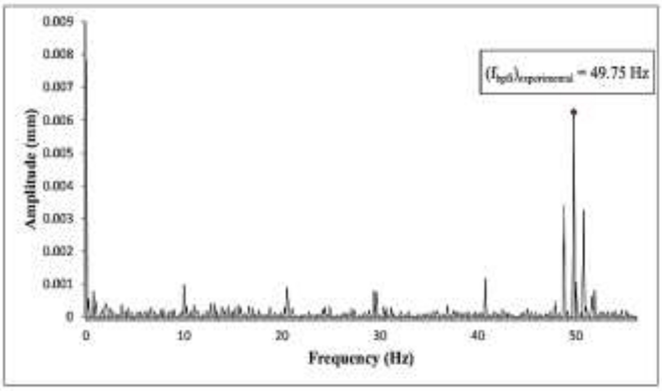

Figure 6. Vibration response for defective bearing at 10 Hz

It is also observed from figure 6 that, there are equally spaced side bands around peak at 49.75 Hz i.e. $(f_{bp}f_i)_{experimental}$ which shows the presence of fault in ball bearing at inner race. Figure 7 shows frequency spectrum for defective bearing at 15 Hz.

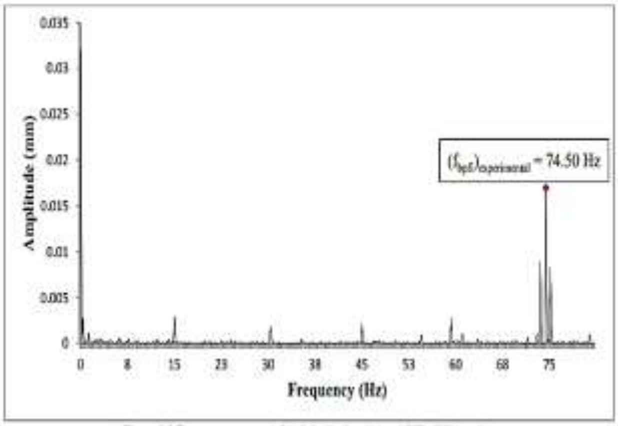

Figure 7. Vibration response for defective bearing at 15 Hz

The highest amplitude peak is observed in figure 7 is at 74.50 Hz which is close to $(f_{bp}f_i)_{theoretical}$ i.e. 75.428 Hz as obtained from analytical value as mentioned earlier. Percentage error is calculated and it comes out to be 1.23% for defective bearing at 15 Hz. Percentage error of 1.23% is within permissible limit.

Figure 8 shows frequency spectrum for defective bearing at 20 Hz and the maximum amplitude is observed at shaft excitation frequency (fs) i.e. 20 Hz and its harmonics. However, due to looseness, smaller peaks are also observed in frequency spectrum.

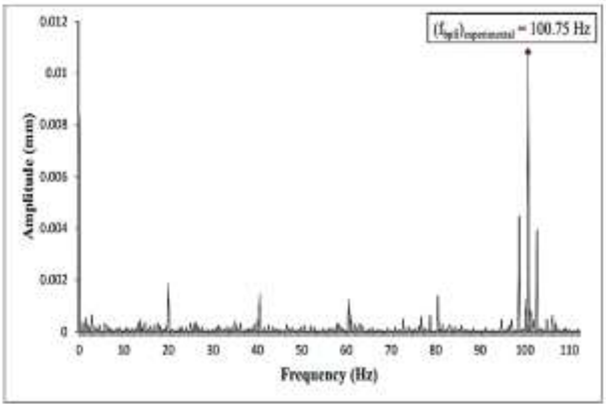

Figure 8. Vibration Response for defective bearing at 20 Hz

The highest amplitude peak is observed in figure 8 at 100.75 Hz which is $(f_{bp}f_i)_{experimental}$. It is close to the value of $(f_{bp}f_i)_{theoretical}$ i.e. 100.57 Hz as obtained from table 1. Percentage error is calculated and it comes out to be 0.17% for defective bearing at 20 Hz. Percentage error of 0.17% is within acceptable limit.

5. RESULT AND DISCUSSION

For both healthy and damaged bearings, experimental results for spectrum have been acquired. This is also the experimental observation for both healthy and faulty bearings. Figure 9 displays the together frequency spectrum that was recorded for the test bearing. It compares the outcomes of bearings with healthy and defective bearings. The peak is seen at shaft excitation frequency (fs), or 10 Hz. At 49.75 Hz, which is near 50.285 Hz, or the value of $(f_{bp}f_i)_{theoretical}$ of the test ball bearing, a substantial rise in amplitude is also clearly seen. Around $(f_{bp}f_i)$, experimental, equally spaced side bands are seen.

The presence of fault in bearing at inner race, will cause sidebands around the bearing tone. Figure 10, shows the combined vibration response for bearing with and without defects. spectrum for healthy and defective bearing at 15 Hz. At 900 rpm, the peak is observed at shaft excitation frequency i.e. 15 Hz and smaller peaks are observed at its harmonics at 2x, 3x, 4x and 5x.

Also high rise in amplitude is clearly observed at 74.50 Hz which is close to 75.428 Hz i.e. the value of $(f_{bp}f_i)_{theoretical}$ of the test ball bearing at 15 Hz.

Vibration signals are also acquired for healthy and defective bearing at 20 Hz, and it shows rise in amplitude at 100.75 Hz which is close to 75.57 Hz i.e. the value of $(f_{bp}f_i)_{theoretical}$ of the test ball bearing at 20 Hz.

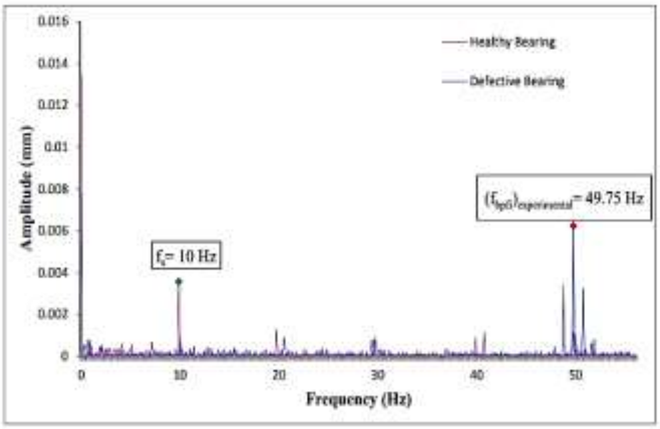

Figure 9. Vibration Response for healthy and faulty bearing at 10 Hz

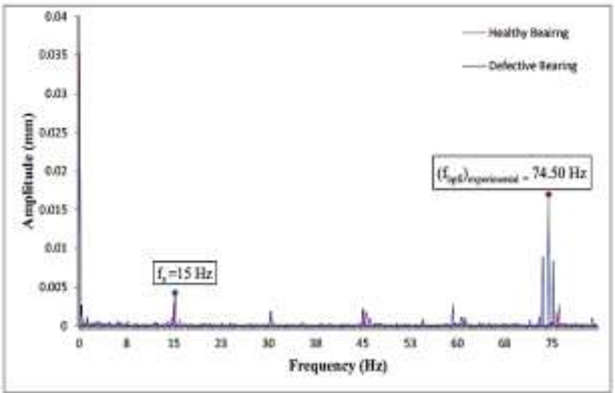

Figure 10. Vibration Response for healthy and faulty bearing at 15 Hz

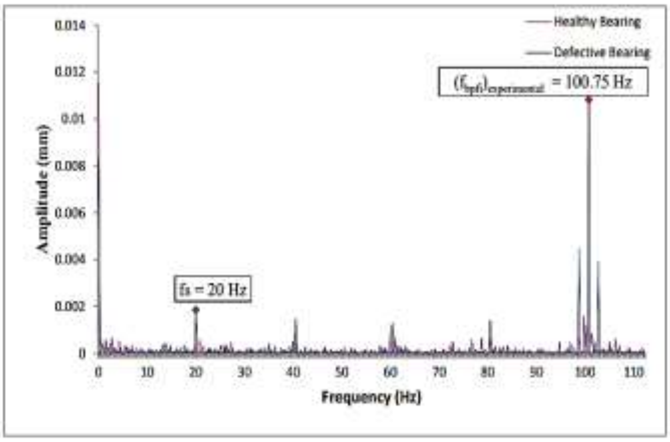

Figure 11. Vibration Response for healthy and faulty bearing at 20 Hz

6. CONCLUSION:

A laboratory experimental model has been designed and built to investigate the impact of the ball bearing's inner race fault on the frequency spectrum. To comprehend the impact of these defects at various frequencies, an experimental investigation was carried out using a triaxial shear type accelerometer and, indeed, the photon+ (Brüel & Kjaer) ultraportable dynamic signal analyzer for data acquisition. The frequency spectrum is extracted for both healthy and defective bearings. The shaft excitation frequency and its harmonics are located where the amplitude peak is visible in the frequency spectrum for a healthy bearing (2x, 3x, 4x, and 5x multiple of shaft excitation frequency). The experimental frequency ($f_{bp}f_i$) does have the largest peak in the frequency spectrum for faulty bearings. It is close to the value of $(f_{bp}f_i)_{theoretical}$. At various shaft excitation frequencies, percentage errors for the $(f_{bp}f_i)_{experimental}$ are within the acceptable range. At $(f_{bp}f_{i})_{experimental,}$ the sidebands with equal spacing are also noticeable around the highest peak. This implies that there is a problem with the ball bearing inner race. Generally, the amplitude values of the spectrum of the inner raceway defect is much less than that of outer race defects. The reason behind this is due to the complexity in transmitting the signals to the acceleromer due to structural interfaces such as ball, oil film, outer race, and bearing housing.
The limitations of current experimental work are the presence of other defects in the system, which will slightly alter the actual vibration response of the bearing defect. Maximum care is taken while fabricating the set up to avoid the presence of such other defects.

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