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Original Article

Experimental Vibrational Analysis of Antifriction Bearing

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Abstract *- Bearing plays a vital role in rotating machinery. Many authors studied failure analysis of the bearing experimentally and numerically by creating defects in the outer race, inner race, and rolling element and keeping clearance between shaft-sleeve, sleeve-inner race and outer race-pedestal block. This causes an increase in vibration in the machine as a whole. In this paper square hole with different depths of cut is created on the ball by EDM, and also clearance is developed between the ball and the outer race. Frequency spectra are obtained by FFT Analyzer. The experiment is conducted with different speeds and a single load. Experimental results show that the peak amplitude of vibration increases with an increase in speed. Peak vibrational amplitudes also increase with the increase in the depth of cut for the same clearance. Clearance has little effect in changing vibrational amplitudes. Harmonics in terms of ball spin frequency is clearly observed.*

Keywords **-** *Ball spin frequency, Clearance, Depth of cut, FFT, Vibrational analysis.*

1. Introduction

Rolling element bearings are chiefly used in rotating machinery across various industries, which include aerospace, construction, mining, steel, paper, textile, railways, and renewable energy. The damage and failure of rolling element bearings is a dominant factor that contributes to machinery breakdown, consequently causing significant economic losses and even loss of human lives. Ball bearings are also used for moderately loaded and high-speed applications, whereas Roller bearings are suitable for supporting heavier loads applications. Moreover, radial bearings are preferred when the load applied is in the radial direction, and to carry axial load, thrust bearings are preferred. But, most of the ball bearings can support both radial and thrust load. Bearing defects may be categorized as distributed or local. Distributed defects include surface roughness, waviness, misaligned races, and off-size rolling elements. They are usually caused by manufacturing errors, improper installation or abrasive wear [1, 2]. Local defects include cracks, pits and spalls on the rolling surfaces. The dominant mode of failure of rolling element bearings is spalling of the races or the rolling elements, caused when a fatigue crack begins below the surface of the metal, and they propagate towards the surface until a piece of metal breaks away to leave a small pit or spall [3]. Whenever a local defect on an element interacts with its mating element, abrupt changes in the contact stresses at the interface result, which generates a pulse of very short duration. This pulse produces vibration and noise, which can be monitored to detect the presence of a defect in the bearing. Even when a local defect grows, it becomes distributed one, generating a more complex signal with strong non-stationary contents. Vibration monitoring has now become a well-accepted part of many planned maintenance regimes and relies on the well-known characteristic vibration signatures which rolling bearings exhibit as the rolling surfaces degrade. However, in most situations bearing vibration cannot be measured directly, and so the bearing vibration signature is modified by the machine structure. This situation is further complicated by vibration from other equipment on the machine, i.e. electric motors, gears, belts, hydraulics, structural resonances etc. This often makes the interpretation of vibration data difficult other than by a trained specialist and can, in some situations, lead to a misdiagnosis, resulting in unnecessary machine downtime. Fault detection in the rolling element bearing was studied experimentally and theoretically by Taha and Dung [4]. In an experimental analysis single point fault on the outer raceway is created, i.e. 2mm in diameter and depth, by using EDM.RMS and peak-to-peak signal parameters are studied and observed to increase the level of vibration in the high-frequency range of the spectrum. From the reading, it is seen that there is a big difference between defective and healthy bearing readings. A review of vibration and acoustic measurement methods for the detection of defects in rolling element bearings was carried out by Tandon and Choudhry [1]. Detection of both localized and distributed categories of defect is considered. They concluded that vibration measurement in the frequency domain has the advantage over the time domain because it can detect the location of the defect. Patel et al. [5] have done theoretical and experimental vibration studies of dynamically loaded deep groove ball bearings having local circular shape defects on either race. It was seen that vibrationenhances the presence of local defects in the outer race in comparison to the inner race. Acoustic emission method was used to study experimentally defect diagnosis in rolling bearings [4]. Results clearly show that whenever the roller roll over the defect point, acoustic emission is released, and acoustic emission signal is periodic impact characteristics. Further reading showsthat intensity of acoustic emission increases with increasing load and speed. Experimentation And validation analysis of fault diagnosis of ball bearing related to rotor system was Studied by Tarle et al.[6]. They concluded that the amplitude at BPFO is higher than BPFI and BSF. And is least in BSF. Kulkarni and Wadkar [7] have analyzed the effect of surface roughness on the response of the outer race of the ball bearing. They found that at a constant speed and constant load with different defect sizes on the outer ring, amplitudes of vibration vary with an increase in the defect size. Tomovic et al. [8] studied the vibration response of rigid rotors in unloaded rolling element bearings. By the application of the defined modal, the parametric analysis of the effect of internal radial clearance value and number of rolling elements' influence on rigid rotor vibrations in unloaded rolling element bearings were performed. They concluded that the BPF linearly increases with the increase of the number of rolling elements. With the increase of the internal radial clearance, the value of amplitude increases linearly. The increase gets much bigger as the total number of rolling elements decreases. Chavan et al. [9]. This paper gives the importance of the vibration monitoring technique and its implementation in the sugar industry and also explains some case studies of the sugar industry related to bearing vibration problems. The case studies show that whenever looseness is present in the bearing assembly, then the predominant peak has occurred at 2*RPM of the machine. The result shows that whenever a bearing fault occurs, there is an increase in vibration to a higher level. Based on the above literature review, the objective of the paper is to study experimentally the vibration response of faulty bearings with square holes with different depths of cut on the ball and its comparison with healthy bearing.

2. Experimentation

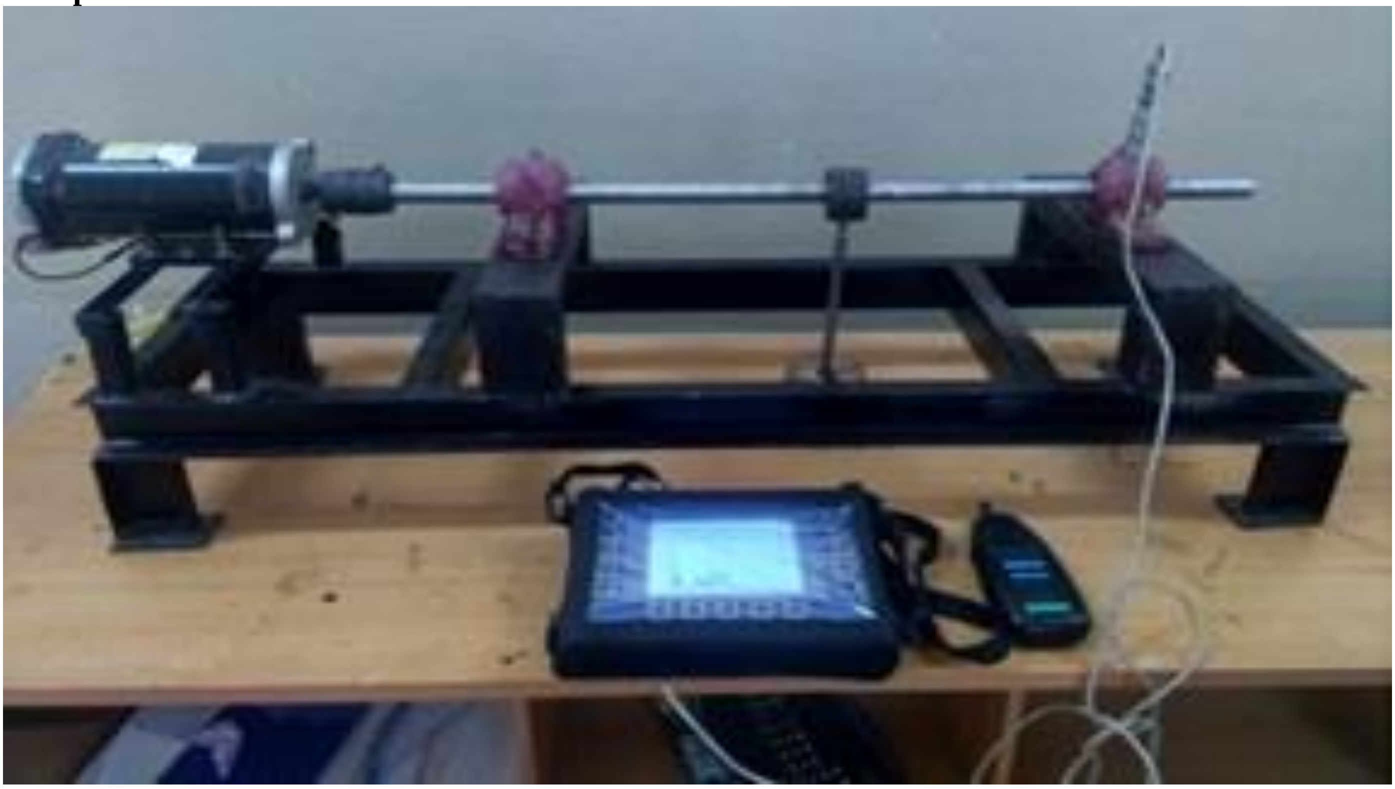

Fig. 1 Experimental setup

The experimental setup shown in Fig. 1 is designed and fabricated to investigate the vibration characteristics of ball bearings. The experimental setup has an electrical motor that varies speed from 0 rpm to 2000rpm. Motor is connected to the shaft through a coupling. Test specimen of healthy bearing and faulty bearing is fitted on the non-driving end of the shaft. The fault on the ball, i.e. a rolling element of self-aligned bearing, is created by electric discharge machining (EDM). A square hole of different depths, i.e.1 mm, 1.5mm, 2mm and 2.5mm, is produced. When the bearing is mounted in the sleeve, a clearance of 0.04mm is maintained by the feeler gauge between the ball and the outer race. The whole setup is placed on a rubber cushion to reduce vibrations. Experiments on the healthy and faulty bearings are carried out at different speeds and single loads. Load is acted on the shaft in the middle. Vibration analysis is investigated through an FFT analyzer and accelerometer mounted in a vertical position. The parameters chosen for experimentation are shown in Table 1, and the Specification of the used Self-aligned 1206K bearing is given in Table 2.

Table 1. Parameters for experimentation

Depth of cut on the ball	Spacing between the ball and outer race	Speed of shaft in rpm
Nil, i.e. healthy bearing	Nil	500,700,900,1100
1mm	0.04mm	500,700,900,1100
1.5mm	0.04mm	500,700,900,1100
2mm	0.04mm	500,700,900,1100
2.5mm	0.04mm	500,700,900,1100

Table 2. Properties of Self-aligning ball bearing 1206K

Bearing number	1206k
Size in mm	30x62x16
Bore Diameter in mm	30
Outer diameter in mm	62
Width in mm	16
Number of balls	28
Ball Diameter in mm	7.94

3. Result and Discussions

3.1. Frequency Response of Healthy Bearing

Fig.2 and Fig.3 show the frequency spectrum of healthy bearing for 700rpm and 900rpm, respectively. In these figures, the peak amplitude of vibrations is 1.12mm/s and 1.28mm/s.

In Fig. 2, harmonics of vibrations are observed and are present at shaft frequency (N) of 6*N to 8*N. Similarly in Fig. 3, it is at 2*N, 4*N and 7*N. They are small in magnitude and may be due to unbalances present in the system.

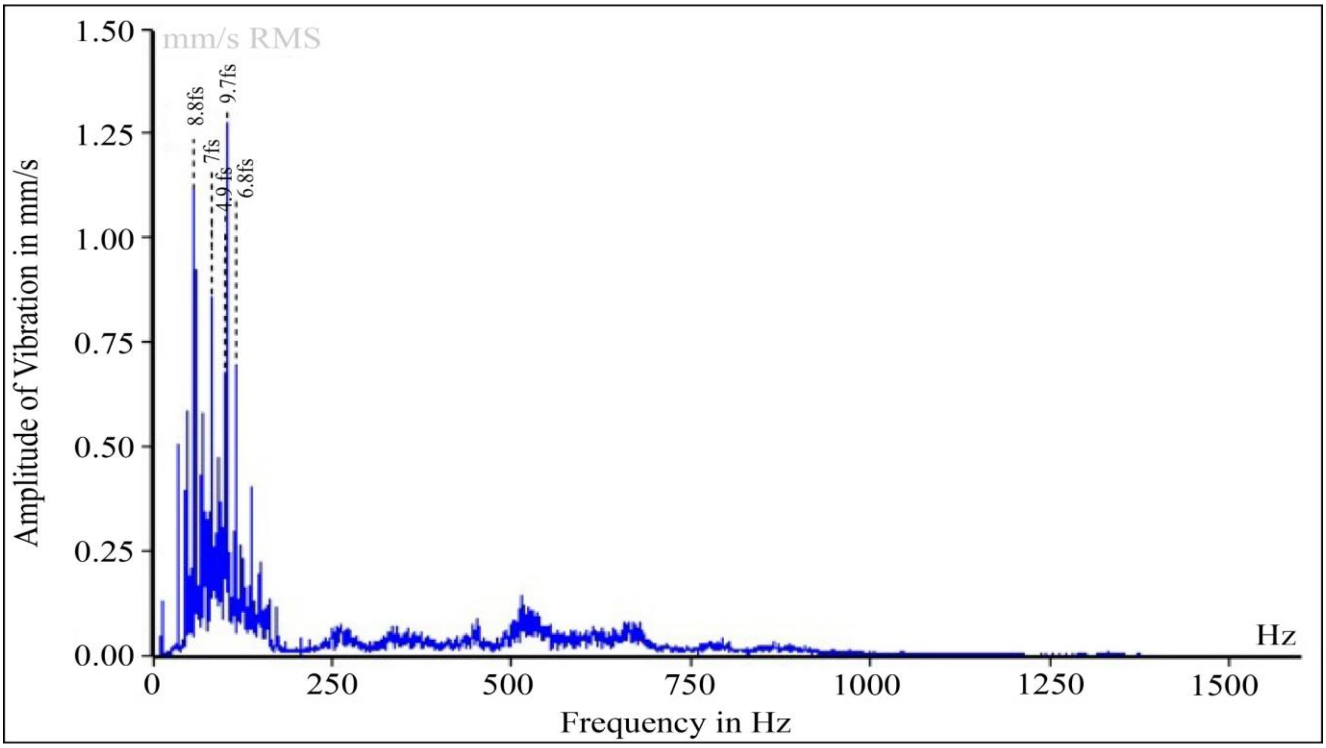

Fig. 2 Frequency spectrum for healthy bearing at 700rpm

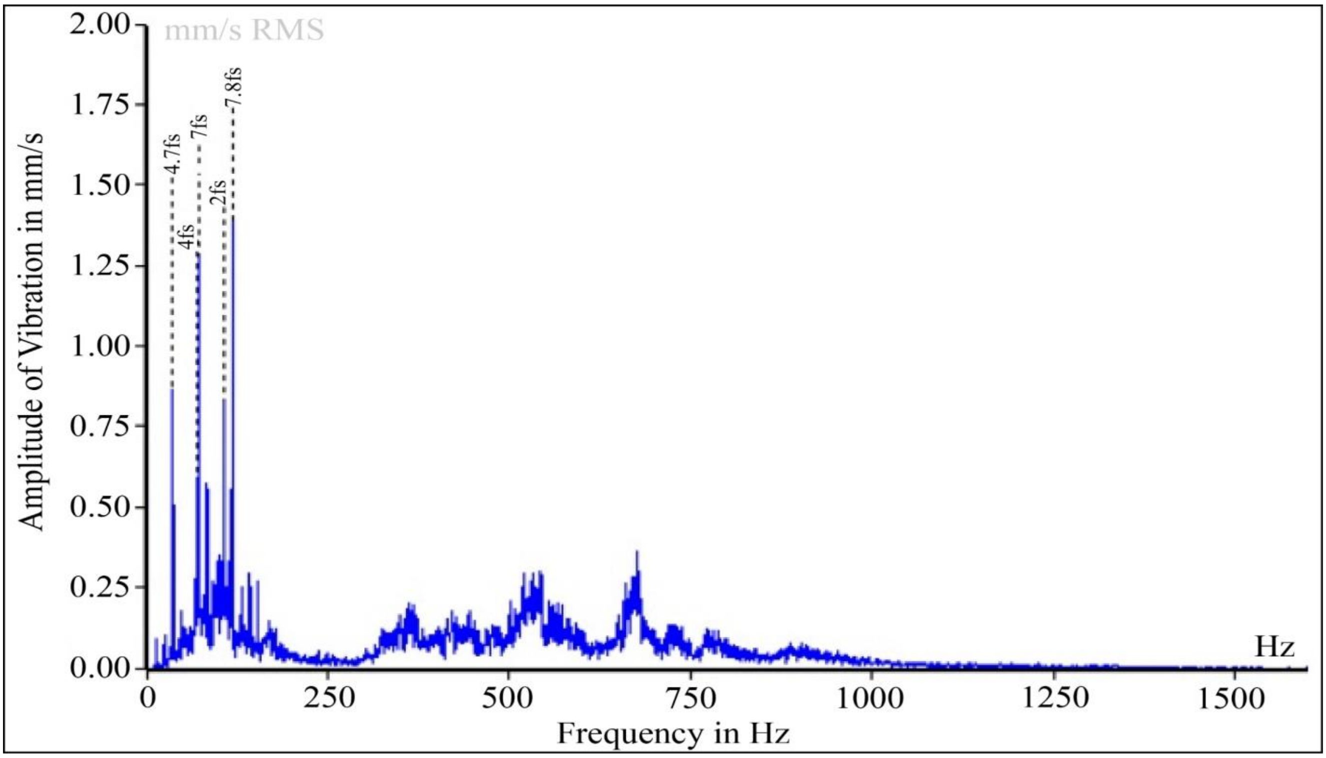

Fig. 3 Frequency spectrum for healthy bearing at 900rpm

3.2. Frequency Response of Faulty Bearing with 1mm Depth and 0.04mm Clearance

Frequency spectra for this case under the speed of 700rpm and 900rpm are shown in Fig. 4 and Fig. 5. Vibration amplitudes increase with shaft speed. It is 1.25mm/s at 700 rpm and 4.31mm/s at 900rpm. At shaft speed(fs) at 700 rpm (11.67Hz) and all defect frequency(fbd) 33.8Hz amplitude of vibration is 0.082mm/s. Frequency spectra associated with harmonic peaks are as fs+1.03fbd, fs+1.37fbd, fs+1.72fbd and fs+2.76fbd at a shaft speed of 700rpm. Likewise, harmonics are associated with other shaft speeds, as shown in Fig. 5.

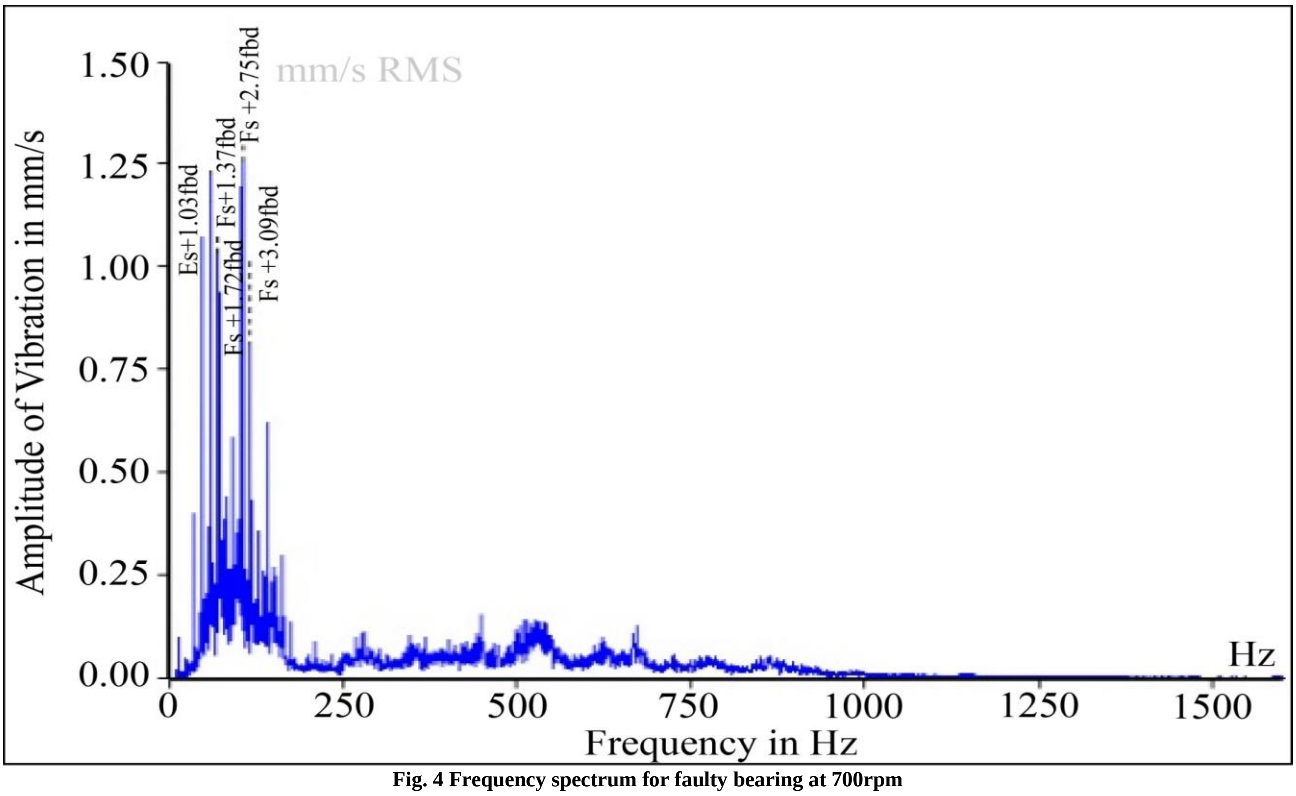

Fig. 4 Frequency spectrum for faulty bearing at 700rpm

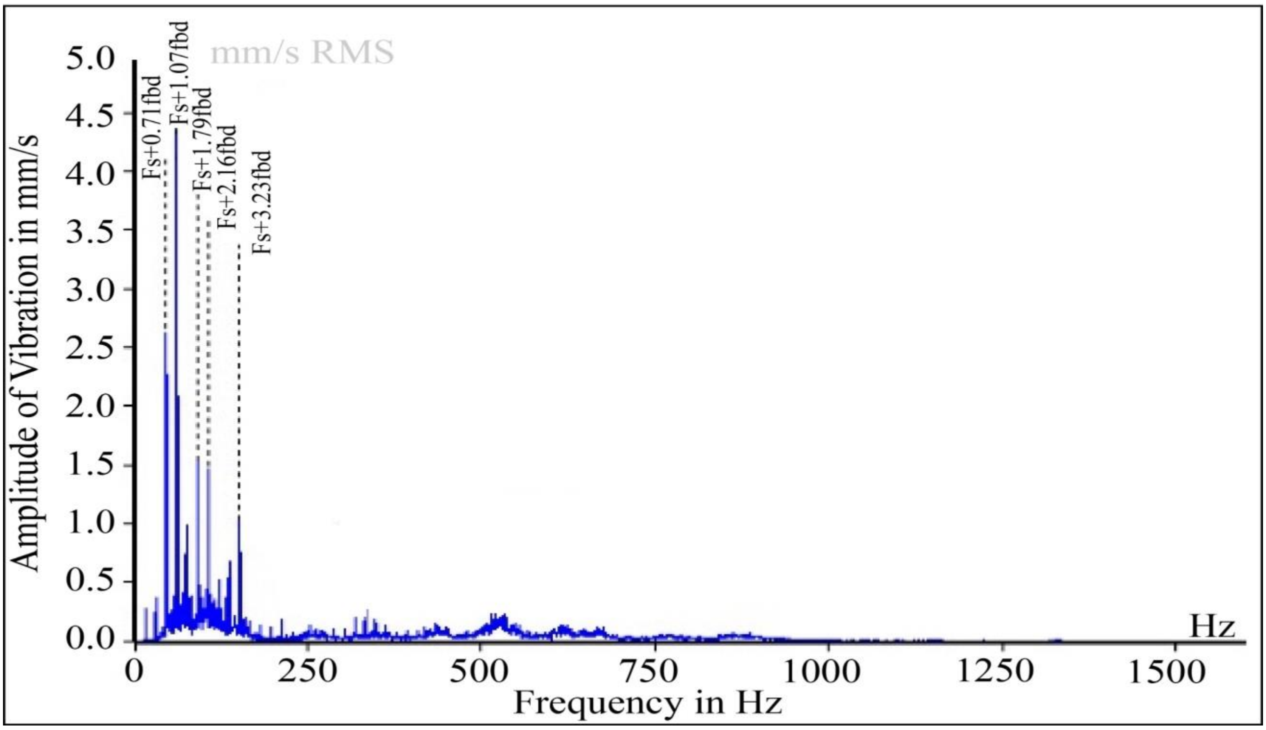

Fig. 5 Frequency spectrum for faulty bearing at 900rpm

3.3. Frequency Response of Faulty Bearing with 1.5mm Depth and 0.04mm Clearance

FFT spectra for faulty bearings with a depth of 1.5mm and 0.04mm clearance are shown in Fig.6 to Fig.7.Here also, vibration harmonicswith side band are observed. At 900rpm i.e.15Hz, the ball defect frequency is 42.15Hz, at which the amplitude of vibration is 0.340mm/s. Harmonics obtained are fs+0.75fbd,fs+1.12fbd,fs+2.23fbd and fs+2.59fbd at 900rpm. The magnitude of the vibration level is higher than healthy and faulty with a 1mm depth.

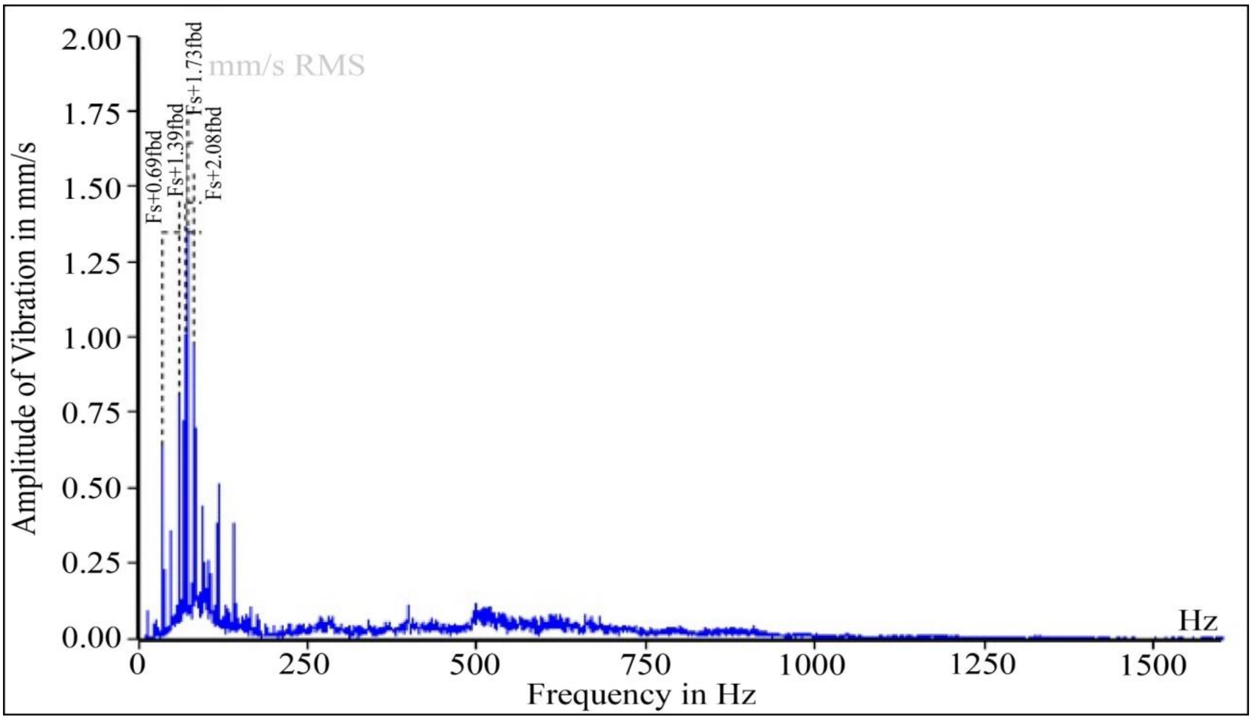

Fig. 6 Frequency spectrum for faulty bearing at 700rpm

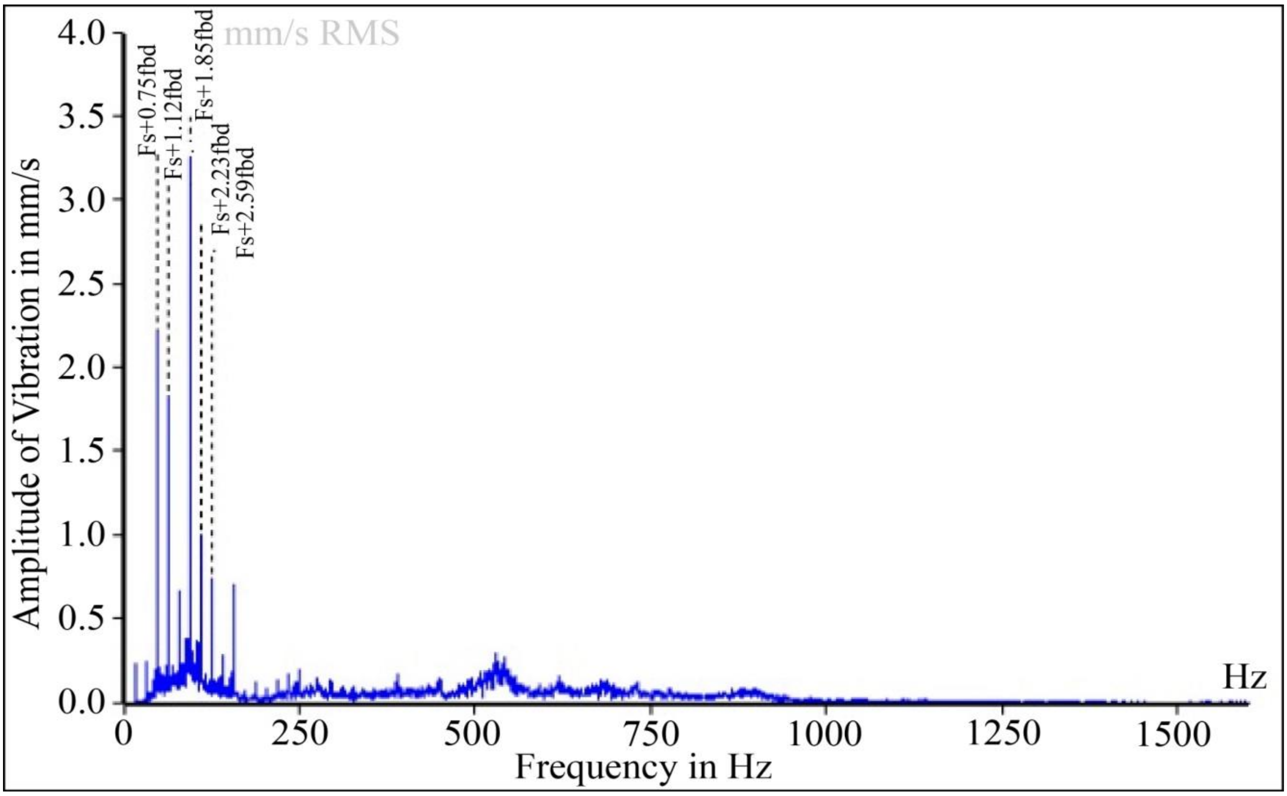

Fig.7 Frequency spectrum for faulty bearing at 900rpm

3.4. Frequency Response of Faulty Bearing with 2mm Depth and 0.04mm Clearance

Amplitude of vibration versus frequency is shown in Fig.8 and Fig.9.Ball defect frequency for shaft speed 700rpm(11.67Hz) is 33.8Hz. At this frequency amplitude of vibration noted is 0.07mm/vibration harmonic observed is fs+1.06fbd, fs+1.41fbd, fs+1.67fbd, fs+2.11fbd and fs+2.36fbd at 700rpm. Similarly vibration modulation is shown in Fig.9

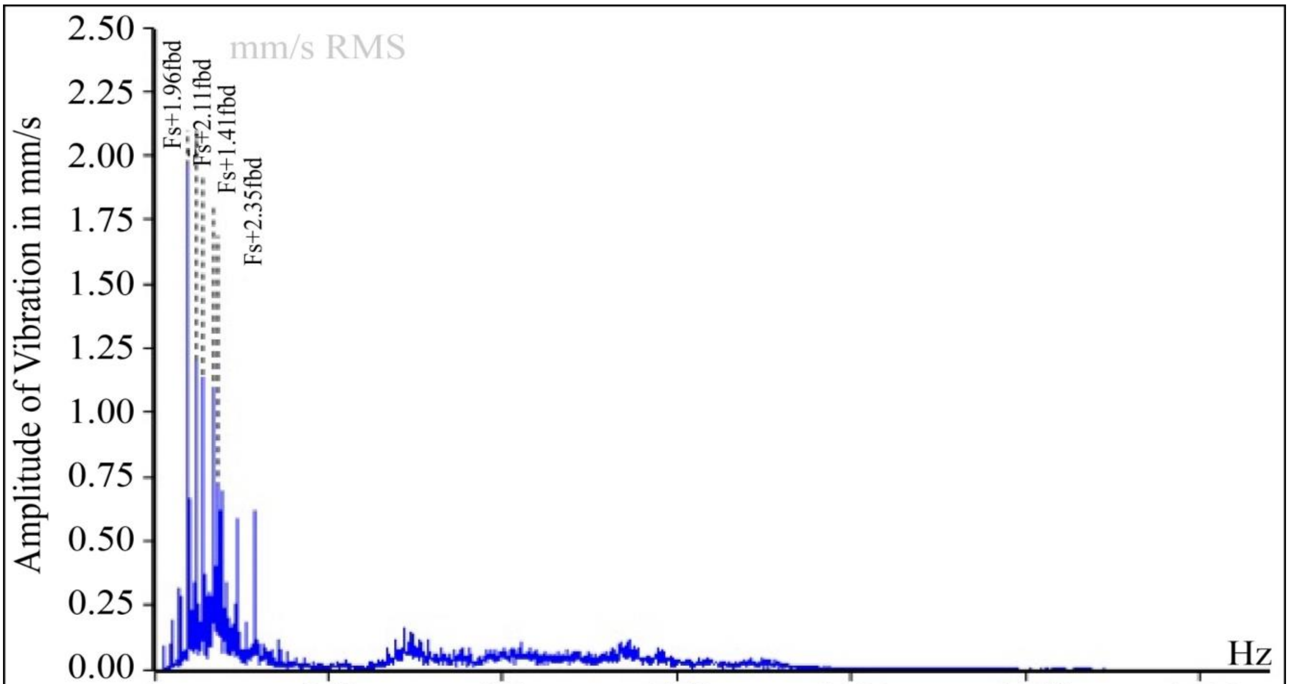

Fig. 8 Frequency spectrum for faulty bearing at 700rpm

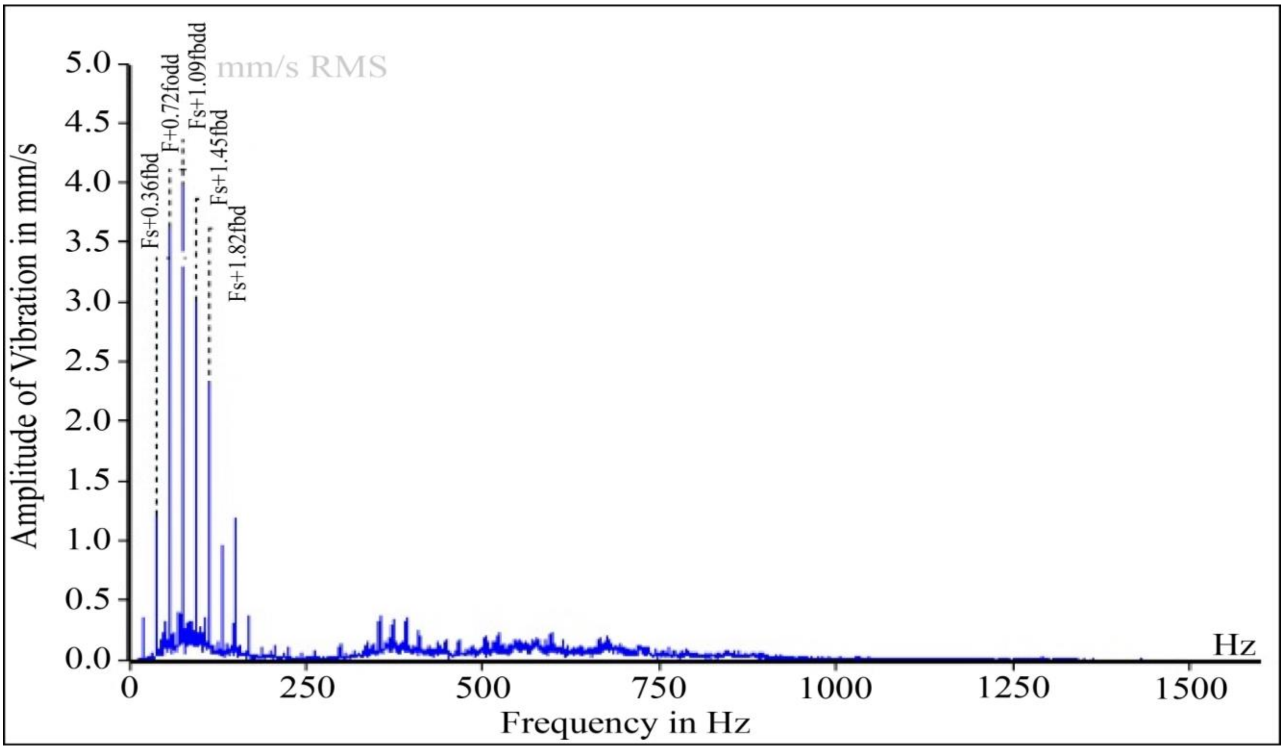

Fig. 9 Frequency spectrum for faulty bearing at 1100rpm

3.5. Frequency Response of Faulty Bearing with 2.5mm Depth and 0.04mm Clearance

FFT spectra are shown in Fig.10 and Fig.11.Consider for speed of 1100rpm or 18.34Hz shaft frequency at which the ball defect frequency is 51.53Hz. At this fbd, the amplitude of vibration is 0.05mm/s. Frequency harmonics at this speed are fs+0.72fbd, fs+1.08fbd, fs+1.44fbd, fs+1.8fbd and fs+2.17fbd. But with speed magnitude of vibration increases.

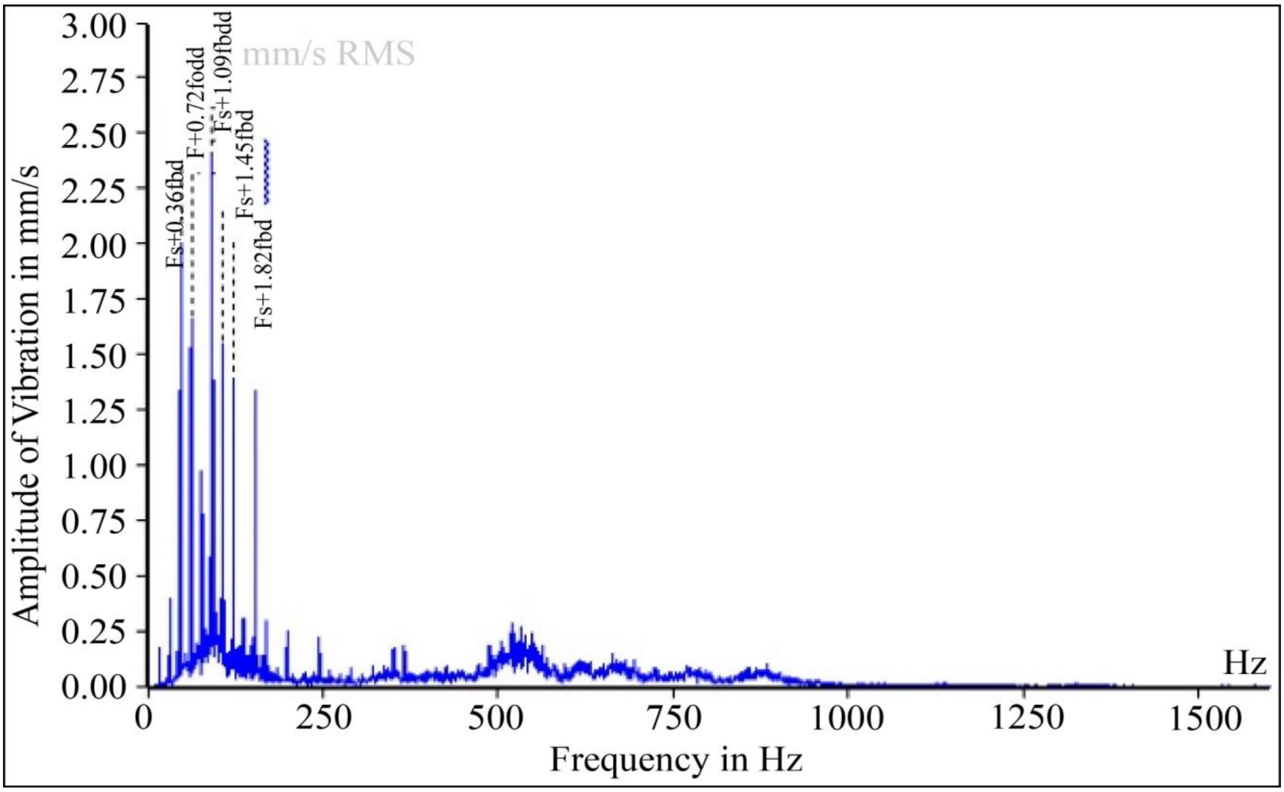

Fig. 10 Frequency spectrum for faulty bearing at 900rpm

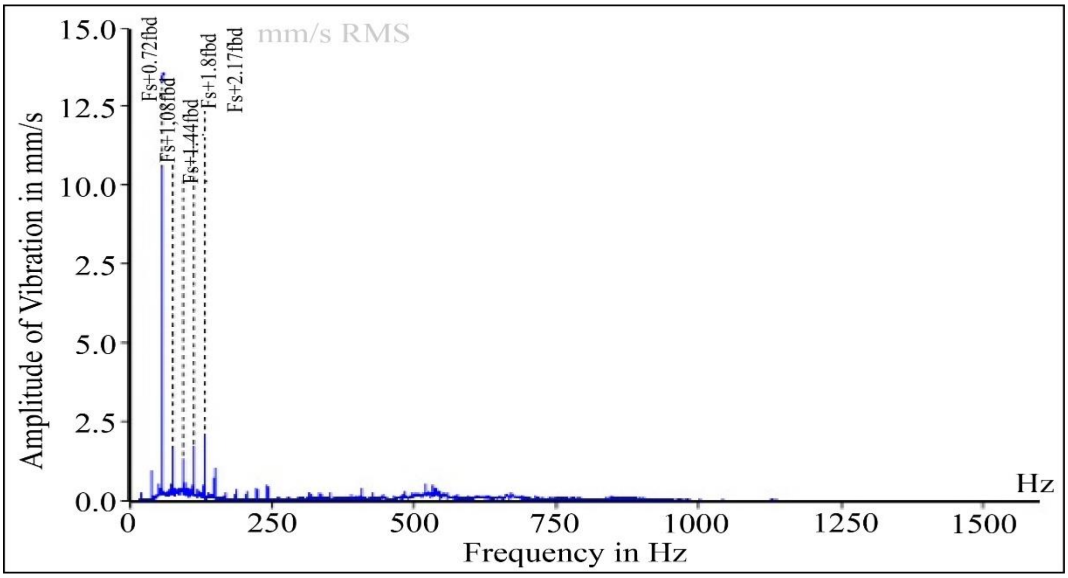

Fig. 11 Frequency spectrum for faulty bearing at 1100rpm

4. Comparison of Amplitude of Vibration with Speed For Healthy and Faulty Bearings

In Fig.12 and Fig.13, the peak amplitude of vibration of healthy bearings and various faulty bearings are compared at different speeds. It can be observed that the vibration level for the healthy bearing is the least, and whatever magnitude is seen is due to unbalances in the system. In all cases, the amplitude of vibrations increases with speed. Also, the increased depth of cut in the ball causes vibrations to rise, and different harmonics can be seen at different ball defect frequencies.

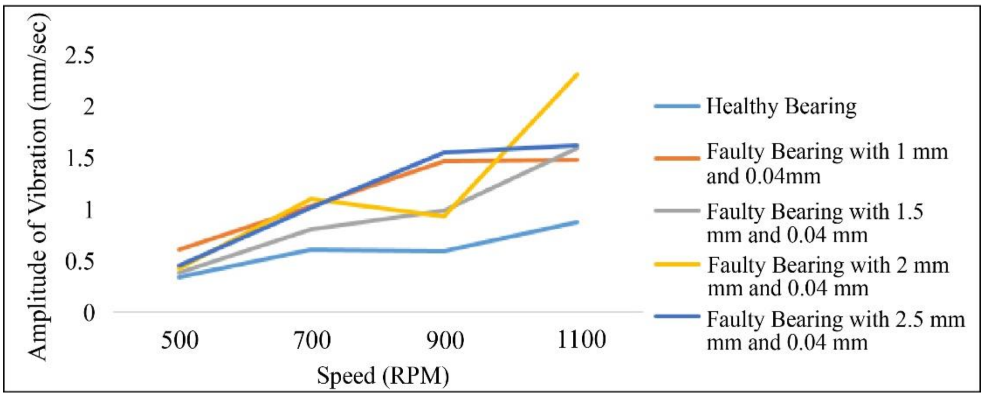

Fig. 12 Amplitude of vibration Vs Speed

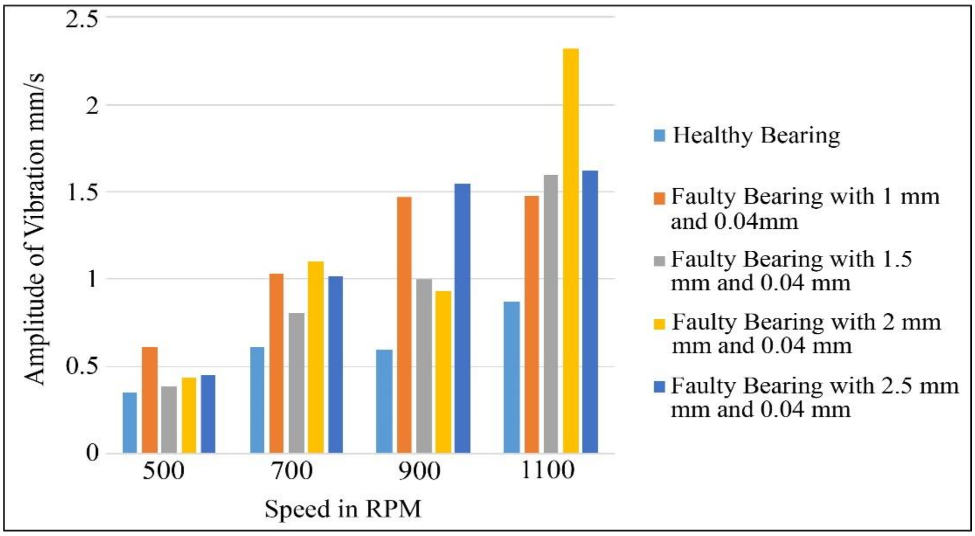

Fig. 13 Variation of amplitude of vibration with speed

5. Conclusion

In this paper, experimental vibrational analysis is done for healthy and faulty self-aligned ball bearing with a clearance of 0.04mm between the ball and outer race. The following conclusions are drawn-

- In a healthy ball bearing, the amplitude of vibrations noted by the FFT analyzer is small in magnitude. It is also accompanied by harmonics in terms of shaft frequency.
- Self-aligned faulty bearing with 1mm and 0.04 clearance is used; there is a rise in the magnitude of peak vibrations as compared to a healthy bearing. Also, harmonics of vibration are noted at ball spin or ball defect frequency.
- For depth of cut of 1.5mm, 2mm, and 2.5mm on the ball induced more vibration levels as the depth of cut increased.
- Ball bearing with a depth of cut 2.5mm has the highest amplitude of vibration at all speeds. The healthy bearing has the least vibration.

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