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(Research Article)

Design and Development of Test Rig for Vibration Measurement of Shaft-Rolling Bearing System

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Abstract

Rolling element bearing is a most critical part of any rotating machinery. The machine could be unsafe if any fault occurs in the bearing elements due to fatigue, manufacturing errors and operating conditions. Due to sudden failure of bearing the machine may cause serious accident, economical loss and time. Therefore, it is necessary to predict the faults in bearing well in advance for preventive maintenance of machinery. The vibration based condition monitoring is most effective method for identifying fault in the bearing during its operation. In this paper a test rig has been designed and developed for vibration measurement of shaft-rolling bearing system. The development of test rig includes design and manufacturing of shaft, selection of support and test bearing along with suitable pedestal, selection of belt, pulley and motor. The vibrations data has been captured by accelerometer for test bearing in presence of normal and defective bearing outer raceway. The measured vibrations has been analysed in frequency domain to predict the fault characteristics. The vibrations generated by bearing depends on bearing geometry, load and shaft speed. It has been also study the effect of defect size, load and shaft speed on vibration response of healthy and defective bearing.

Keywords: Deep groove ball bearing, test rig, vibration analysis, local defect, design and analysis, defect frequency.

1. Introduction

The rolling element bearing is widely used in industrial machinery. The main function of bearing is to support the load and permit relative motion. Due to longer period of use for rolling bearing it may generate local defects viz, crack, pits, spalls, etc., due to fatigue of mating parts of bearing. Even defect free bearing may produce vibrations due to time varying contact forces between mating parts or due to varying compliance effect [1]. The nature of vibrations response changes due to absence and presence of defect on contacting elements of bearing.

Due to failure of bearing in the machine, it will cause sudden break down or in some cases possible catastrophic failure of machine. The failure of bearing in the machine will cause shut down the production process and accident to the operating personnel. The various condition monitoring techniques are used for identifying health of machinery. The main techniques for condition monitoring of machinery are vibration measurement, temperature measurement, wear debris and sound measurement which are used in the industry. The vibration measurement is a most effective technique and widely used in industry for fault identification in a rolling element bearing.

The main elements of rolling bearing are inner race, outer race, cage (Retainer) and rolling element (ball / roller). The mainly two types of defect (Localized and Distributed defect) are observed on elements of bearing. The bearing faults like spalls, pits, dirt, dent, and crack on the rolling surfaces, brinelling and contaminations in lubricant are considered as localized defect while surface roughness, manufacturing errors, off size rolling element, raceway or rolling element waviness and misaligned races are classified as a distributed fault in a bearing.

In vibration analysis the vibration signals are captured from machine parts and analysed in time and frequency domain. The presence of fault in the element of bearing indicates peak in the vibration spectrum. However amplitude of vibration peak is depends on size, shape, location and type

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of defect. The characteristics defect frequencies for inner, outer and cage are termed as ball pass frequency for inner / outer (BPFO / BPFI), ball spin frequency (BSF) and fundamental train frequency (FTF) respectively. These characteristics defect frequencies are calculated using following eqs. (1-4). The defect frequency of bearing elements depends on the shaft rotational speed and bearing geometry [2].

Ball pass frequency for outer race (BPFO)

$$\mathrm{BPFO}=\frac{n}{2}f_s\left(1-\frac{d}{D\cos\varphi}\right) \tag{1}$$

Ball pass frequency for inner race (BPFI)

$$\mathrm{BPFI}=\frac{n}{2}f_s\left(1+\frac{d}{D\cos\varphi}\right) \tag{2}$$

Ball Spin frequency (BSF)

$$\mathrm{BSF}=\frac{n}{2}f_s\left(1-\frac{d^2}{D^2\cos^2\varphi}\right) \tag{3}$$

Fundamental Train frequency (FTF)

$$\mathrm{FTF}=\frac{1}{2}f_s\left(1-\frac{d}{D\cos\varphi}\right) \tag{4}$$

Many of the researchers have studied theoretically and experimentally the vibration response of healthy and defective bearing in presence of varying defect size, shape and its type [3-4]. It is required to improve the existing proposed dynamic model for the better prediction of fault in bearing elements through vibration monitoring technique. In theoretical model researchers have considered additional deflection or excitation force of rolling element due to interaction of defective races with rolling element [5-6]. A test rig has been designed and developed to perform the experiment and measure the vibrations generated by defective and healthy bearing for validation of theoretical results and bearing condition monitoring. The developed test rig is also useful for testing the bearing vibrations when it is not possible to check on site.

The vibration measurement of healthy and defective deep groove ball bearing (SKF BB1B 420206) have been captured through magnetic piezo-electic accelerometer with a sensitivity of 100 mv/g is used. The accelerometer is connected to the vibration analyzer and the output of analyzer is connected to the computer for further interpretation of the data in time and frequency domain. The theoretical characteristics defect frequencies for elements of bearing have been calculated using eqs. (1-4) which are verified with experimental defect frequencies obtained using performed experiment. The developed test rig must have sufficent stiffness for support structure of system to avoid unnecessary vibrations due to some other sources. It is only possible to see clearly the dominent frequencies and vibration amplitude for test bearing if the test rig supports are rigid in nature.

2. Design of Test Rig for Vibration Monitoring

This section deals with design and development of test rig parts for the vibration measurement of shaft-bearing system. The main parts of test rig are electric motor, belt, shaft, support bearing, test bearing, test bearing housing, support bearing housing, table, shaft pulley and motor pulley. In design of test rig includes selection and design of test rig parts as per the following procedure.

2.1 Selection of Electric Motor: In the present study a single phase AC motor has been selected with following specifications:

Power Rating: 1 HP, Voltage: 220 V, Speed: 2880 RPM, Current: 8 Amps, Diameter of Motor shaft: 0.019 m. A variation in speed of AC motor is controlled by AC Speed varic / drive.

2.2 Selection of Belt and Pulleys: A V belt of Fanner make B1360B52 has been selected due to its major advantage of less chances of slippage during operation. As per IS- 2494-1974 for power rating and selection of V–belt, B–cross section of V- belt has been selected. Based on the power rating of motor and maximum rotation speed of motor shaft, Minimum pitch diameter of motor pulley has been selected as per IS – 2494 is 125 mm for B section. The diameter of shaft pulley has been find out considering velocity ratio V. R. = 2 or by selecting available pulley for required shaft. Considering 2% slip of belt at each drive,

$$V.\ R. = D_{mp} / D_{sp}\ (1-S) \tag{5}$$

A standard size of shaft pulley of 100 mm diameter has been chosen for the belt drive.

The centre distance between motor pulley and shaft pulley is selected based on following relations.

$$C = D_{sp} < C < 3\ (D_{mp} + D_{sp}) \tag{6}$$

Taking centre distance between driving and driven pulley is 505 mm. The length of belt calculates using eq. (7)

$$Belt\ length\ (L) = \pi/2(D_{mp}+D_{sp})+2C+(D_{mp}-D_{sp})^2\ /4C \tag{7}$$

$$\text{No. of V-belts required} = \frac{\text{Design of Power rating}}{\text{Modified power rating of V-belt}} \tag{8}$$

Where, Modified power rating of V- belt = Power rating of belt (Kw) x Angle of wrap correction factor (F_d) x Belt length correction factor (F_c).

2.3 Design of Shaft: A shaft has been designed with considering subjected to fluctuating load. A steeped diameter shaft on the right side has been designed for allowing minimum of 30 mm diameter. A shaft has been designed with a maximum of 1000 N load to be sustained. A carbon steel C-45 material has been selected for the shaft design. A maximum length of shaft of 600 mm has been considered. The variable and mean bending moment has been find out considering central shaft weight, weight of pulley at left side and load applied at right side which is as shown in Figure 1. The endurance limit for shaft has been find out using eq. (9) with considering factors for size, surface finish, reliability and fatigue stress concentration[7]. Using Modified Goodman's theory the maximum diameter of shaft has been calculated is 65 mm.

$$\sigma_e = \sigma_e' \times K_{sur} \times K_{sz} \times K_{rel} \times K_f \tag{9}$$

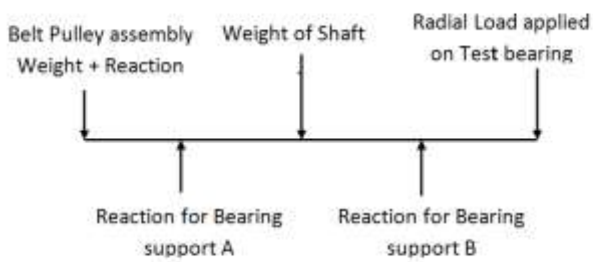


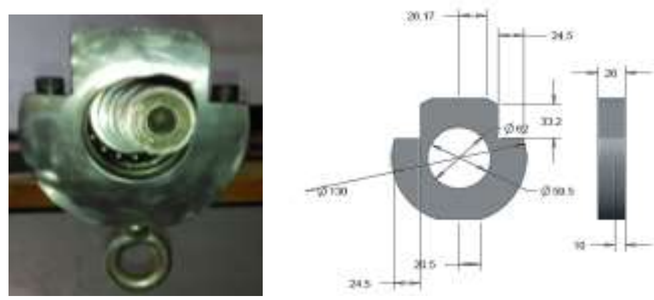
Figure 1. Line diagram of Shaft

2.4 Selection of Support bearing and Housing: Based on the calculations of static and dynamic load carrying capacity a suitable double raw self-aligning ball bearings SKF make 1212 ETN9 alongwith split type pedestal (SKF SE-212) have been selected from manufacturer's catalogue [8] for the present study. The main advantage of this bearing is that it compensate minor misalingement of shaft during rotation.

The Specifications of the support bearing is given as below. Inner diameter = 60 mm, Outer diameter = 110 mm, Width = 22 mm, Dynamic load carrying capacity=31.2 kN, Static load carrying capacity=12.2 kN.

2.5 Selection of Test Bearing and Housing: Based on literature review in present study a polymeric cage SKF BB1B420206 single row deep groove ball bearing has been selected as test bearing. For the purpose of ease of disassembly the elements of bearing and for generating artificial local defect on either of raceway a polymeric cage deep groove ball bearing has been selected.

A suitable split type test bearing housing has been designed and developed for the ease of dismantling and change of bearing during performing experimental work. The photographic view of test bearing housing and its 2D drawing are shown in Fig. 2(a) and Fig. 2(b) respectively.



(a) **(b)**

Figure 2. (a) Photographic view of Test bearing housing **(b)** 2D drawing of Test bearing Housing

An assembly solid model of shaft-bearing system with attached test bearing and its housing created in Creo 2.0 CAD software which has been shown in Figure 3.

Figure 3. Assembly Model of Shaft-Bearing System

3. Experimentations

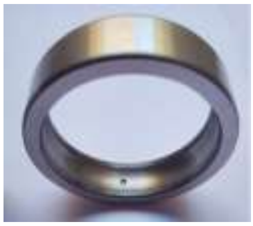
The experiments have been conducted on designed and developed test rig for vibration monitoring of deep groove ball bearing. The vibrations responses have been measured for the healthy (defect free) bearing for varying load of 2 kg, 12 kg, 22 kg and 32 kg with shaft rotational speed of 1000 rpm and 1500 rpm. The experiments have been also conducted for study of vibration response for outer raceway localized defective bearing with fault size of 1 mm, 1.5 mm and 2 mm diameter having uniform depth of 1 mm for shaft speed of 1800 rpm and 2800 rpm. The artificial defect on outer raceway has been generated using electro discharge machining (EDM) process. The shaft rotational speed has been varied using speed variac for electric motor. The experimental study for defective test bearing has been performed at a steady radial load of 100 N which has been applied at the bottom of test bearing housing by means of hook and load attachment. The weight of test bearing housing and load attachment is neglected. The dimensions for test bearing SKF BB1B420206 used in experimentation are shown in Table 1.

The vibrations generated for healthy and defective outer raceway of bearing is captured by permanent magnet type piezoelectric accelerometer CTC make (Model Number AC 102-1A) mounted on top of the housing. The measured vibration signals are sampled at sampling frequency of 1000 Hz with a sampling size of 4096 samples. The captured vibrations data are analyzed in vibration analyzer IMVA 440 in frequency domain.

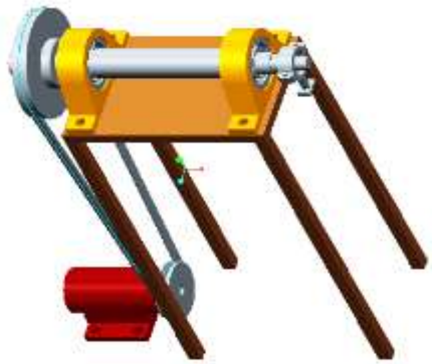
Table 1. Specifications of Test Bearing

Polymeric Cage Deep groove ball bearing	SKF BB1B 420206
Bore of Bearing, mm	30
Inner Race Diameter, mm	39.38
Outer Race Diameter, mm	51.82
Groove radii, mm	5.5
Pitch Diameter (D), mm	45.6
Ball Diameter (d), mm	10.36
Number of Balls (n)	8

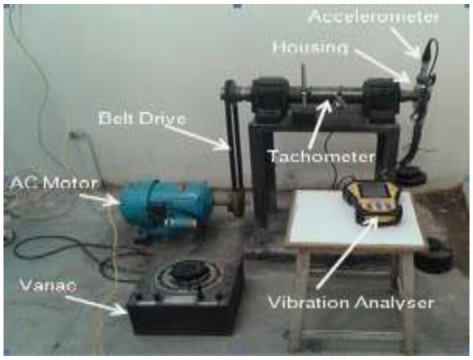
Figure 4. Image of bearing outer race defect



In Figure 4 shows an image of artificial defect generated on outer raceway of circular shape having 2 mm diameter. Figure 5(a) and 5(b) shows 3D assembly model created in Creo 2.0 software and image of experimental set up respectively.



(a)

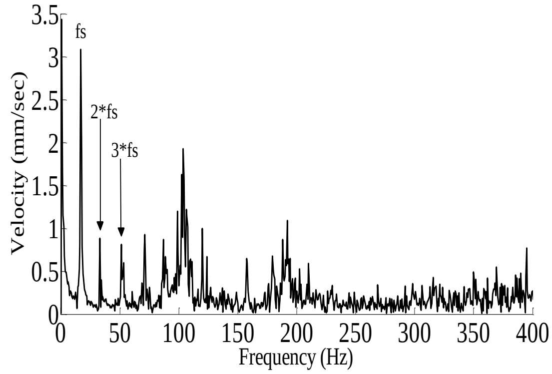


(b)

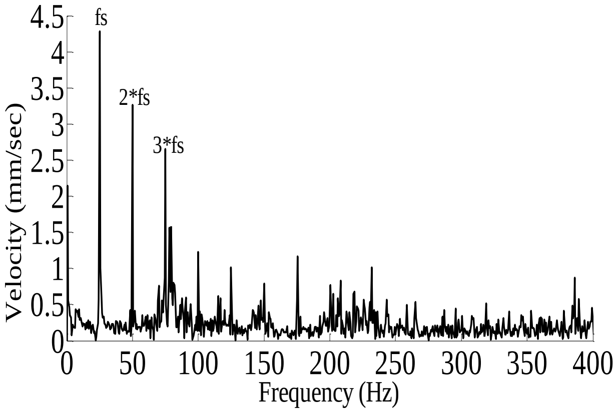
Figure 5. (a) 3-D Assembly Solid Model of Test Rig **(b)** Image of Experimental Set up.

4. Results and Discussions

The experiments have been performed for varying shaft rotational speed and radial load for dry contact healthy (defect-free) and defective bearing. Figure 6(a) and 6(b) shows vibration response of defect free (healthy) bearing at a rotational speed of 1000 rpm (16.66 Hz) and 1500 rpm (25 HZ) for 0 kg radial load. In Fig. 6(a-b) a peak at shaft rotational frequency (fs) 16.88 Hz and 25 Hz with its 2nd and 3rd harmonics have been noticed in the housing velocity spectrum respectively.



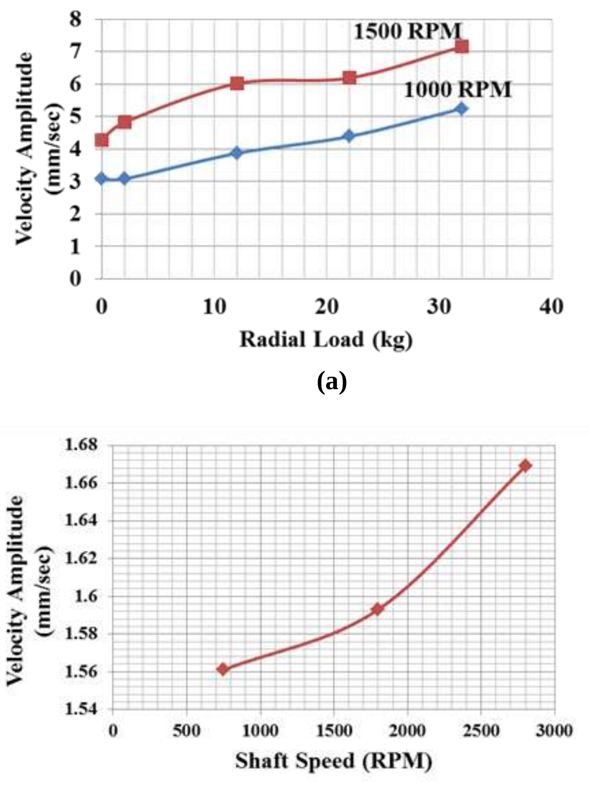
(a)



(b)

Figure 6. Housing Velocity Spectrum for Healthy Bearing with 0 kg radial load **(a)** Shaft Speed 1000 rpm **(b)** Shaft Speed 1500 rpm.

The results comparision for healthy bearing at 1000 rpm and 1500 rpm with varying radial load has been shown in Fig. 7(a). The results shown in Fig. 7(a) and 7(b) indicate that amplitude of vibrations increased at dominent shaft frequencies with increase of radial load and shaft speed respectively.



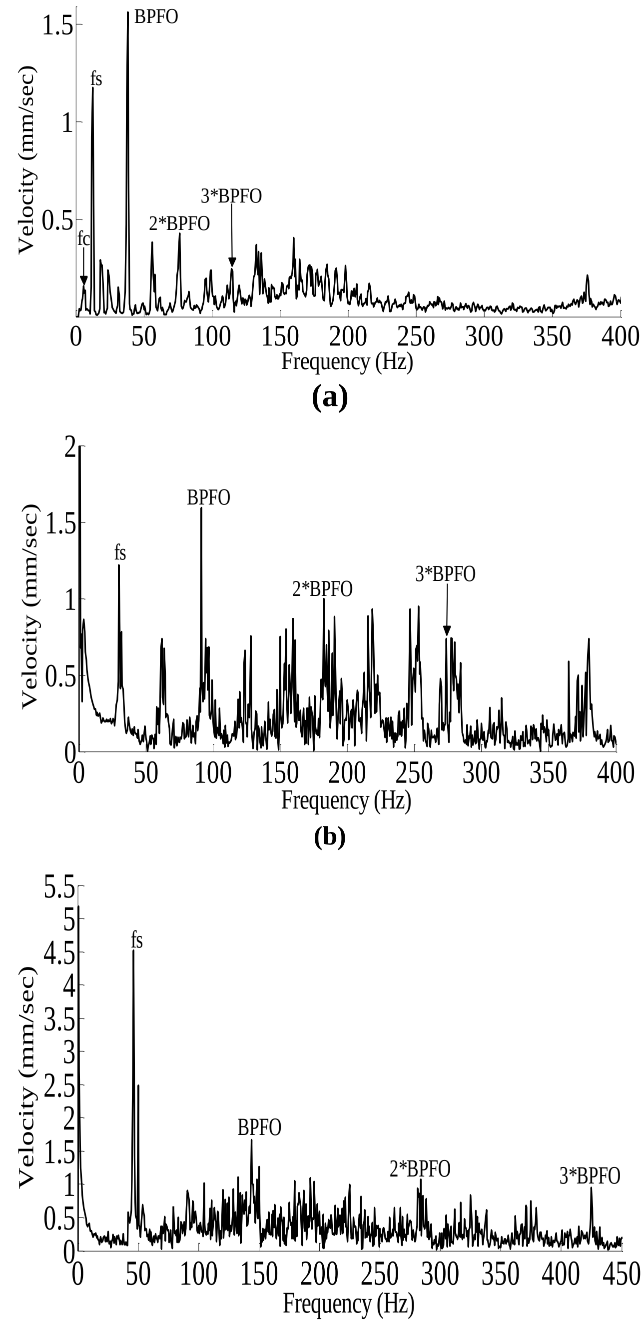
(b)

Figure 7. (a) Velocity amplitude against radial load (For Healthy bearing) **(b)** Velocity amplitude against shaft Speed (For 1 mm Defect Size)

Table 2 shows the comparisons of theoretical shaft frequency calculated using eqs. (1-4) and experimental shaft rotational frequency with varying shaft speed for defect free bearing. It also shows the effect on experimental vibration amplitude for varying radial load on test bearing.

Table 2 Comparisons of vibration amplitude and shaft frequency with varying load and speed for Healthy bearing.

Sr. No.	LOAD (Kg)	Speed (Ns) (RPM)	Shaft rotational frequency (fs) (Hz)		Experimental Velocity Amplitude (mm/sec)
			Theory (Ns/60)	Experiment	
1	0	1000	16.67	16.88	3.079
2		1500	25	25	4.283
3	2	1000	16.67	16.88	3.079
4		1500	25	25	4.822
5	12	1000	16.67	16.88	3.868
6		1500	25	25	6.015
7	22	1000	16.67	16.88	4.388
8		1500	25	25	6.187
9	32	1000	16.68	16.88	5.245
10		1500	25	25	7.149



(c)

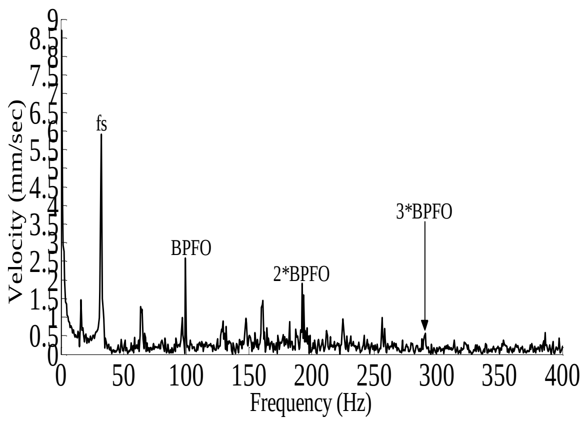
Figure 8. Housing Velocity Spectrum for defective bearing outer race of defect size Ø1 mm **(a)** 750 rpm **(b)** 1800 rpm **(c)** 2800 rpm.

In Figure 8(a-c) shows the experimental vibration response for bearing having 1 mm defect diameter on outer raceway with shaft speed of 750 rpm, 1800 rpm and 2800 rpm respectively. The peaks at shaft rotational frequency (fs) and defect frequency BPFO with 38.13 Hz, 91.25 Hz and 143.8 Hz respectively along with 2nd harmonics 2*BPFO and 3rd harmonics 3*BPFO have been clearly visible in Figure-8(a-c).

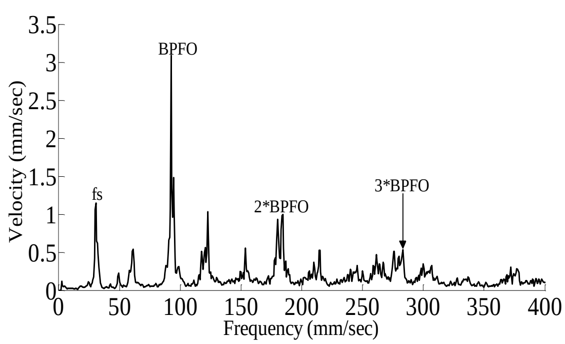
Table 3 Comparisons of Theoretical and Experimental Bearing Defect Frequencies

Outer Race Defect Diameter (mm)	Speed (Ns) (RPM)	Defect frequency BPFO (Hz)		Experiment Velocity Amplitude (BPFO) (mm/sec)
		Theory From eq. (2)	Experiment	
1 Ø	750	38.64	38.13	1.561
1 Ø	1800	92.74	91.25	1.593
1.5 Ø			99.38	2.584
2 Ø			92.5	3.092
1 Ø	2800	144.26	143.8	1.669
1.5 Ø			143.8	2.887
2 Ø			147.5	3.129

Table 3 shows the comparisons of experimental and theoretical (eqs. (1-4)) results for defective bearing under varying speed and radial load. From the obtained results it is clear that amplitude of vibrations at dominant peak as shown in Fig. (10-11) is increases with increase in defect size, load and shaft speed.



(a)



(b)

Figure 9. Housing Velocity Spectrum for Outer race Defect (Shaft speed 1800 rpm) **(a)** Defect Diameter 1.5 mm **(b)** Defect Diameter 2 mm

The vibrations response at shaft speed of 1800 rpm for defective outer raceway having defect diameter 1.5 mm and 2 mm is shown in Figure 9(a) and 9(b) respectively. These results show that amplitude of vibration is increased with increase in defect size on outer raceway.

The experiment has been also conducted for the study of vibration response at shaft speed of 2800 rpm for defective outer raceway having defect size 1mm, 1.5 mm and 2 mm respectively. The results for 2800 rpm speed with varying defect size have been shown in comparison form of vibration velocity amplitude in Fig. 11.

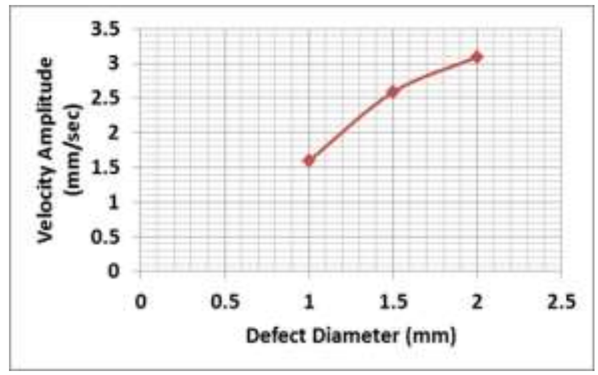


Figure 10. Velocity amplitude against Defect size for 1800 RPM Speed and 10 kg load.

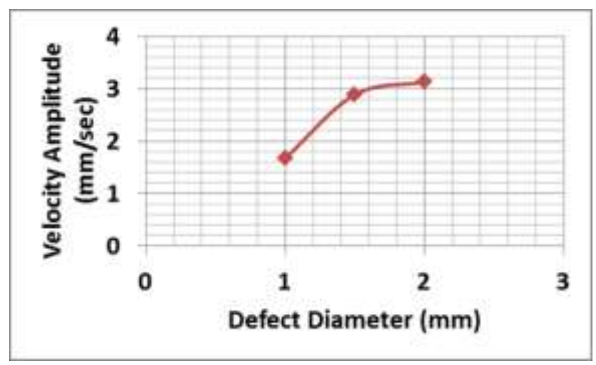


Figure 11. Velocity amplitude against Defect size for 2800 RPM Speed and 10 kg load.

It is also observed in Fig. (10-11) that amplitude of vibration is increased with increase in shaft speed from 1800 rpm to 2800 rpm for same outer race defect diameter 1mm, 1.5 mm and 2 mm respectively. The comparison for theoretical and experimental frequencies shows the good agreement for their values. It has been concluded from the obtained results that only average 0.51% deviation in defect frequency from theoretical value using eq.(2) to experimental frequency. The experimental results obtained from the developed test rig are also in line with similar studies available in published literatures [6, 9-12].

4. Conclusions

The comparisons indicates the good matching of theoretical results and experimental results obtained for normal and defective test bearing by performing experiments on developed test rig. The developed test rig can be used for vibration measurement with varying shaft rotational speed and loading conditions in presence and absence of bearing defects. In experimental results the peaks only at the shaft rotational frequency and its multiple harmonics are clearly visible for defect free bearing. Whereas for outer raceway defect the main peaks at shaft rotational frequency, bearing characteristics defect frequency (BPFO) and its harmonics are visible in the spectra. Good agreements between theoretical and experimental defect frequencies have been observed. The experimental results show that amplitude of vibration is increased with increase in defect size. It is also noticed from the experimental results that there is rise in magnitude of vibrations due to increase in radial load on test bearing and speed of shaft rotation. The test rig design and developed in this article is multifaceted in nature.

Nomenclature

C Centre distance
d Ball diameter
D Pitch diameter
D_{mp} Motor pulley diameter
D_{sp} Shaft pulley diameter
f_s Shaft frequency
k_{sz} Size factor
k_{sur} Surface finish factor
k_{rel} Reliability factor
k_f Fatigue stress concentration factor
L Belt length
n Number of balls in rolling element bearing
S Slip of belt
φ Contact angle
Ø Diameter
$\sigma_{e'}$ Endurance limit stress
σ_e Equivalent Endurance limit stress

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