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Study on Excitation Forces Generated by Defective Races of Rolling Bearing

Dipen S. Shah[a*] V. N. Patel[b]
[a]Research Scholar, C. U. Shah University, Wadhwan - 363030, Gujarat, India.
[a]Sardar Vallabhbhai Patel Institute of Technology (SVIT), VASAD - 388306, Gujarat, India.
[b] Professor, G H Patel College of Engineering and Technology, Vallabh Vidyanagar -388120, Gujarat, India.

Abstract
The vibrations generated by defective rolling elements bearing play vital role in dynamics of rotating machines. In this paper different approach for simulation and numerical analysis of defective bearing races presented by researchers have been compared and extended for multiple defects. This work has been attempted to demonstrate the way of modelling and simulation of local and distributed defect on inner and outer race of deep groove ball bearing. The bearing defect has been modelled as impulse train force or impact force to cause an additional deflection or excitation of rolling elements. The simulated results have been analysed in time and frequency domain. The characteristics defect frequency and its harmonics and amplitude of vibration response of defective bearing is broadly investigated through simulated results.

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Keywords: Rolling element bearing; Excitation force; Pulse shape; Impulse train force; Rolling element load; Local defect; Waviness.

1. Introduction

Rolling element bearings are widely used in the industrial and domestic applications. The proper working of bearing in the machinery is a great importance to avoid any unexpected accident and loss of economy in the industry. The varying rolling element load with respect to angular position of the cage or time varying contact forces results in variation of stiffness and generation of nonlinear vibrations [1-2].

The defects of rolling element bearings are mainly classified into localized defect and distributed defect. The local defects like pits and cracks are mainly occur due to long use of bearing, corrosion or manufacturing errors. While, the distributed defect like surface roughness, race waviness, off size rolling element and misaligned races occur due to manufacturing errors. These defects change the dynamic behaviour of rotor bearing systems. Therefore, researchers have considered the bearing defects in dynamic modelling of shaft bearing systems.

In review paper Shah and Patel [3] have studied various dynamic models of bearings having defect on the races. The bearing defects produce exciting force due to interaction of defect with rolling element [4-5]. McFadden and Smith [4] have suggested that each time a single inner race local defect strikes its mating element, a pulse of short duration is generated that excites the resonances periodically at the characteristic frequency related to the defect location. The rate of the repetition with which this impact occurs is known as the inner race element passing frequency, denoted here by BPFI.

*Corresponding author. Tel.: +91 2692-249264; Fax: +91 2692-274766.
E-mail address: dipenhah.mech@svitvasad.ac.in, dips.engineer@gmail.com (Dipen S. Shah), vinodpatel@gcet.ac.in (Dr. Vinod N. Patel).

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The authors have modeled the single point defect as an impulse described by an impulse function. Tandon and Choudhary [5] have extended the concept of McFadden and Smith [4] model to predict the vibration response of localized defective ball bearing. They have correlated finite duration pulses with width of bearing element's defects under radial and axial load. The authors have considered the regular shape like rectangular, triangle and half sine pulse during simulation. While, Patel et al. [6] have reported the forces generated due to additional deflection of balls when balls pass through defective bearing races. In recent paper Khanam et al. [7] have found the excitation forces mainly at three events, when the rolling element enters in inner race defect, impact with defect and exit edge of defect. In another study of Khanam et al. [8] have derived the impact force based dynamic model to predict the dynamic behavior of outer raceway single point defect based on the concept of excitation due to ball mass striking the raceway.

The researchers [9-12] have also studied the effect of races waviness and off size rolling element on vibration response for shaft bearing system. Babu et al. [11] have incorporated the combined effect of frictional moment and waviness of the races and balls as sinusoidal functions for various waviness orders and amplitudes. Harsha et al. [12] have concluded that severe vibrations occur for outer race waviness when balls and number of waves are equal.

The literature review revels that the presence of defects on the bearing races play vital role on dynamic behaviour of the machines. Therefore, in present study the numerical simulation for excitation forces generated by defective bearing has been carried out. Moreover, these excitation forces have been analysed in time and frequency domain. The concept of impulse train representing exciting forces generated due to inner race defect discussed in reference [4] has been extended for outer race defect and ball pass frequency of outer race (BPFO) has been validated numerically. The pulse shapes related to defect shape modelled by Tandon and Choudhary [5] have also been simulated. The impact force excitation [8] and excitation forces generated by bearing race waviness have been simulated and bearing defect frequencies have been validated numerically. Moreover, excitation forces generated due to multiple defects on either race have also been studied.

2. Load distribution and effect of local defect.

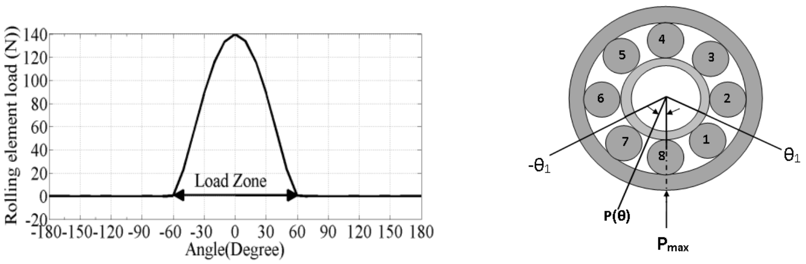
Based on the geometry of the bearing, the normal load on the ball/raceway at ball position 'θ' is calculated by the following equation [13].

$$P(\theta) = P_{max}\left[1-\left(\frac{1}{2\varepsilon}\right)(1-\cos\theta)\right]^n \text{ for } -\theta_1 < \theta < \theta_1$$
$$= 0 \quad \text{elsewhere.} \qquad (1)$$

A deep groove ball bearing SKF BB1B420206 is the study bearing. The detailed specifications and theoretical defect frequencies are tabulated in table 1. The load distribution and rolling elements position for a study bearing is shown in Fig. 1. The load zone for this bearing is 120^0 ($\theta_1 = \pm 60°$) and maximum radial load on rolling element found by eq. (1) is 139.8 N, that can be observed from Fig. 1(b). The short duration impulses are generated due to the interaction of the defect and rolling elements. These impulses may excite the resonance frequency of other machine components. The radial load 'P' at point of excitation, will affect the resultant excitation force, 'F (t)'.

Table 1 Bearing Specifications

Deep groove ball bearing	SKF BB1B420206	Number of balls (Nb)	8
Outer race diameter (do), mm	51.82	Inner race diameter (di), mm	39.38
Ball diameter (d), mm	10.36	Radial Load (Q), N	200
Local defect Width, mm	0.5	Speed of shaft (N_s), rpm	1500 (f_s =25 Hz)
Raceway waviness Amplitude, μm	0.01	Cage Frequency (*fc*)	$\frac{\omega_s}{2}\left(1-\frac{d}{D}\cos\alpha\right)$ = 9.76Hz
Ball Pass Frequency for Inner race (BPFI)	$\frac{N_b.\omega_s}{2}\left(1+\frac{d}{D}\cos\alpha\right)$ = 77.52 Hz	Ball Pass Frequency for Outer race (BPFO)	$\frac{N_b.\omega_s}{2}\left(1-\frac{d}{D}\cos\alpha\right)$ = 122.72 Hz



(a) Angular position of ball (b) Load distribution

Fig. 1 Rolling Element Load Distribution Curve

It is assumed that this excitation force depends linearly on impulsive force ‘f (t)’ and radial load ‘P (θ)’. For a constant rotational shaft speed the excitation force in the radial direction is as follows [14]:

$$F(t) = f(t)P(t)\cos\theta \tag{2}$$

3. Simulation for local defect.

In present study the disturbing forces or excitation forces generated by local or distributed defects on either race of the study bearing have been simulated using following considerations:

1. Local defect can be modelled as an infinite series of impulses of equal amplitude [4],
2. Excitation force as a pulse of rectangular shape, triangular shape or half sine wave [5],
3. Additional deflection of ball, when it passes through local defects,
4. Impact based force analysis of bearing having local defects [8].

3.1 Impulse train generated due to localized defect.

The excitation force produced due to interaction of defect with rolling element can be modeled as a series of impulse train. The impulse train generated due to repetitive interaction of the rolling elements with defect is expressed as [4]:

$$f(t) = d_0 \sum_{k=-\infty}^{\infty} \delta(t - kT_d) \tag{3}$$

Where, d_0 = Impulse amplitude, $\delta(t)$ = Dirac delta function, T_d = Time period between impulses (1/BPFI or 1/BPFO).

The resultant excitation forces ‘F(t)’ generated for study bearing having local defect on its inner race or outer race are computed using eqs. (1-3) and plotted in Fig. 2. It can be observed from Fig.2 (a, c), when the rolling element is in unloaded zone the amplitude of impulse has become zero. The repetition rate of impulses is 0.0081 sec for inner race defect (refer Fig. 2(a)) and 0.0129 sec for outer race defect (refer Fig. 2(c)). The peaks at cage frequency (f_c), bearing defect frequency BPFI (Fig. 2(b)), BPFO (Fig. 2(d)) and their harmonics along with sidebands BPFI ± f_c or BPFO ± f_c are observed clearly.

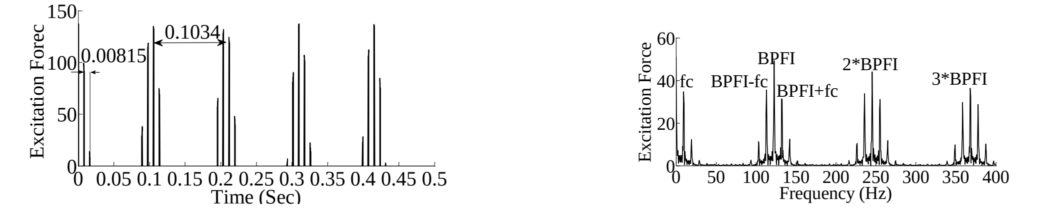
3.2 Pulse shape of excitation force.

Generally the generated pulses ‘f (t)’ is periodic in nature. For simplicity, these have been assumed as even functions and can be expanded in Fourier series as expressed in eq. (4).

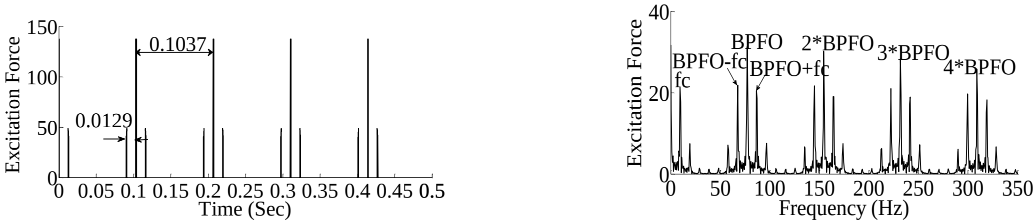
$$f(t) = F_0 + \sum_{s} F_s \cos(s\omega t) \tag{4}$$

Where, ‘ω’ is pulses generation frequency, which depends on the location of the defect. The Fourier coefficients, ‘F_0’ and ‘F_s’, for a rectangular pulse, triangle pulse and half sine pulse form and for rolling element load ‘P(t)’ computed by Tandon and Choudhary [5] have been used for this simulation. The excitation force of rectangular pulse, triangular pulse and half sine wave computed using Fourier coefficients for defective races have been demonstrated in Fig. 3, Fig. 4 and Fig.5, respectively.

The impulses are separated clearly with time period of 1/BPFO in Fig. 3(c), Fig. 4(c) and Fig. 5(c), while impulses are not separated with time period of 1/BPFI in Fig. 3(a), Fig. 4(a) and Fig. 5(a). This may be due to rotation of the inner race defect at the shaft speed. The corresponding defect frequency (BPFI or BPFO) are visible

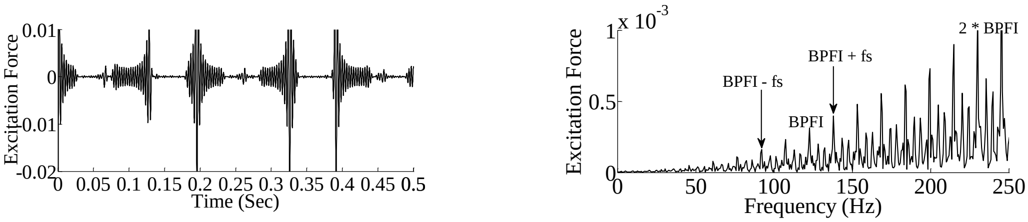


(a) Time response for inner race defect (b) Frequency response for inner race defect

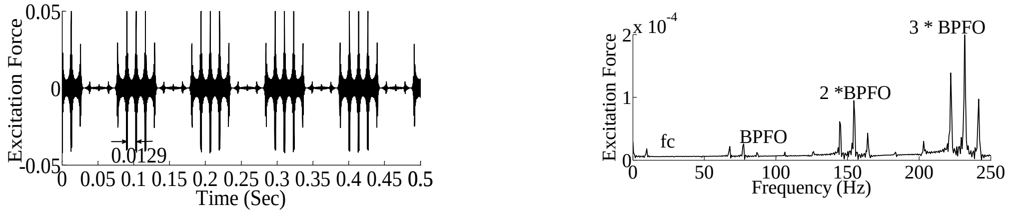


(c) Time response for outer race defect (d) Frequency response for outer race defect

Fig. 2 Impulse Train generated due to defective races.



(a) Time response for inner race defect (b) Frequency response for inner race defect



(c) Time response for outer race defect (d) Frequency response for outer race defect

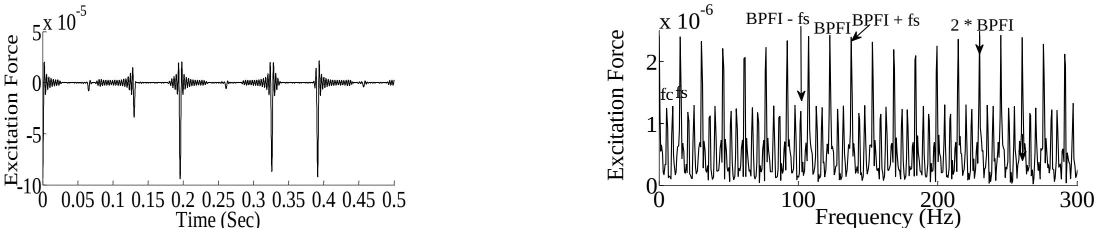
Fig. 3 Excitation force of rectangular pulse shape for defective races

in their respective frequency response. The amplitude of the defect frequency harmonics (2*BPFO, 3*BPFO) has increased than the amplitude of fundamental defect frequency (BPFO) in case of rectangular defect (refer Fig. 3(b) and Fig. 3(d)). While, the fundamental defect frequencies have become more clear in case of triangular pulse and half sine wave, which can be observed in Fig. 4(b), Fig. 4(d) for triangular pulse, Fig. 5(b) and Fig. 5(d) for half sine wave. The peaks at sidebands (BPFI ± f_s) are noticed due to the rotation of inner race defect at shaft rotation speed, while the peaks at side bands (BPFO ± f_c) are observed due to cage rotation.

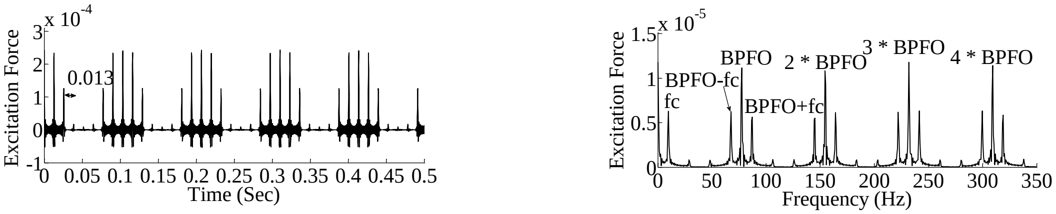
3.3 Additional deflection due to defect on races.

The load–deformation relationship for Hertzian contact is expressed as follows [13]:

$$F = K\delta^{1.5} \tag{5}$$

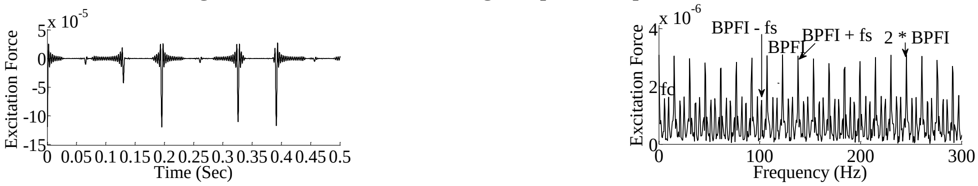


(a) Time response for inner race defect (b) Frequency response for inner race defect

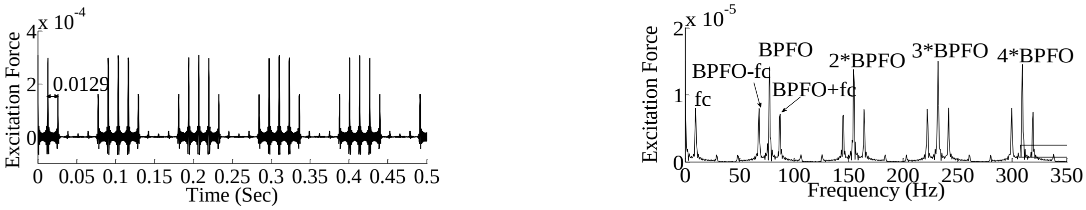


(c) Time response for outer race defect (d) Frequency response for outer race defect

Fig. 4 Excitation force of triangular pulse shape for defective races



(a) Time response for inner race defect (b) Frequency response for inner race defect



(c) Time response for outer race defect (d) Frequency response for outer race defect

Fig. 5 Excitation force of half sine wave for defective races

Where, ‘K’ is nonlinear stiffness at contact point and ‘δ’ is radial deflection at any rolling element position (θ).The deflection of the rolling element varies when it reaches the defective zone of the races. This additional deflection depends on the size, shape and location of the defect [6].

Additional deflection $\square = \left(\frac{d}{2} - 0.5d\cos(\Phi_{ball})\right)$ Where, Φ_{ball} = Width of defect/ radius of the ball. (6)

The excitation forces generated due to the additional deflection of rolling elements have been computed using the eqs. (5, 6). The time response and frequency response for defective races have been plotted in Fig. 6. The time period between interaction of the defect and successive balls can be visible clearly from the time response (refer Fig. 6 (a) and (c)). Moreover, the peaks at corresponding defect frequencies and their harmonics are observed in Fig. 6(b) and Fig. 6(d).

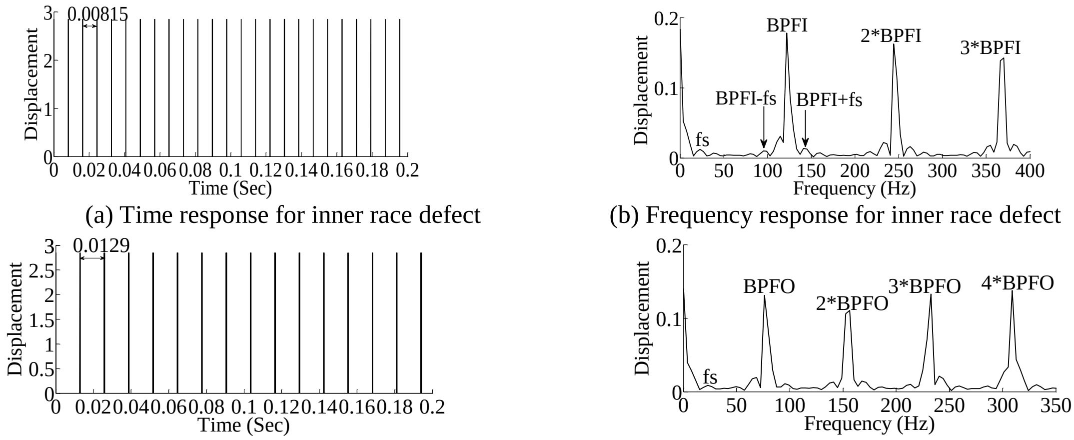
3.4 Excitation force generated due to impact.

Khanam et al. [8] have proposed 2 DOF impact force based excitation model for outer raceway local defect in bearing. In their study the excitation force due to single defect is assumed to be periodic force pulse train with periodic time ‘T’. This periodic force pulse train is represented in Fourier series as follows [8].

$$f(t) = a_0 + \sum_n (a_n \cos n\gamma t + b_n \sin n\gamma t) \tag{7}$$

Where 'a_0', 'a_n' and 'b_n' are Fourier coefficients as given in reference [8], γ = fundamental frequency, n=harmonics.

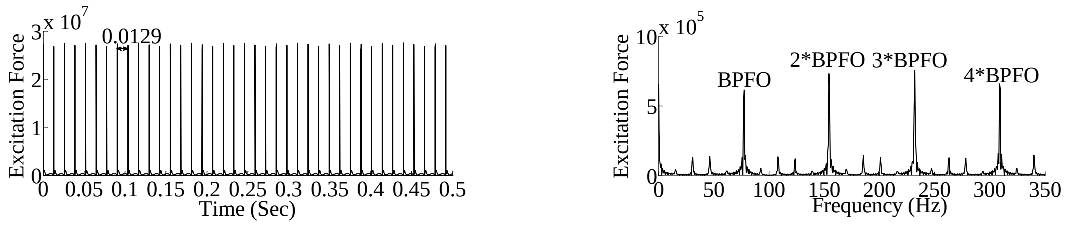
The simulated results for single local defect of the outer raceway of bearing are shown in Fig.7. The peaks are observed at outer race defect frequency BPFO.



(a) Time response for inner race defect (b) Frequency response for inner race defect

(c) Time response for outer race defect (d) Frequency response for outer race defect

Fig. 6 Excitation force generated due to additional deflection



(a) Time response (b) Frequency response

Fig. 7 Impact force excitation for single defect on outer race

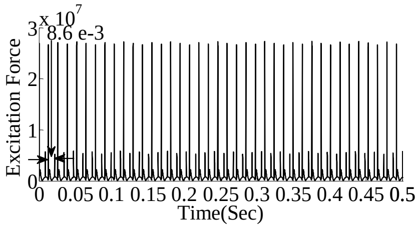
This impact based force excitation model has been extended for multiple defects on outer race. The computation for contact forces have been repeated as per the number of defects on outer race. In this simulation two impulse trains generated for two defects during one cage rotation as a result of inter action of each defect with rolling elements. The time delay between two impulse trains depends on angular difference between two defects. This time delay can be computed by following expression [15]:

$$\tau = \frac{\Psi}{(\Phi * BPFO)} \tag{8}$$

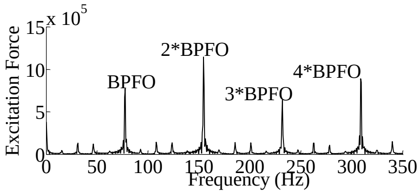
Where, 'τ' is time delay, 'ψ' is angular difference between two defects, 'φ' is angle between two balls. To study the effect of multiple defects on excitation forces, two defects on outer race of the bearing at an angular difference of 30° has been considered. The time delay computed through eq. (8) for angular difference of 30° is 8.6×10^{-3} sec, this can be observed from Fig. 8(a). The frequencies content for this excitation force in Fig. 8(b) is same as the frequency content of the single defect excitation force in Fig. 7(b). Although the amplitude of the excitation force has increased due to multiple events.

Khanam et al. [7] have modeled the spall like defect on inner raceway based on the concept of multi-event excitation force F(t). This excitation force depends on impact force (f), and load (L).The net excitation force vector is as follows [7]:

$$F(t) = f(\omega_{id}) L(\omega_{id}) \cos(\omega_s t) \tag{9}$$



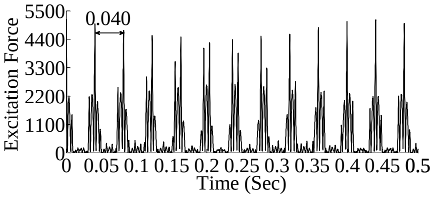
(a) Time response



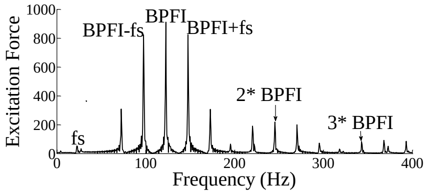
(b) Frequency response

Fig. 8 Impact force excitation for two defects on outer race at angular difference of 30°

Figure 9(a) and Fig. 9(b) show the time and frequency response for excitation force generated due to inner race defect. In Fig. 9(b) the peaks at shaft rotation frequency, BPFI along with side bands at BPFI ± f_s are clearly visible.



(a) Time response

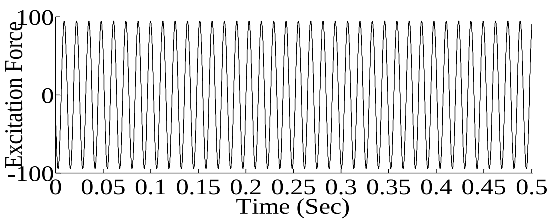


(b) Frequency response

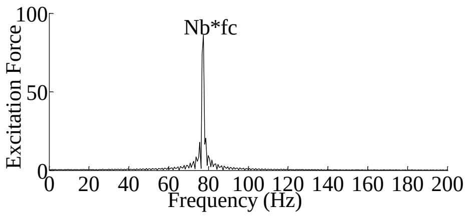
Fig. 9 Impact force excitation for single defect inner race

4. Excitation force generated due to race waviness.

The waviness on the races has been modelled as a periodic function. The expression for excitation force derived by Tandon and Choudhary [10] has been used for the simulation of inner race and outer race waviness of the bearing. The simulated results for outer race waviness are plotted in Fig. 10. The frequency of excitation force depends on the waviness order. The waviness order considered in present study is 7 (= N_b-1, i =1). The peak at defect frequency (i*N_b*fc) is dominant in Fig.10 (b), which was also noticed by Babu et al. [11] during their experimental study for the same bearing. The same methodology has been adopted for inner race waviness and simulated results are presented in Fig. 11. The peaks at shaft rotation frequency, f_s and at i* N_b*(f_s-f_c) ±f_s are found.

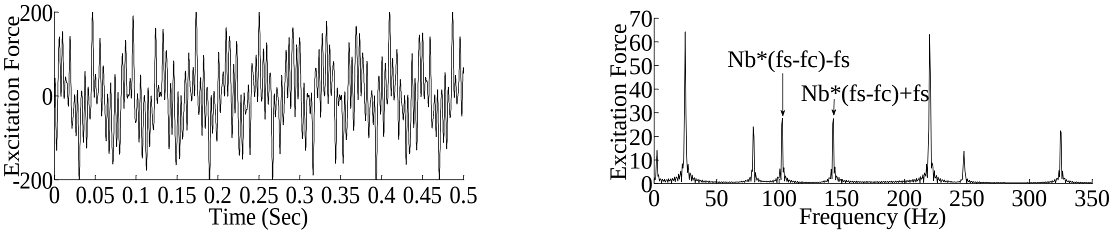


(a) Time response



(b) Frequency response

Fig. 10 Excitation force caused due to outer race waviness



(a) Time response (b) Frequency response

Fig. 11 Excitation force caused due to inner race waviness

5. Conclusion

In the present investigation excitation forces caused due to bearing race defect have been simulated and the characteristic defect frequencies have been confirmed with their theoretical values. It has been observed that the triangular pulse train and half sine wave are more appropriate for local defect simulation than the rectangular pulse train due to their convincing frequency response. The time period between two successive impulses generated due to interaction of multiple defects on outer race can be precisely predicted using extended impact force model. In case of race waviness the dominant frequency peaks are governed by waviness order. Authors of this paper believe that the additional deflection of rolling element or impact based excitation is more reliable for simulation of local bearing defect.

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