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EFFECT OF VARIATION IN CLEARANCE ON THE VIBRATION RESPONSE OF DEFECTIVE ROLLING ELEMENT BEARINGS

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Abstract. *A number of researchers have proposed theoretical models to explain the vibration response obtained from defective rolling element bearings. The effect of variation in bearing clearance has not been considered in these models. In the present work, the effect of variation of radial/diametral clearance on the vibration spectra obtained from defective bearings has been investigated. The model proposed by Choudhury and Tandon [1] has been extended in this work. Vibratory forces due to an inner race defect and the response of the rotor bearing system as depicted in the model [1] has been further investigated for variation in bearing clearance. Numerical results have been obtained for NJ 204 bearing with different values of clearances ranging from C2 through normal to C4 values. Different values of loads and speeds have also been considered for obtaining the results. The results show significant variation in the spectra obtained for different values of diametral clearances. These results are expected to satisfy the difference in theoretical and experimental spectra as obtained by earlier researchers.*

1 INTRODUCTION

Rolling element bearings are frequently encountered in rotating machinery due to their load carrying capacity and low-friction characteristics. They work under heavy loadings generated in the machinery and are subjected to time and space varying dynamic loads. These bearings are subjected to fatigue loading and are prone to failure. It is important to detect defects at its incipient stage in order to prevent long-term break downs or in some cases possible catastrophic failures. Bearing defects may be categorized into two types, viz., local and distributed defects. The local defects include cracks, pits and spalls which are primarily caused due to fatigue failure. The distributed defects, on the other hand, include surface waviness and unequal rolling element diameter etc. These are caused either due to abrasion during operation or due to manufacturing inaccuracy. Vibration analysis is carried out with an objective of condition monitoring and quality inspection in case of distributed defects.

For condition monitoring of bearings with localized defect, vibration measurement has been done in time domain, frequency domain and time-frequency domain. In time domain, some statistical parameters of vibration signal such as probability density, kurtosis etc. have been studied [2]. Frequency domain approach, which essentially means spectral analysis of vibration signal, has been widely applied for bearing defect detection. In recent years, some time-frequency domain analysis such as wavelet transform has been applied by researchers for analysis of vibration signal from defective bearings when such signals are masked by high noise or are of non-stationary nature [3]. However, frequency domain approach remains the most popular technique till date for bearing defect detection because the location of defect can be uniquely correlated to appearance of characteristic defect frequencies in the spectrum. These frequencies can be calculated from shaft speed and bearing geometry for location of defect on inner race, outer race or on one of the rolling element. The expressions for characteristic defect frequencies are well established and are listed in Table 1 (ω_s: shaft rotation frequency; d: rolling element diameter; D: pitch diameter; Z: number of rolling elements; α: contact angle).

Characteristic frequency	Expression
Cage frequency, ω_c	$\left(\frac{\omega_s}{2}\right)\left[1-\left(\frac{d}{D}\right)\cos\alpha\right]$
Outer race defect frequency, ω_{od}	$\left(\frac{Z\omega_s}{2}\right)\left[1-\left(\frac{d}{D}\right)\cos\alpha\right]$
Inner race defect frequency, ω_{id}	$\left(\frac{Z\omega_s}{2}\right)\left[1+\left(\frac{d}{D}\right)\cos\alpha\right]$
Rolling element defect frequency, ω_{red}	$\left(\frac{D\omega_s}{d}\right)\left[1-\left(\frac{d}{D}\right)^2\cos^2\alpha\right]$

Table 1: Characteristic defect frequencies for a rolling element bearing

Researchers have developed many models [1, 4-7] to explain vibration spectra obtained from defective bearings. In some of these models [1, 6, 7], attempt has been made to obtain amplitude information of spectral components. In all these models, excitation has been considered to be due to the impulse or short-duration pulses generated when a defect on a bearing element interacts with one of the mating elements. These pulses excite natural frequencies of bearing parts and housing structures and result in increase in vibration energy. Periodicity of these pulses is related to characteristic defect frequency as stated above.

In case of radially loaded bearings, the vibratory forces produced due to these pulses will also depend on the load at the location of defect. This in turn will be affected by bearing clearance. However, the effect of variation in bearing clearance has not been investigated in the above models. In this paper, the effect of variation in bearing clearance on the vibratory forces for an inner race defect has been studied in details.

2 FORMULATION OF VIBRATORY FORCES

The excitation in case of a bearing with a local defect is caused by pulses generated due to the interaction of defect with the mating elements. Generation of excitation force in case of inner race defect in a radially loaded bearing is depicted in Figure 1. The interaction of defect on inner race with mating rolling element generates pulses. The load at the point of pulse generation as well as its relative position with respect to the direction of measurement will also influence the excitation force. Therefore the excitation force in the direction of measurement can be expressed as

$$F(t) = f(t)P(t)\cos\psi \tag{1}$$

where $f(t)$ represents generated pulse form, $P(t)$ is the load at the point of excitation and ψ represents the angular position of the pulse generation with respect to direction of measurement. Since the measurement is done along the direction of maximum load, ψ therefore represents the angular position of pulse generation with respect to the direction of radial load.

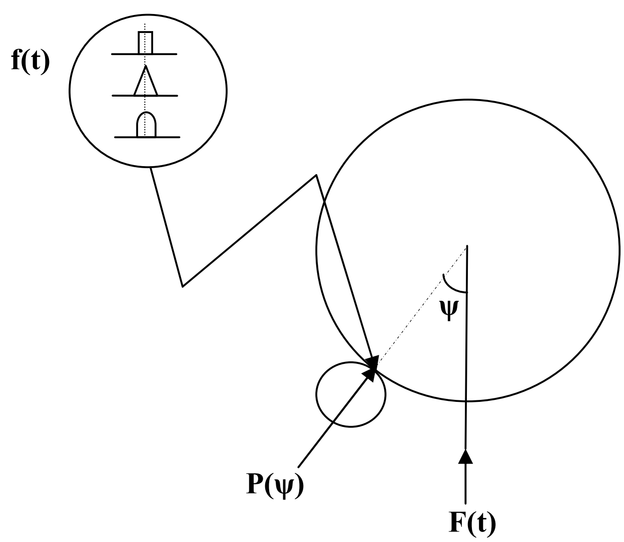


Figure 1: Generation of pulses due to interaction of defect

2.1 Pulse form

The generated pulse form *f(t)* depends on severity, extent and age of damage. A typical rectangular pulse form is shown in Figure 2. The pulse width, ΔT, can be determined by dividing the defect width by the relative velocity between the mating elements. Since these pulses occur at regular time intervals, they are periodic in nature. In case of inner race defect the frequency of occurrence is same as inner race defect frequency, ω_{id}. Assuming these pulses to be even functions, they can be expressed in Fourier series as

$$f(t) = f_0 + \sum_s f_s \cos s\omega t \tag{2}$$

The Fourier coefficients, f_0 and f_s, for a simple rectangular pulse form are given in Table 2 [6]. The actual pulse forms are, however, quite complex.

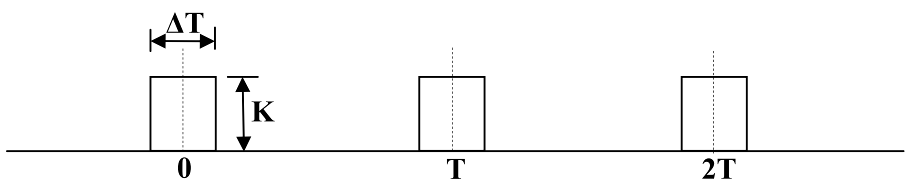


Figure 2: A rectangular pulse form

f_0	f_s
Km	$\left(\frac{2K}{\pi s}\right)\sin \pi s m$

Table 2: Fourier Coefficients for rectangular pulse form: K: pulse height; m: ΔT/T; ΔT: pulse width; T: time period

2.2 Load factor

The load distribution in a radially loaded bearing is as shown in Figure 3. The load on a rolling element at any angle, ψ, can be expressed as [8]

$$P(\psi) = \begin{cases} P_{\max}\left[1 - (1/2\varepsilon)(1-\cos\psi)\right]^n, & -\psi_0 < \psi < \psi_0 \\ 0 , & elsewhere \end{cases} \tag{3}$$

where P_{max} is the maximum rolling element load which acts in the direction of radial load, ε is the load distribution factor, $\pm\psi_0$ is the extent of load zone and $n = 1.5$ for ball bearing and 1.11 for roller bearing. P_{max} and ε are given by the following expressions which show their dependence on bearing clearance.

$$P_{\max} = K_d\left(\delta_{\max} - 0.5C_d\right)^n \tag{4}$$

$$\varepsilon = 0.5\left(1 - \frac{C_d}{2\delta_{\max}}\right) \tag{5}$$

where K_d is the deformation constant, δ_{max} is the maximum deflection and C_d is is the diametral clearance. For a steel roller bearing with L as the effective roller length, K_d is given by the following expressions

$$K_d = 3.46 \times 10^4 L^{8/9} \left(N / mm^{1.11} \right) \tag{6}$$

For a ball bearing,

$$K_d = \frac{34{,}300}{\kappa^{0.35}} d^{0.5} \left(N / mm^{1.5} \right) \tag{7}$$

where $\kappa = \dfrac{r_o + r_i - d}{d}$, r_i and r_o being the groove radii for inner and outer races of ball bearing.

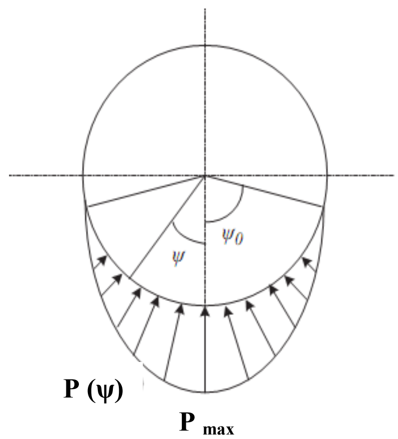


Figure 3: Load distribution in a bearing under radial load

For a moving defect, as in the case of an inner race defect, the angular position of the point of excitation changes with time. Therefore the load at that point changes periodically at a frequency that depends on the location of the defect. Since the load is evenly distributed about the point of maximum deflection, it can be expanded in Fourier series for even functions as

$$P(\omega t) = P_0 + \sum_r P_r \cos r\omega t \tag{8}$$

where ω is the frequency which depends on the defect location. The Fourier coefficients, P_0 and P_r, are given in ref. [6].

2.3 Excitation force due to inner race defect

Excitation force is caused due to interaction of a defect on the inner race with mating rolling element. Under this condition, the pulses are generated at the inner race defect frequency, ω_{id}, and the defect itself moves at the shaft speed, ω_s. Therefore. Eq. (1) can be rewritten as

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

The expressions for pulse form, f, having a periodicity of ω_{id} and the load, P, having a periodicity of ω_s can be obtained from Eqs. (2) and (8) respectively. Since all the factors on the right hand side of the Eq. (9) are periodic, the resultant *F(t)* is a sum of harmonic components. The amplitudes of these components for various frequencies are given in the following equations

$$F(\omega_s) = f_0\left(P_0 + P_2/2\right) \tag{10a}$$

$$F(r\omega_s) = \left(f_0/2\right)\left(P_{r-1} + P_{r+1}\right) \tag{10b}$$

$$F(s\omega_{id}) = f_s\, P_1/2 \tag{10c}$$

$$F(s\omega_{id} \pm \omega_s) = f_s\left(P_0/2 + P_2/4\right) \tag{10d}$$

$$F(s\omega_{id} \pm r\omega_s) = \left(f_s/4\right)\left(P_{r-1} + P_{r+1}\right) \tag{10e}$$

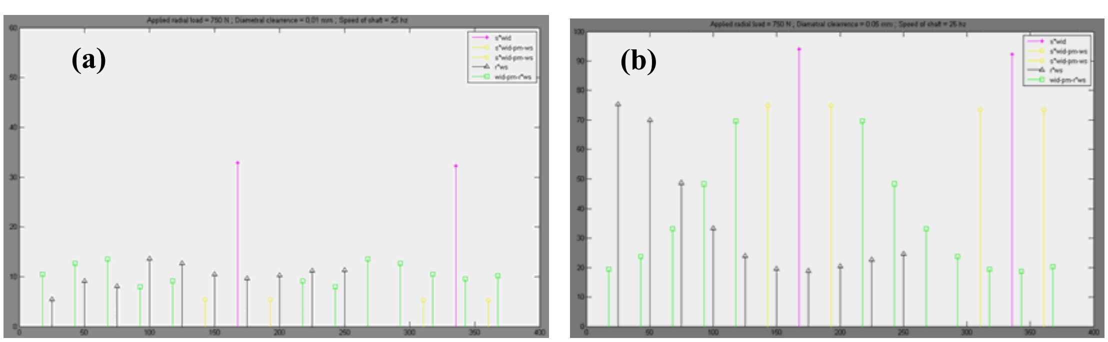
3 NUMERICAL RESULTS

The expressions for vibratory forces, as obtained in previous section, have been applied to an NJ 204 cylindrical roller bearing with different values of clearances. NJ 204 bearings have 20 mm bore, 47 mm outside diameter, 14 mm width, and 34 mm pitch diameter. They have 11 rollers and the nominal contact angle is zero degree. Each roller of an NJ 204 bearing has a length of 9 mm and a diameter of 7.5 mm. The ranges of diametral clearances for such a bearing are given in Table 3 [9].

Bore	C2		Normal		C3		C4	
(mm)	Min	Max	Min	Max	Min	Max	Min	Max
20	0	50	40	90	70	120	100	150

Table 3: Ranges of diametral clearances in microns for NJ 204 bearing.

Vibratory forces for different values of clearances under a load of 750 N and at a speed of 1500 rpm have been calculated using Eqs. (10a) – (10e) and are plotted in Figures 4(a) – 4(d). The values of shaft frequency and inner race defect frequency for NJ 204 bearing at 1500 rpm are 25 Hz and 167.83 Hz respectively. It is observed from Figures 4(a) – 4(d) that the amplitudes of the components at inner race defect frequency, ω_{id}, increases with increasing values of clearances. This could be because of generation of pulses with greater pulse area [1] for higher values of clearances.



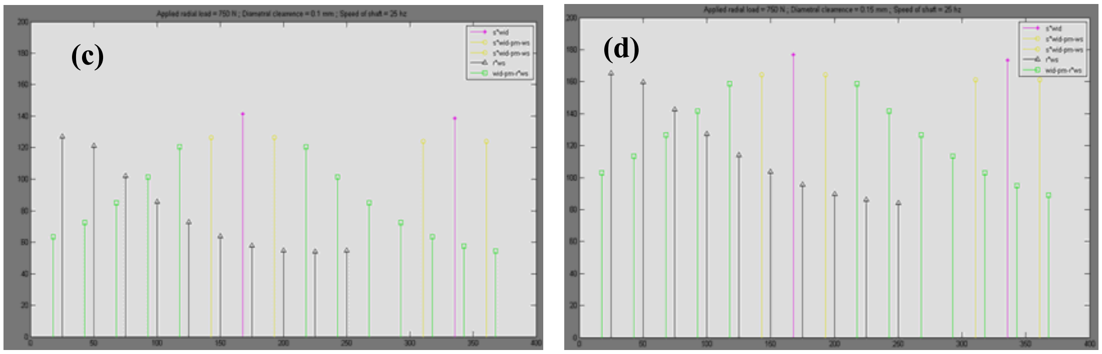


Figure 4: Spectra of vibratory forces under 750 N and 1500 rpm for clearance values of (a) 10 µm; (b) 50 µm; (c) 100 µm; and (d) 150 µm.

Vibratory forces for 50 microns of clearance and two different conditions of load and speed, viz., (i) 750 N and 3000 rpm; and (ii) 1500 N and 1500 rpm, have been shown in Figure 5(a) and (b). A comparison with the corresponding plots of Figure 4 shows that the pattern of the spectra is almost same for different loads and speeds.

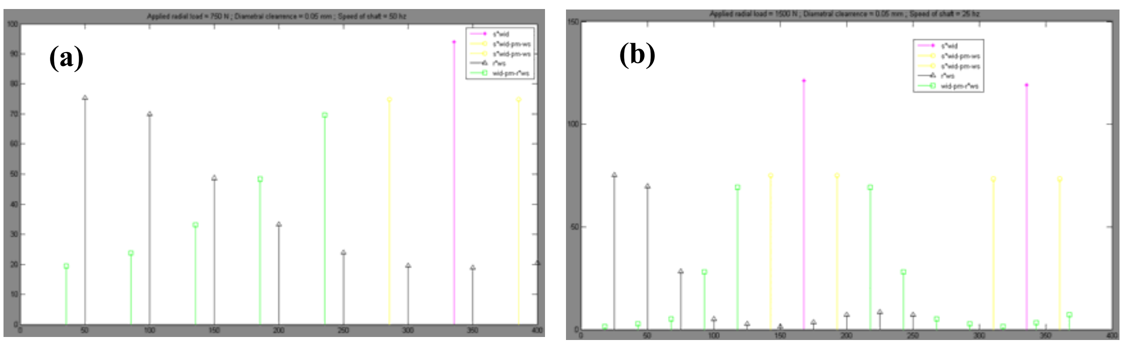


Figure 5: Spectra of vibratory forces under (a) 750 N and 3000 rpm; and (b) 1500 N and 1500 rpm for clearance of 50 µm.

4 CONCLUSIONS

- Spectra of vibratory forces have been obtained for different values of clearances in a bearing with inner race defect.
- Amplitudes of the components at inner race defect frequency increase with increasing values of clearances.
- Distribution of sidebands around the inner race defect frequency significantly changes for different values of clearances.
- Distribution of components in the spectra of vibratory forces does not change significantly with change in load and speed.

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