Influence of a local defect on the cage vibrations for a ball bearing

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Abstract. Impacts will occur when the balls pass through the local defect. The cage vibrations can be affected by the impacts, which may influence the steady state of ball bearing system. Thus, the relationship between defect size and the cage vibrations can be helpful for ascertaining the mechanisms of the cage vibrations. In this paper, a multi-body dynamic model of a ball bearing with a local defect is proposed by using the multibody commercial software. Effects of the defect sizes and inner ring speed on the angular velocity of the cage are studied. The relationship of the contact forces between the balls and bearing components are discussed too. The results show that the multi-body dynamic model can be applied to study the cage vibrations for a defective ball bearing.

1. Introduction

As one important transmission component, ball bearings are commonly used in kinds of rotating machines. Due to its special structure and adverse operation circumstances, various failures including cracks, pits, and spalls may produce on the raceways [1], which can cause unacceptable vibrations of the cage and furtherly damage the steady state of bearing system. Hence, it is very helpful to study the vibrations of cage caused by local defect for avoiding catastrophic economic losses and accidents.

Many scholars have focused on the issues of cage flexibility modelling, as well as the influence of cage unbalance on the cage vibrations and ball-to-cage stress distribution. Pederson et al. [2] developed a ball bearing dynamic model considering the cage flexibility. The cage is modelled as a mass-spring-damper system. This work investigated the effects of ball-to-cage pocket clearance and cage stiffness on the cage motion and ball-to-cage pocket contact forces. Nick et al. [3] investigated the effects of cage flexibility on the bearing dynamics. Ashtekar et al. [4] developed a new dynamic model by an explicit finite element modelling method and discrete element modelling method. They studied the effects of shaft misalignment on the stress distribution of cage. Ye et al. [5, 6] produced a cage dynamic model under the oil lubrication condition and studied the influences of cage clearance ratio, bearing load and bearing rotation speed on the cage vibrations. Niu et al. [7] studied the effects of ball-cage pocket contact force and the cage-guiding ring forces on the stable whirl motion of the outer ring guided cage under the solid lubrication conditions. The effect of ball spacing on the whirl
radius was studied too. Nogi et al. [8] developed the theory of cage vibrations and proposed a critical cage friction coefficient formula. The prediction of the formula shows a great coincidence with the experimental results obtained by other scholars. Wen et al. [9] used a bearing test rig and measured the motions of bearing cage in three dimensions. The patterns of cage motion under different operating conditions were obtained. Yang et al. [10] studied the influence of elastohydrodynamics on the cage. They provided a quasi-dynamic model to study the effects of cage clearance on the heat emission. Han et al. [11] proposed a dynamic model of a ball bearing with an unbalanced cage. They investigated the influences of unbalance on trajectories of cage center and cage motions. The experimental results were used to validate the results. Cui et al. [12] established the nonlinear dynamic differential equations of a high-speed cylindrical bearing to investigate the relationship between roller dynamic unbalance and the cage dynamic motion. Cui et al. [13] produced a dynamic model of high-speed cylindrical roller bearing and a finite element model of roller and cage to study the influence of roller dynamic unbalance on cage stress.

In this paper, a multi-body dynamic model of ball bearing with different sizes of local defects is proposed by utilizing MCS. ADAMS software. The pulse waveforms of angular velocity (relative angular velocity of cage for ball bearings with different cases of defects versus to healthy bearings) of cage around Z direction are compared, its corresponding mathematical statistics are compared too. The pulse waveforms of contact forces of ball-to-cage and ball-to-outer ring are compared. Effects of the defect sizes and inner ring speed on the angular velocity of cage are investigated, as well as the relationship of contact force between balls-to-outer ring and balls-to-cage. The results show that the multi-body dynamic model can be applied to study the cage vibrations for a defective ball bearing.

2. Formulas for the contact force and stiffness calculation

2.1. Formulas for the contact force

Since no deformation is allowed in a multi-body dynamic model of ball bearing, an explicit contour recognition method is utilized for the contact detection between the balls and raceways.

The contact forces between balls and raceways are calculated as [14]

\[
F = \begin{cases} 
0 & \delta \geq 0 \\
F_e - F_c = K_c \delta' - F_c & \delta < 0 
\end{cases}
\]  

(1)

where \(F\) is the contact force; \(F_e\) is the elastic force; \(F_c\) is the damping force; \(\delta\) is the distance between the two undeformed point of the ball and raceway; \(K_c\) is the stiffness of the boundary surface interaction; \(e\) is the exponent of the force deformation characteristic, which is usually set as 1.5 for the steel bodies; the damping of the ball bearing is typically in the order of 0.25-2.5×10^{-5} times the linearized stiffness of the bearing [15].

2.2. Formulas for the contact stiffness

As shown in Fig. 1, an elliptical point contact model of two ellipsoids is presented to reveal the calculation process of Hertzian contact stiffness between raceways and balls. \(Q\) is the external force applied on contact body 1.

![Figure 1. Schematic of elliptical Hertzian point contact.](image)
The contact stiffness between two contact bodies is represented as [16]

\[
K_c = \frac{1}{\sqrt{2.7910 \times 10^{-10} K \mu E \pi^2 (\rho_{\alpha I} + \rho_{\alpha II} + \rho_{\beta I} + \rho_{\beta II})}}
\]

(2)

where \(K_c\) is the contact stiffness; \(\rho_{\alpha I}, \rho_{\alpha II}, \rho_{\beta I}\) and \(\rho_{\beta II}\) are principle curvatures of the two contact bodies, subscripts 1 and 2 are representing the contact body 1 and 2, subscripts I and II representing two mutually perpendicular principal planes; \(k\) is an intermediate variable with no meaning; \(K\) and \(E\) are the first and the second order elliptic integral, separately, and they are represented as

\[
K = \int_{0}^{\pi/2} \frac{1}{\sqrt{1-k^2 \sin^2 \phi}} d\phi
\]

(3)

\[
E = \int_{0}^{\pi/2} \sqrt{1-k^2 \sin^2 \phi} d\phi
\]

(4)

The relation between \(K\) and \(F(\rho)\) can be represented as

\[
F(\rho) = \frac{(2-k^2)E - 2(1-k^2)K}{k^2 E}
\]

(5)

where \(F(\rho)\) is an instrumental variable relevant to the dimension of two contact bodies, represented as

\[
F(\rho) = \frac{|\rho_{\alpha I} \rho_{\alpha II} + |\rho_{\beta I} \rho_{\beta II}|}{\Sigma \rho}
\]

(6)

3. Dynamic modelling of a defective ball bearing

On the bases of the geometric parameters of 6308-2RZ ball bearing, as presented in Table. 1, the multi-body dynamic model of a defective ball bearing is modelled.

In this model, a point motion is applied on the mass center of inner ring, as well as a planar joint which is used to connect with ground. It is used to keep inner ring moving in the working plane while rotating around Z axis through centroid. The spring damper system is utilized both in horizontal and vertical directions between inner ring and ground. It is used to represent the supporting of driving shaft. The cage is limited to rotate around Z axis through the centroid of inner ring. It is used to focus on the perturbation of its angular velocity. The torsion spring damper system is utilized in RZ (around Z axis) direction between outer ring and ground, so as to limit the rotational degree of the outer ring around the centroid axis. 24 contact constraints are defined between each ball and the cage, inner ring and outer ring. The defects are all modeled as rectangular shape in the outer raceway by removing a section. A schematic of the multi-body dynamic model is shown in Fig. 2.

Table 1. Parameters of 6308-2RZ bearing

| Parameter                  | Value   |
|----------------------------|---------|
| Inner ring diameter (\(d_i\))/mm | 40      |
| Outer ring diameter (\(D\))/mm     | 90      |
| Ball diameter (\(D_b\))/mm         | 15.081  |
| Pitch diameter (\(d_m\))/mm        | 65      |
| Width (\(C\))/mm                  | 23      |
| Number of Balls (\(Z\))           | 8       |
Figure 2. Schematic of multi-body dynamic model.

4. Results and discussion

In the multi-body dynamic model, a radial load of $Q=150$ N is applied on the outer ring along the $Y$-direction. The effects of defect sizes and inner ring speed on the cage vibrations are studied. The pulse waveforms of the angular velocity of cage are compared, the pulse waveforms of contact force between balls-to-outer ring and balls-to-cage are also compared.

4.1. Effects of defect sizes

The inner ring is keep rotating at the speed of $N_r=1800$ r/min. Figure 3 plots the effects of defect sizes on the angular velocity of cage for the ball bearing with different defect cases. Here, O1, O2 and O3 denote the defect cases with defect length of 1mm, 2mm and 3mm, separately. As shown in Fig. 3(a), the pulse waveforms of angular velocity of cage for the ball bearing with different defect cases can be greatly affected. To plot the relationship between defect size and angular velocity of cage, the root mean square (RMS) and Peak to peak (PTP) value of the pulse waveforms are calculated as given in Fig. 3(b). As shown in Fig. 3(b) the RMS and PTP value increase with the length of defect.

Figure 3. The effects of the defect size on the angular velocity of cage. (a) Pulse waveforms, and (b) RMS value and Peak to Peak value.

4.2. Effect of inner ring speed

Three different speeds of motions, from $N_r=900$ r/min to $N_r=3600$ r/min, are applied on the inner ring. Figure 4 plots the effects of inner ring speed on the angular velocity of cage for the ball bearing with defect case O1. As shown in Fig. 4(a), the pulse waveform of angular velocity of cage under condition of $N_r=3600$ r/min has more perturbations than that under conditions of $N_r=900$ r/min and 1800 r/min; and the maximum amplitude of the pulse waveform under condition of $N_r=3600$ r/min is larger than
that under conditions of \( N_r = 900 \text{ r/min} \) and 1800 \text{ r/min}. Fig. 4(b) plots the RMS and PTP value of the pulse waveforms, the RMS and PTP value increase with increment of inner ring speed.

**Figure 4.** The effect of the inner ring speed on the angular velocity of cage. (a) Pulse waveforms, and (b) RMS value and PTP value.

### 4.3. Relationship between contact forces of ball-to-cage and ball-to-outer ring

As shown in Fig. 5 (a), when the balls #3 and #7 (The balls are numbered clockwise as depicted in Fig. 2) passes through the localized defect O3, the impacts can be clearly observed on the curves Ball#3-to-outer ring and Ball#7-to-outer ring (represent the contact force of ball 3 to outer ring and ball #7 to outer ring). For clear observation, the enlarged view is presented in Fig. 5 (b). As shown in Fig. 5 (b), there are totally three great impacts on the curve Ball#1-to-cage (represents the contact force between ball #1 and cage), and the three impacts are numbered by #1, #2 and #3 orderly. Moreover, a great coincidence between the impacts #1, #3 and the impacts on the curve Ball#7-to-outer ring can be observed, which is better than that between impact #2 and the impact on the curve Ball#3-to-outer ring. This is due to the fact that there is an interval of 90 degree between ball #1 and balls #3 and #7. When balls #3 and #7 passes through the defect, ball #1 is exactly at a horizontal position, then the impact force caused by local defect is perpendicular to the contact area between ball #1 and cage. Moreover, when ball #3 passes through the defect, ball #1 is leaving load region and its slippage increase; inversely, when ball #7 passes through the defect, ball #1 is entering load region and starts to compress tightly with the outer ring.

**Figure 5.** Relationship between contact forces of Ball#1-to-cage and Balls#3, 7-to-outer ring. (a) Pulse waveforms, and (b) enlarged view of (a).

As shown in Fig. 6 (b), similar results can be observed as depicted in Fig. 5 (b). A great coincidence between the impacts #2, #4 and the impacts on the curve Ball#1-to-outer ring can be observed, which is better than that between impacts #1, #3 and the impacts on the curve Ball#5-to-outer ring. When ball #1 passes through the defect, ball #3 is entering load region; when ball #5 passes through the defect, ball #3 is leaving load region.
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Figure 6. Relationship between contact forces of Ball#3-to-cage and Balls#1, 5-to-outer ring. (a) Pulse waveforms, and (b) enlarged view of (a).

5. Conclusions

Influence of a local defect on the cage vibrations for a ball bearing is investigated. Effects of defect size and inner ring speed on the angular velocity of cage are studied by the multi-body dynamic method for a defective ball bearing. The relationship of the contact force between ball-to-outer ring and ball-to-cage is studied. Some conclusions of this study can be drawn: The RMS and PTP value of angular velocity of cage increase with the increment of defect size and inner ring speed. The contact force between one ball and cage is greatly affected by the impacts caused by the near balls (which have intervals of 90 degree refer to the ball). There is a more great coincidence between the impacts on curves ball-to-cage and ball-to-outer ring when the ball is entering load region than that when the ball is leaving load region.

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References

[1] Liu J and Shao Y 2018 Overview of dynamic modelling and analysis of rolling element bearings with localized and distributed faults Nonlinear Dynamics 93(4) 1765-98.
[2] B. Pederson, F. Sadeghi and C. Wassgren 2006 The Effects of Cage Flexibility on Ball-to-Cage Pocket Contact Forces and Cage Instability in Deep Groove Ball Bearings SAE Technical Paper 43–54.
[3] N. Weinzapfel and F. Sadeghi 2009 A discrete element approach for modeling cage flexibility in ball bearing dynamics simulations ASME J Tribol 131, 021102.
[4] A. Ashtekar and F. Sadeghi 2012 A New Approach for Including Cage Flexibility in Dynamic Bearing Models by Using Combined Explicit Finite and Discrete Element Methods Journal of Tribology 134(44). 041502.
[5] Ye Z and Wang L 2013 Cage Instabilities in High-Speed Ball Bearings Applied Mechanics and Materials 278-280 3-6.
[6] Ye Z and Wang L 2015 Effect of external loads on cage stability of high-speed ball bearings Proc IMechE, Part J: J Engineering Tribology 229 1300–18.
[7] Niu L, Cao H and He Z 2016 An investigation on the occurrence of stable cage whirl motions in ball bearings based on dynamic simulations Tribology International 103 12-24.
[8] T. Nogi, K. Maniwa and N. Matsuoka 2018 A dynamic analysis of cage instability in ball bearings Journal of Tribology 140(1) 011101.
[9] Wen B, Ren H, Zhang H and Han Q 2017 Experimental Investigation of Cage Motions in an Angular Contact Ball Bearing Proc IMechE Part J 231(8) 1041-55.
[10] Yang Z, Yu T, Zhang Y and Sun Z 2017 Influence of cage clearance on the heating characteristics of high-speed ball bearings Tribol Int 105 125–34.
[11] Han Q, Wen B, Wang M and Deng S 2017 Investigation of cage motions affected by its unbalance in a ball bearing *Proc IMechE Part K: J Multi-body Dynamics*, 0(0) 1–14.

[12] Cui Y, Deng S and Zhang W 2017 The impact of roller dynamic unbalance of high-speed cylindrical roller bearing on the cage nonlinear dynamic characteristics *Mechanism & Machine Theory* 118 65-83.

[13] Cui Y, Deng S, Ni Y and Chen G 2018 Effect of roller dynamic unbalance on cage stress of high-speed cylindrical roller bearing *Industrial Lubrication and Tribology* 70(9) 1580-9.

[14] MSC. Software 2013 *Adams/View help - Adams 2013*, (America: MSC. Software).

[15] Liu J and Shao Y 2017 Dynamic modeling for rigid rotor bearing systems with a localized defect considering additional deformations at the sharp edges *Journal of Sound and Vibration* 398 84-102.

[16] T. A. Harris and M. N. Kotzalas 2006 *Essential Concepts of Bearing Technology 5th Ed*, (Beijing :China Machine Press).