Dynamic characteristics of high-speed railway vehicle with axle box bearing faults

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Abstract. Based on the motion characteristics of axle box bearings and taking high-speed railway vehicles as the research object, a vertical dynamic model of 17-DOF car body-bogie-axle box bearings-wheel set is proposed. Aiming at the surface fault of axle box bearing, the New-MarK fast numerical integration method is used to simulate and analyze. The dynamic characteristics of railway vehicle under early failure of axle box bearing on inner ring and outer ring raceway surface are studied. The vibration characteristics of axle box bearings under different fault degrees and different vehicle speeds (axle box bearing speed) are analyzed. The changes of vehicle dynamic performance and vibration characteristics under fault evolution are discussed. The analysis results show that the vibration of car body and bogie is little affected by the early fault of axle box bearing; with the deepening of the fault degree, the vertical vibration acceleration and displacement of axle box bearing are increasing; with the increase of vehicle speed, the motion state of axle box bearing undergoes a change from periodic motion, quasi-periodic motion to chaotic motion. Finally, the dynamic simulation results of axle box bearing fault evolution in high-speed vehicle system are validated on the experimental platform. The research work has certain reference value for vehicle condition tracking and monitoring.

1. Introduction

As a key component of railway vehicles, axle box bearings connect the car body, bogie and wheel set, bearing the vertical load of the car body, the traction force of the wheel set and the wheel-rail force. Under the harsh conditions of high speed and strong noise for a long time, the axle box bearings alarm frequently and failures occur from time to time, which are directly related to the safe and stable operation of the vehicle[1-2]. Early vibration signals of axle box bearings are disturbed by transmission paths and environmental noise, so it is difficult to detect and diagnose them in time. Moreover, axle box bearings do not disintegrate in daily vehicle operation and maintenance, and only do external inspection [3-5]. Therefore, it is necessary to study the dynamic characteristics of axle box bearings in high-speed vehicle system and to analyze the motion mechanism of vehicle system under the condition of axle box bearings failure.

In recent years, many achievements have been made in the research of high-speed vehicle dynamics system modeling. In these coupled models, axle box bearings and wheels are often combined or simply treated as damping systems [6-8]. Scholars at home and abroad have also made a lot of progress in the dynamic analysis of high-speed bearings: a dynamic model for axle box bearings of high-speed trains has been established by using different dynamic simulation software [9-10], the calculation of impact force due to faults and the mathematical model under surface defects have been studied, and the influence of surface defects on the dynamic performance of bearings has been analyzed [11-
13]. These studies rarely take into account the dynamic characteristics of high-speed vehicles under the fault evolution of axle box bearings, and can not reflect the interaction between axle box bearings and vehicle systems.

Based on the research of scholars at home and abroad, the vertical dynamic model of vehicle body-bogie-axle box-wheel system is established according to the characteristics of axle box bearings. The motion and dynamic characteristics of high-speed vehicle system in different fault evolution of axle box bearings are simulated and analyzed.

2. Methods

2.1. Force analysis of bearing

Double-row tapered or cylindrical roller bearings are usually used in axle box bearings of railway vehicles. Taking double row tapered roller bearings as the object of analysis, assuming that the internal and external of double row bearings keep the same motion state, it is equivalent to a bearing object for analysis [14-16]. The radial load distribution of tapered roller bearings is shown in Figure 1 below.

![Figure 1. Radial load distribution of the bearing.](image)

The initial radial clearance and preload are not considered. The radial load area of the bearing is half of the circumference. Each roller’s contact load can be divided into radial load and axial load. Under the action of the load, the radial displacement \( \delta_r \) and axial displacement \( \delta_a \) of the outer ring relative to inner ring are generated. The relative displacement \( \delta_j \) between outer ring and roller at the angle \( \phi_j \) to radial displacement should be:

\[
\delta_j = \delta_r \cos \phi_j + \delta_a \sin \alpha_e
\]  

Then, the radial load \( F_r \) and axial load \( F_a \) of bearing should be:

\[
F_r = \sum_{j=1}^{n} Q_j \cos \alpha_e \cos \phi_j
\]

\[
F_a = \sum_{j=1}^{n} Q_j \sin \alpha_e
\]

The contact load \( Q_j \) is \( K\delta_j \) with each roller when \( \delta_j \) is bigger than zero. The raceway contact stiffness coefficient \( K \) can be calculated with Palmgren Formula.

\[
K = 8.06 \times 10^5 \alpha_{ef} \times \left[ 1 + \left( \sin (\alpha_e + \alpha_f) / \sin (\alpha_e + \alpha_f) \right)^{0.9} \cos (\alpha_e - \alpha_f) \right]^{10}^{9}
\]

Among them, \( l \) is the effective length of the roller; \( \alpha_e, \alpha_e \) and \( \alpha_i \) are respectively the contact angles of the roller with outer ring, inner ring and gear edge.

2.2. Analysis of the surface fault of the bearing raceway

After a long period of operation, rolling bearings often show point defects in the early stage of roller raceway cracks, peeling and defects. Expansion diagram of contact between roller and defect surface is shown in Figure 2:
Figure 2. The contact diagram of roller and raceway defects.

It can be seen from the figure that the roller does not contact the bottom of the defect in the early stage of failure, and the displacement of the roller can be expressed as follows:

\[
h = \begin{cases} 
(D_i - r) (1 - \sin \theta) & 0 < \theta \leq \arcsin(\frac{w}{D_i}) \quad \text{outer} \\
(D_o - r) (1 - \sin \theta) & 0 < \theta \leq \arcsin(\frac{w}{D_o}) \quad \text{inner}
\end{cases}
\]

Among them, \(D_o\), \(D_i\), \(r\), and \(w\) are outer ring diameter, inner ring diameter, average roller radius and defect width respectively. Angle \(\theta\) between roller and defect is the function of rotational angular velocity.

2.3. A 17 degrees vertical dynamic model of high-speed railway vehicle system

In the process of train operation, the axle box bearings are stimulated by the wheel set under the influence of track excitation, and bear the weight of the whole body and the impact force transmitted by the frame. Taking the vehicle [17] as the research object, taking vertical vibration as the main factor, considering the floating and sinking, rolling and nodding motions of the body, front and rear bogies, four axle box bearings and four wheel sets, the vertical dynamic model of the vehicle with 17 degrees of freedom is established as shown in Figure 3 below.

Figure 3. Vertical model of vehicle system.

Assuming that the car body and bogie are rigid bodies, they are connected by secondary suspension. The outer ring of the bearing is fixed relative to the axle box and is connected with the bogie through a series of suspensions. The inner ring of the bearing rotates with the wheel set and generates force between the roller and the outer ring of the bearing. Wheel set is stimulated by wheel/rail. The 17 degrees vertical dynamic equations of the whole railway vehicle are given below.

Wheel sets’ vertical motion:

\[
M_{wz} + 2 p_z(t) - M_{wz}g = F_{wz}(t)
\]

Axle box bearings’ vertical motion:

\[
M_{i1} \ddot{Z}_{i1} + C_i \dot{Z}_{i1} + K_i Z_{i1} - C_v \dot{Z}_{i1} - K_v Z_{i1} - C_p \dot{Z}_{i1} - K_p Z_{i1} = M_{i1}g
\]

Bogies’ vertical motion:

\[
M_{o1} \ddot{Z}_{o1} + (C_o + 2C_v) \dot{Z}_{o1} + (K_o + 2K_v) Z_{o1} - C_v \dot{Z}_{o1} - K_v Z_{o1}
- C_p (\dot{Z}_{o1} + \dot{Z}_{o2}) - K_p (Z_{o1} + Z_{o2}) - C_{p1} \dot{\beta}_x - K_{p1} \beta_x = M_{o1}g
\]

\[
M_{o2} \ddot{Z}_{o2} + (C_o + 2C_v) \dot{Z}_{o2} + (K_o + 2K_v) Z_{o2} - C_v \dot{Z}_{o2} - K_v Z_{o2}
- C_p (\dot{Z}_{o2} + \dot{Z}_{o3}) - K_p (Z_{o2} + Z_{o3}) - C_{p2} \dot{\beta}_x - K_{p2} \beta_x = M_{o2}g
\]
Bogies’ nodding motion:

\[ \begin{align*}
I_j \ddot{\phi}_j + 2C_j \dot{\phi}_j \dddot{\phi}_j + 2K_j \dot{\phi}_j \dddot{\phi}_j - C_j \phi_j \dddot{\phi}_j + C_j \phi_j \dddot{\phi}_j - K_j \phi_j \dddot{\phi}_j + K_j \phi_j \dddot{\phi}_j = 0 \\
I_j \ddot{\beta}_j + 2C_j \dot{\beta}_j \dddot{\beta}_j + 2K_j \dot{\beta}_j \dddot{\beta}_j - C_j \beta_j \dddot{\beta}_j + C_j \beta_j \dddot{\beta}_j - K_j \beta_j \dddot{\beta}_j + K_j \beta_j \dddot{\beta}_j = 0
\end{align*} \]

(10)

Bogies’ side rolling motion:

\[ \begin{align*}
J_j \dot{\phi}_j + K_{st} \dot{\phi}_j - (Z_{s1} - Z_{s2}) / l_j + C_{st} \phi_j - (\dot{Z}_{s1} - \dot{Z}_{s2}) / l_j + K_{st} \phi_j = 0 \\
+ C_{st} (\phi_j - \phi_j) + K_{st} (Z_{s1} + \phi_j l_j - Z_{s2}) - K_{st} (Z_{s2} - \phi_j l_j - Z_{s1}) \\
+ K_{st} (Z_{s1} - \phi_j l_j - Z_{s2}) - K_{st} (Z_{s2} - \phi_j l_j - Z_{s1}) - M_{st} \phi_j = 0
\end{align*} \]

(11)

Body’s vertical motion:

\[ M_c \dddot{Z}_c + 2C_c \dot{Z}_c + 2K_c Z_c - C_c \dddot{Z}_c - K_c \dddot{Z}_c = M_c \ddot{g} \]

(14)

Body’s nodding motion:

\[ \begin{align*}
I_j \ddot{\phi}_j + 2C_j \dot{\phi}_j \dddot{\phi}_j + 2K_j \dot{\phi}_j \dddot{\phi}_j - C_j \phi_j \dddot{\phi}_j + C_j \phi_j \dddot{\phi}_j - K_j \phi_j \dddot{\phi}_j + K_j \phi_j \dddot{\phi}_j = 0
\end{align*} \]

(15)

Body’s side rolling motion:

\[ \begin{align*}
J_j \dot{\phi}_j + K_{st} \dot{\phi}_j - (Z_{s1} - Z_{s2}) / l_j + C_{st} \phi_j - (\dot{Z}_{s1} - \dot{Z}_{s2}) / l_j + K_{st} \phi_j = 0 \\
+ C_{st} (\phi_j - \phi_j) + K_{st} (Z_{s1} + \phi_j l_j - Z_{s2}) - K_{st} (Z_{s2} - \phi_j l_j - Z_{s1}) \\
+ K_{st} (Z_{s1} - \phi_j l_j - Z_{s2}) - K_{st} (Z_{s2} - \phi_j l_j - Z_{s1}) - M_{st} \phi_j = 0
\end{align*} \]

(16)

\[ M_{wi}, M_{bi}, M_{tj} \text{ and } M_c \text{ are the mass of wheel set, axle box bearing, bogie and body, respectively (i=1..4; j=1...2; the same below). } Z_{wi}, Z_{bi}, Z_{tj} \text{ and } Z_c \text{ are the vertical displacement of wheel set, axle box bearing, bogie and body, respectively. } \beta_j \text{ and } \beta_j \text{ are the nodding displacement of bogie and body, respectively. } \Phi_j \text{ and } \Phi_j \text{ are the side rolling displacement of bogie and body, respectively. } F_{bi}(t) \text{ and } p_i(t) \text{ stand for the excitation force function at each wheel set and the vertical force of single-side wheel track, respectively. } l_j \text{ and } l_i \text{ are the half of the length between bogie centers and the half of bogie fixed axles distance, respectively. } C_{pj}, K_{pj}, C_{st} \text{ and } K_{st} \text{ are the vertical damping, vertical stiffness, side rolling angle damping, and side rolling angle stiffness of primary suspension, respectively. } C_{st}, K_{st}, C_{st} \text{ and } K_{st} \text{ are the vertical damping, vertical stiffness, side rolling angle damping, and side rolling angle stiffness of secondary suspension, respectively. } I_j \text{ and } J_j \text{ represent the nodding inertia and side rolling rotation inertia of body, respectively. } I_j \text{ and } J_j \text{ represent the nodding inertia and side rolling rotation inertia of body, respectively. } h_2 \text{ is the distance between the centroid of body and that of frame; } h_j \text{ means the distance between the centroid of frame and the wheel set axis center.}
\]

3. Results and Discussion

Trigonometric series method is used to simulate the track spectrum of grade 6 in the United States as the input of track irregularity. New-Mark compensation method is used to simulate the vertical vibration characteristics of vehicle under bearing failure. The values of vehicle model parameters can be found in the References [17].

3.1. Dynamic characteristics of fault-free system

Taking the vehicle running speed as 100 km/h and the axle box bearing without fault, the vertical vibration of the vehicle system running in a straight line is analyzed, as shown in Figure 4 below.

From the figure, it can be seen that the vertical vibration acceleration of the front and rear bogies of the vehicle system is basically the same, and the vertical vibration acceleration of the front left axle box and the front right axle box are the same, which verifies the correctness of the model.
3.2. Dynamic characteristics under different bearing faults

The width of the raceway defect of the front left axle box bearing is 0.5 mm and the running speed of the vehicle is 100 km/h. Vertical vibration characteristics of vehicle system in straight line operation are simulated and calculated respectively when the raceway defect occurs in outer raceway and inner raceway. Data comparison is shown in Tables 1 below.

Table 1. Maximum vertical vibration acceleration of vehicle parts under different faults.

| Fault located       | No       | Inner ring | Outer ring |
|---------------------|----------|------------|------------|
| Body                | 0.297    | 0.298      | 0.298      |
| Front bogie         | 10.174   | 10.191     | 10.187     |
| Rear bogie          | 9.671    | 9.813      | 10.010     |
| Left-front axle box | 30.373   | 40.013     | 46.323     |
| Right-front axle box| 31.075   | 34.156     | 34.323     |

It is observed from Tables 1. It can be seen that the surface faults of inner raceway and outer raceway have little effect on the vertical vibration acceleration of vehicle body and front/rear bogie, which is mainly caused by the primary and secondary suspension systems of vehicles. The vertical vibration of axle box bearings without fault is slightly increased due to the influence of coaxial fault axle box bearings. Affected by faults, the maximum vertical vibration acceleration and the average acceleration amplitude of the faulty side axle box bearings increase significantly, and the vibration is aggravated, regardless of whether the faults occur in the inner raceway or the outer raceway.
3.3. The effect of bearing speed on system dynamics

Taking the outer raceway fault as an example, the front left axle box bearing has a raceway defect of 0.8mm width, and the vehicle keeps a straight line running at a uniform speed. The influence of bearing speed on the vertical vibration displacement amplitude of the fault side axle box bearing is analyzed, as shown in Figure 5 below.

![Figure 5](image)

As it can be seen from Figure 5, that the vertical vibration displacement amplitude of axle box bearings increases significantly with the increase of bearing rotation speed (vehicle running speed). Poincare mapping cross-sectional diagrams of vibration response of axle box bearing at 170, 330, 600 and 1500 rpm are taken respectively, as shown in Figure 6 below.

![Figure 6](image)

As shown in Figure 6, that there is only one fixed point (0.010911,0.0159945) in the figure when the speed of axle box bearing is 170 rpm/s; when the speed is 330 rpm/min, the Poincare map increases to three discrete points, and the vibration displacement basically remains unchanged, and the maximum point of vibration speed increases to 0.02826 m/s; it can be seen that the axle box bearing system
is in a periodic motion state under these two rotational speeds. When the rotational speed is 600 rpm, the points in the Poincare map show closed curves, and the system is in quasi-periodic state. The vibration displacement did not increase significantly. The vibration velocity was linear symmetrical with respect to y=0, and the maximum amplitude increased to 0.1603 m/s. The vertical vibration response of axle box bearings increases when the rotating speed is 1500 rpm. The vibration displacement is extended to (-0.01565, 0.01763). The vibration velocity is extended to (-2.20438, 3.0722). Poincare mapping points present discrete state, and the system enters chaotic state.

3.4. Vibration characteristics analysis of bearing fault evolution system
The vehicle runs at a uniform speed of 100 km/h in a straight line. The raceway of axle box bearing with left front position has outer raceway fault. The fault width increases with the speed of \([0, \pi/2]\) sinusoidal function. The variation of vibration acceleration kurtosis of axle box bearing under fault evolution is analyzed as shown in Figure 7 below.

![Figure 7. Kurtosis under the evolution of bearing.](image)

From the change of vibration kurtosis of axle box under fault evolution, it can be seen that the vibration kurtosis is 5.6989 when the defect width is 0.3127 mm. When the defect width reaches 0.7650, the vibration kurtosis value is 17.3244. With the increase of defect, the kurtosis value of vibration shock is increased rapidly. When the defect width reaches 1.9752 mm, the vibration kurtosis will increase to 594.0875.

The vertical vibration of the bearing of the fault side axle box is intercepted when the fault defect width is 0.25mm, 0.50mm and 0.75mm respectively. The comparison of vibration acceleration of axle box under three fault degrees is shown in Figure 8.

![Figure 8. Vibration of axle box under different fault width.](image)

From Figure 8, it can be seen that with the deepening of the fault degree, the vibration acceleration of the fault axle box bearing increases, and the vertical displacement increases. The aggravation of
vibration will accelerate the deterioration of the fault. Therefore, real-time monitoring and accurate diagnosis of early failure are very important.

4. Experimental verification

In order to verify the validity of theoretical analysis, RD2197726 wheel set rolling bearings, which are commonly used in 60t railway freight cars, are taken as the research object. The running-in test bench is tested and analyzed with railway freight wheel. The faults of axle box bearing used in the experiment are surface faults machined by machine. Main Working Conditions and Characteristics of Bearing Failure Dynamics Experiments are shown in Table 2 below.

| RPM(r/min) | Fault width(mm) | Sampling frequency(Hz) | Sampling number |
|-----------|-----------------|------------------------|----------------|
| 467       | 0.5             | 5120                   | 10204          |
| 467       | 1               | 5120                   | 10204          |
| 715       | 1               | 5120                   | 10204          |
| 467       | 1.5             | 5120                   | 10204          |

After many vibration tests, the bearing vertical vibration signal data are selected for analysis. At different rotational speeds, the Poincare map of bearing outer ring faults with fault width of 5 mm is shown below.

![Poincare map under different rotation speeds.](image)

It can be seen from figure 9 that the Poincare mapping points of bearing vibration response increase with the increase of rotational speed under the same fault degree. The variation trend is similar to the variation characteristics of simulation results, which verifies the validity of theoretical research to a certain extent.

Under the same rotational speed and other experimental conditions, the bearing with different fault degrees is tested, and the vibration acceleration response and vibration kurtosis of the bearing are obtained as shown in Figure 10 and figure 11 below.

![Figure 10](image)  (a) 467 RPM

![Figure 11](image)  (b) 715 RPM

From figure 10 and figure 11, it can be seen that with the deepening of the fault degree, the vibration acceleration amplitude of the bearing increases continuously, and the vibration kurtosis of the bearing increases sharply, which indicates that the bearing fault has brought severe impact.
Figure 10. Acceleration under different severities of faults.

Figure 11. Kurtosis under different severities of faults.

5. Conclusions
Based on the analysis of the force acting on axle box bearings of railway vehicles, a vertical dynamic model of 17 DOF railway vehicle with body-bogie-axle box bearings-wheelset is established. The influence of fault type, fault degree and running speed on the vertical vibration of vehicle system under the early fault evolution of axle box bearing is simulated and analyzed. The simulation analysis is validated on the experimental platform. The results show that the early failure of axle box bearings has no obvious influence on the vibration of car body and bogie, but has a great influence on axle box bearings, especially on the fault side. With the increase of the rotational speed of the fault axle box bearing (vehicle running speed), the motion state of the axle box bearing develops from periodic motion to quasi-periodic motion. When the speed continues to increase, the system evolves into chaotic motion. With the deterioration of the fault degree, the vertical vibration of axle box bearings intensifies, while the vibration of axle box accelerates the deterioration process of the fault. Identifying the dynamic characteristics of axle box bearing under early fault is the basis of tracking and monitoring the running state of high-speed train.

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