Explicit Dynamics Simulation of High-Speed Railway Bearing Based On ANSYS/LS-DYNA

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Abstract. A solid model of a high-speed train axle box bearing was built by using 3D modeling software. On the platform of ANSYS/LS-DYNA, the combined dynamic and finite element analysis method is used to apply the combined radial and axial loads to the high-speed rail bearing. In order to demonstrate the dynamic contact characteristics of the bearing during operation, two operating conditions were designed. One is running at a uniform speed in a straight line, the other is running at an assumed speed with a minimum turning radius. The results show that the stress distribution, velocity and acceleration of bearing components under two different working conditions are obviously different. The research can provide reference for the design, selection and optimization of domestic high-speed rail bearings.

1. Introduction

High-speed railway has gradually become the preferred transportation means for people, and the use of high-speed rail transportation has become an important development direction of the transportation industry in the world [1]. Axle box bearing (hereinafter referred to as high-speed rail bearing) is one of the most important components of high-speed train, which is related to the safe operation of high-speed train. The dynamic characteristics of the high-speed rail bearing process directly affect the service life of the bearing, and improving the life and performance of the bearing has great significance for the stable and safe driving of the high-speed rail vehicle. Therefore, it is necessary to study the changes of speed, acceleration, displacement and stress of various parts of high-speed rail bearings during operation. According to the changes of various parts, the position of fatigue failure and instability during bearing operation is judged, so as to provide reference for structural optimization design of high-speed rail bearings.

At present, double row tapered roller bearings are mostly used in high-speed railway axle box bearings, which mainly bear combined radial and axial loads. Literature [2] analyzed the influence of bearing outer contact angle, number of rollers and roller length on bearing life. The literature [3] studied the influence of the geometry of the contact between the tapered roller and the raceway on the stress distribution, stress level and motion state of the bearing. In literature [4], W C Tang et al analyzed the fault characteristics during bearing operation. The above literature has carried out dynamic analysis of
the bearing, and the dynamic characteristics of the bearing under radial and axial combined loads have been rarely studied. In this paper, a high-speed axle box bearing with a speed of 300km/h is taken as an example. Based on the dynamic finite element method, the motion simulation of the bearing under two different working conditions is realized by using the finite element analysis software ANSYS/LS-DYNA in the case of radial and axial combined loads. The result of this simulation, on the one hand, provides a certain degree of explanation for the fatigue damage of the bearing during operation, and on the other hand can provide reference for the modeling design of high-speed railway bearing.

2. Explicit Dynamic Finite Element Method

The explicit integration method is also called the closed solution method or the predictive solution method. The explicit algorithm does not have a convergence problem because it requires a small step size and a faster solution speed. It does not require equilibrium iterations, nor directly solves the tangent stiffness, and can demonstrate the speed advantage of explicit dynamics solution when there are few elements. And for collision problems, this short-term high-level nonlinear problem, the explicit integration method will be more suitable. Therefore, this paper uses explicit integration method. ANSYS/LS-DYNA is solved by this method.

The LS-DYNA explicit dynamics analysis uses a central difference method. In dynamic finite element analysis, the dynamics governing equation of the system is:

\[ M \ddot{a}_t + C \dot{a}_t + K a_t = Q_t \]  

(1)

Here \( \ddot{a}_t \) is the acceleration vector of the system node at time \( t \), \( \dot{a}_t \) is the velocity vector of the system node at time \( t \), \( a_t \) is the vector of the system node at time \( t \), \( M \) is the mass matrix, \( C \) is the damping matrix, \( K \) is the stiffness matrix, \( Q_t \) is the node load vector.

In ANSYS/LS-DYNA, the equation of motion is solved using the direct integral center difference scheme. In the central difference method, the acceleration and velocity can be expressed as displacements:

\[ \ddot{a}_t = \frac{1}{\Delta t^2} (a_{t-\Delta t} - 2a_t + a_{t+\Delta t}) \]  

(2)

\[ \dot{a}_t = \frac{1}{2\Delta t} (-a_{t-\Delta t} + a_{t+\Delta t}) \]  

(3)

Where \( \Delta t \) is the time step. The recursive formula of the displacement of each discrete time point can be deduced by the formulas (1), (2) and (3):

\[ (\frac{1}{\Delta t^2} M + \frac{1}{2\Delta t} C)a_{t+\Delta t} = Q_t - (K - \frac{2}{\Delta t^2} M)a_t - (\frac{1}{\Delta t^2} M - \frac{1}{2\Delta t} C)a_{t-\Delta t} \]  

(4)

The stability conditions for the solution of the central difference method are:

\[ \Delta t \leq \Delta t_{cr} \leq \frac{T_n}{\pi} \]  

(5)

Where \( T_n \) is the minimum natural vibration period of the finite element system; \( \Delta t_{cr} \) is the critical value.
After the initial conditions of the element motion are given, the displacement of each element at a certain point in time can be solved by equation (4). Then the stress, strain and acceleration of each element can be obtained [5, 6, 7].

3. Dynamic modeling of high-speed railway bearing

3.1. Geometry parameters of bearing
Take the double-row tapered roller bearing (Figure 1) used in a high-speed railway train with a speed of 300km/h or more. The basic structural parameters are shown in Table 1 below:

![Figure 1. Double-row tapered roller bearing in high-speed railway.](image)

**Table 1.** Parameters of double-row tapered roller bearings

| Parameter                                      | Numerical value |
|-----------------------------------------------|-----------------|
| Nominal outer diameter $D$(mm)                | 240             |
| Nominal inner diameter $d$(mm)                | 130             |
| Bearing width $T$(mm)                         | 160             |
| Diameter of the big end of the roller $D_w$(mm)| 27.9            |
| Maximum diameter of outer raceway $D_2$(mm)   | 226             |
| Contact angle $\alpha$ (°)                    | 9°50’           |
| Nominal width of inner ring $B$(mm)           | 72              |
| Nominal width of outer ring $C$(mm)           | 80              |
| Length of roller $l$(mm)                      | 50              |
| Number of rollers $Z$                         | 2x17            |
| Half cone angle of roller $\phi$ (°)          | 1°12’30”        |

3.2. Establishment of the finite element model
Since the load distribution of the two rows of roller rollers of the double row tapered roller bearing is uneven after the load is applied, the value of the inappropriate detail portion may cause stress concentration, which causes fatigue peeling. So, establishing an accurate model is a key factor in ensuring accurate finite element simulation. The overall model of the high-speed railway bearing is taken for display dynamic analysis. The three-dimensional model of the bearing is built by SolidWorks software and input into the pre-processor of ANSYS/LS-DYNA in the format of .x_t file through the standard amount of graphic input interface. The 3D model imported into ANSYS/LS-DYNA is shown in Figure 2.
In view of the kinematics of high-speed rail bearings, this paper simplifies the high-speed rail bearing model as follows [8]:

1. The influence of bearing chamfers and edges on the internal stress distribution of the bearing is not considered;
2. The effects of bearing diameter, axial clearance and grease are not considered;
3. Regardless of the nonlinearity of the bearing material, it is assumed that the bearing rolling elements, the inner and outer ring, the spacer and the cage are all linear materials.

Material parameters of relevant parts and components of high-speed rail bearings are shown in Table 2. It should be noted that the uniform unit system should be used when defining the material properties. Incorrect units not only affect the material response, but also affect the contact stiffness of the material.

### Table 2. Material properties of parts and components.

| Part name       | Material name                              | Density($\text{kg/m}^3$) | Elastic modulus (Pa) | Poisson's ratio |
|-----------------|--------------------------------------------|---------------------------|----------------------|-----------------|
| Inner and outer ring | GCr15                                      | $7.85 \times 10^3$        | $205 \times 10^{11}$ | 0.29            |
| Roller          | GCr15                                      | $7.85 \times 10^3$        | $205 \times 10^{11}$ | 0.29            |
| Cage            | Glass fiber reinforced nylon 66            | $1.15 \times 10^3$        | $8.3 \times 10^{11}$ | 0.28            |
| Spacer ring     | 40 steel                                   | $7.85 \times 10^3$        | $206 \times 10^{11}$ | 0.3             |

3.3. Meshing

In the process of meshing, the 8-node SOLID164 body unit is selected, and the combination of sweeping, mapping and free partitioning is adopted. When meshing, the grid of the contact area is densified, which means that the mesh of the inner and outer ring raceways, rollers and cages is denser than the non-contact area, so as to improve calculation accuracy and reduce calculation amount. When the bearing is actually running, the outer ring is fixed in the bearing housing, and the inner ring is connected with the rigid shaft. Therefore, we can set the inner surface of the inner ring and the outer surface of the outer ring as rigid surfaces. Since the SOLID 164 unit has no rotational freedom, the inner surface of the inner ring and the outer surface of the outer ring are both defined as a rigid surface SHELL 163 thin shell unit, wherein the inner surface of the inner ring is used to apply an external load and a rotational speed. In addition, it is necessary to define the shear factor, the number of integration points, and the thickness of the shell unit of the thin shell unit by a real constant. The finite element model of high-speed railway bearing is shown in Figure 3.
3.4. Contact pair setting

Set the contact to automatic face-to-face contact. After the bearing is working normally, the friction factor attenuation coefficient caused by various factors is ignored. According to the working characteristics of the high-speed bearing, three sets of contact pairs are established between the outer surface of the roller and the inner ring, the inner surface of the roller and the outer ring, and the roller and the cage.

The static friction coefficient and dynamic friction coefficient were selected by considering the materials of bearing components and actual working conditions, as shown in table 3.

Table 3. Contact friction coefficient between bearing parts.

| Friction type          | Roller - the outer surface of the inner ring | Roller - the inner surface of the outer ring | Roller-cage pocket |
|------------------------|---------------------------------------------|---------------------------------------------|--------------------|
| Static friction coefficient $f_s$ | 0.12                                       | 0.12                                       | 0.002              |
| Dynamic friction coefficient $f_d$  | 0.08                                       | 0.08                                       | 0.001              |

3.5. Applying constraints and loads

According to the installation and working conditions of the high-speed rail bearing, the RBO conditions and loads of the bearing applied in the LS-DYNA are as follows:

(1) Apply a radial force to the inner surface of the inner ring to simulate the actual radial load.
(2) Apply an axial force to the inner surface of the inner ring to simulate the actual axial load.
(3) Applying the rotate speed to the inner surface of the inner ring.
(4) Apply full restraint to the outer surface of the outer ring to simulate installation in the bearing housing.
(5) The X and Y rotation degrees of freedom are constrained to the inner surface of the inner ring to simulate the inner ring and the shaft.

In the case of explicit kinetic analysis, axial forces, radial forces and rotational speeds are functions of time, and sudden loads should be avoided as much as possible.

This paper simulates two operating conditions, where the force and speed conditions are as follows:

Working condition 1: Corresponding to the train running on the straight road, the bearing is only subjected to the radial load of 21KN due to gravity. The maximum speed of the bearing is 2019r/min.

Working condition 2: Corresponding to the train turning with the minimum turning radius, the axial force generated by the centrifugal force is 4KN due to the radial load and the axial load.

The bearing load conditions are set as shown in Table 4:
Table 4. Bearing load table

| Time T(s) | Radial force $F_r$(N) | Axial force $F_a$(N) | Speed of bearing n(r/min) |
|-----------|------------------------|----------------------|---------------------------|
| 0         | 21000                  | 0                    | 0                         |
| 0.002     | 21000                  | 0                    | 1738                      |
| 0.01      | 21000                  | 0                    | 1738                      |
| 0.02      | 21000                  | 4000                 | 1738                      |
| 0.03      | 21000                  | 4000                 | 1738                      |
| 0.04      | 21000                  | 4000                 | 1738                      |

3.6. LS-PREPOST post processing

After applying the boundary conditions and loads to the model of high-speed rail bearing, set the output format to d3plot binary file for post-processing analysis by LS-PREPOST. Set the number of output steps to 1000 and the solution time to 0.045s. The calculation conditions refer to Section 2.5, which is divided into normal straight driving conditions and turning conditions.

4. Explicit dynamics results and analysis of high-speed rail bearing

4.1. Contact stress distribution of bearing

It can be seen from Table 4 that 0-0.01s is the working condition 1, and 0.01s-0.04s is the working condition 2. In order to compare the contact stress of the bearing and components under the two working conditions, the analysis results of ANSYS/LS-DYNA are selected at 0.01s and 0.03s. Figure 4 shows the stress nephogram of the bearing inner ring at 0.01s. At this time, the maximum contact stress of the bearing inner ring appears at unit number 44674, which is 798.529 MPa. Figure 5 shows the stress nephogram of the bearing inner ring at 0.03s. At this time, the maximum contact stress of the bearing inner ring appears at unit number 40955, which is 451.428 MPa.

Figure 4. Stress nephogram of inner ring in 0.01s
Comparing Figure 4 with Figure 5, we can see that the stress of the two sets of rollers is basically the same when the roller is only subjected to radial load at 0.01s. At 0.03 s, when the roller is subjected to both radial and axial loads, the roller group with the axial component force and the axial load direction is compacted, and the other roller is relaxed. Therefore, the stress distribution is eccentric. Similarly, we compare the stress distribution of the outer ring under two different working conditions and get similar conclusions [9].

4.2. Contact stress curve of elements
This section takes the bearing outer ring and roller as examples to analyze the stress curve of the element which changes continuously with time. Three elements are selected in the bearing area of the outer ring, the element numbers are 2098, 2458 and 2298 respectively. They are different in distance from the contact area of the roller and outer ring. In these three elements, 2098 element is the farthest from the contact area and 2458 element is the closest to the contact area. The stress-time history curves of these three elements are plotted, as shown in Figure 6.

Figure 5. Stress nephogram of inner ring in 0.03s

Figure 6. Stress curve of elements on the outer ring.
As can be seen from Figure 6, the stress values of the same element on the outer ring of the bearing vary with time, and the stress values of different elements on the outer ring are different. The closer the unit is to the contact area of the roller, the greater the stress value. From the stress-time curve, the maximum stress of the outer ring changes sharply. The dynamic change of stress affects the fatigue life of the bearing. The simulation results show that the outer ring is a fragile component, and the outer ring raceway may be pitting, which is consistent with the actual situation.

Similarly, in order to analyze the stress changes on the rollers, six elements at different positions on the same roller are selected, and the element numbers are 2098, 2458, 2298, 28031, 28206, and 28202, respectively. Among them, 2098 element are the farthest from the contact area, and 28202 is the closest to the contact area. The stress-time curves for the six elements are shown in Figure 7.

![Figure 7. The stress-time curves for the six elements on the roller.](image)

As shown in Figure 7, according to the positions of the selected six units, it can be seen that the unit closest to the contact area of the roller and the ferrule has the largest stress value; the farther away from the contact area of the ferrule, the smaller the stress value. The maximum stress on the roller is greater than the maximum stress on the outer ring, which proves that the roller is also prone to damage in the contact area.

4.3. Kinematics characteristics of nodes

![Image of speed curve of inner ring nodes](image)
Figure 8. Speed and acceleration curves of inner ring nodes

Figure 8 shows the velocity-time curve of node 44492 at the edge of the largest radius of the end face of the inner ring. From the figure we can see that in the initial short period of time, the inner ring speed increases from 0 to the rated speed, and then presents a jagged jitter, but the overall change in speed value is relatively stable. At the same time, the acceleration curve changes drastically.

Figure 9. Speed and acceleration curves of roller nodes

On the roller, we selected three nodes to analyze the kinematics of the roller. Node A represents the roller center, node B represents the contact point between the roller and the outer ring, and node C
represents the contact point between the roller and the inner ring. Figure 9(a) shows the velocity-time curve for three nodes. It can be seen from the figure that the rotation speed of the roller center and the inner ring are consistent, starting from 0 and gradually becoming stable. The change of the contact point between the roller and the inner and outer rings indicates that the roller has just started to revolve, and then gradually becomes rotation. The overall speed after rotation is a sinusoidal change trend. The peaks of the curves in the graph indicate that the velocity is greatest when the node is in contact with the inner ring. The valley of the curve indicates that the velocity is minimal and close to zero when the node is in contact with the outer ring, which indicates that the roller is similar to pure scrolling at the time [10, 11]. Figure 9(b) shows the acceleration curves of the three nodes of the roller. It can be seen that there is no obvious change in the acceleration curve, but only a kind of jitter occurs. This is due to a collision between the cage and the roller.

5. Conclusion
Through the whole paper, our summary is as follows:
(1) The finite element model of high-speed railway bearing under the speed of 300km is established. The dynamic analysis of the radial and transverse combined loads is realized by the ANSYS/LS-DYNA platform.
(2) From the contact stress distribution of the inner ring of the bearing, it can be seen that under the condition of straight line running at a constant speed, the force of the two rows of rollers is basically the same. However, the bearing will be eccentric under the condition of driving at the minimum turning radius. In the extreme case, only one row of rollers is loaded. From the perspective of probability, the two rows of rollers have the same possibility of eccentric load, the overall stress level of the bearing is not high, and the vibration is small. At the same time, it can be seen from the contact stress distribution of the outer ring of the bearing that the closer the contact area between the roller and the outer ring is, the greater the contact stress on the outer ring is, which is consistent with the pitting corrosion of the bearing outer ring.
(3) Under normal working conditions, the movement of the high-speed bearing is similar to pure rolling, and the collision between the cage and the roller causes the roller acceleration curve to be a kind of jitter without change law.
(4) The finite element method provides a certain explanation for the fatigue damage occurred during the operation of high-speed rail bearings, and provides some references for the structural optimization of high-speed rail bearings.

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