Fatigue life prediction of the axle box bearings for high-speed trains

Predicción de la vida a fatiga de la caja de rodamientos de eje para trenes de alta velocidad

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RESUMEN

- La predicción exacta de la vida de los cojinetes es fundamental para asegurar el funcionamiento normal de los trenes de alta velocidad. Además, debido a sus altas velocidades de circulación, la carga sobre cojinetes en los trenes de alta velocidad es compleja y expuesta a condiciones muy duras; Por lo tanto, predecir la vida de los cojinetes basada en la carga fija usando un método tradicional es a menudo inexacta. Para resolver estos problemas, en este artículo se aplica la norma ISO 281:2007 de la Organización Internacional de Normalización y los métodos de cálculo de la vida útil de cojinetes de la teoría L-P que, combinados con la teoría del daño lineal de Miner, conforman un método de predicción de la vida útil de los cojinetes basado en la carga medida. A continuación, en base a técnicas de ensayo de carga en los muelles de la caja de ejes y su carcasa, se hacen mediciones sobre la caja de ejes de bogie de la línea del ferrocarril de la de alta velocidad CRH380AL, para calcular el historial temporal de carga del muelle y de la carcasa en una sección de línea de ferrocarril típica, así como la información de velocidad de marcha del tren. Finalmente, se calcula la vida prevista de la caja de ejes bajo la condición de marcha real utilizando este método. Los resultados se comparan con los obtenidos por el cálculo de vida usando la teoría clásica de Lundberg-Palmgren y el método de la norma ISO. Los resultados muestran que la vida de servicio calculada por el método de predicción de vida basado en la norma ISO es la más corta, en aproximadamente un octavo de la misma por el método estándar ISO; además, la vida calculada de servicio es significativamente más larga que la vida real de revisión de la caja actual del eje del tren de alta velocidad (2,4 millones de km), lo que indica que la vida calculada es razonable y segura. El método de cálculo de la predicción de la vida útil de los rodamientos se puede utilizar para guiar el diseño y la investigación teórica relacionada con cojinetes de caja de eje de tren de alta velocidad.
- **Palabras clave:** Tren de alta velocidad, caja de ejes, rodamiento cónico de doble fila, carga real medida, método de predicción de vida.

ABSTRACT

Accurate life prediction of bearings is critical in ensuring the normal run of high-speed trains. However, due to their high running speed, the bearing load of the high-speed trains is complex and exposed to hostile conditions; thus, predicting the life of the bearings by the fixed load using a traditional method is often inaccurate. To solve these problems, the International Organization for Standardization (ISO) 281:2007 and L–P theory bearing life calculation methods are applied in this study, combined with the Miner linear damage theory to propose a bearing life prediction method based on the measured load. Then, on the basis of axle box spring and tumbler load testing technology, the railway line of the motor bogie axle box of CRH380AL high-speed train is measured to obtain the load time history of the spring and tumbler at the typical railway line section, as well as the running speed information of the train. Finally, the predicted life of the axle box bearing under the actual running condition is calculated using this method. Results are compared with the life calculation results using the classic Lundberg–Palmgren theory and the ISO standard method. Results show that the service life calculated by the life prediction method based on ISO standard is the shortest, which is approximately one eighth of that by the ISO standard method; moreover, the calculated service life is significantly longer than the actual overhaul life of the current high-speed train axle box bearing (2.4 million km), indicating that the calculated life is reasonable and safe. The proposed life prediciton calculation method of the bearings can be used to guide the design and the related theoretical research of high-speed train axle box bearings.

Keywords: High-Speed Train, Axle Box, Double Row Conical Roller Bearing, Actual Measured Loadings, Life Prediction Method.

1. INTRODUCTION

As one of the most widely used and vulnerable parts of rotating machinery, the rolling bearing directly affects the reliability of the entire machinery operation [1]. The axle box bearing of high-speed trains is the key in ensuring the running quality and safety of the trains. Failure damage of the train bearing under high-speed running may cause huge economic losses and even lead to major disasters [2].

The fatigue life of bearing is a comprehensive reflection of its quality. Considering bearing fatigue life and reliability can prevent bearing failure accidents. Thus, bearing research have been focused on the accurate prediction of its service life. If the predicted service life is relatively long, there is a great potential of safety issues. Once the bearing fails prematurely, the entire train may breakdown or encounter accidents. On the other hand, if the predicted service life is remarkably short, the bearing will be replaced too early, which greatly increases the maintenance cost and causes significant waste of high-quality resources. The first theoretical basis for the formulation of a bearing life model was provided by the seminal work of Lundberg and Palmgren [3]. Lundberg–Palmgren (L–P) theory has been extensively used since the 1950s. Despite its wide acceptance, L–P theory contains several

Nomenclature			
List of symbols		k	number of columns of the rollers
i, j	running indices	a_1	modified coefficient of the reliability life
ϕ_j	azimuth angle of j^{th} roller	a_{ISO}	life correction coefficient
F_r, F_a	radial and axial loads	C	basic rated dynamic load in ISO method
Q_j	contact load of j^{th} roller	b_m	rated coefficient of the commonly used high-quality hardened bearing steel and good processing method
δ_r, δ_a	radial and axial displacements of bearing	f_c	coefficient related to the geometry, manufacturing precision, and materials of the bearing parts
α	nominal contact angle	P	equivalent dynamic load in ISO method
Z	number of rollers in the single-row bearing	e_c	contamination coefficient
C_i, C_e	rated dynamic loads of the inner and outer raceways in L-P method	C_u	fatigue load limit
P_i, P_e	equivalent loads of the inner and outer raceways in L-P method	κ	viscosity ratio
λ	stress concentration factor	C_0	basic rated static load
D_{we}	roller diameter	m	total load data points
D_{pw}	pitch circle diameter of the bearing	θ	bearing temperature
L_{we}	effective length of the roller	β	viscosity-temperature coefficient
K_t	load-displacement stiffness coefficient	D	fatigue damage
n	load-displacement exponent		

limitations. Thus, several researchers have modified and extended the L-P theory. These studies have gradually formed the International Organization for Standardization (ISO) method. The latest ISO 281:2007 [4] has been widely applied in engineering. However, the ISO standard method can only calculate the reference rated life of bearing under a fixed load. Hence, this method is not capable of estimating the bearing life in complex loading conditions, such as in high-speed train axle box; it can not reflect the actual bearing life in actual complex varying load conditions. Therefore, a new bearing life prediction method is necessary to accurately calculate the bearing life under complex varying load conditions.

2. STATE OF THE ART

Fatigue is the predominant mode of failure in rolling element bearings; thus, the life of bearings is governed by its Rolling Contact Fatigue (RCF) life. Given the special nature of RCF and its inability to relate directly to classic component fatigue, most of the early works in determining the service life of rolling bearings were based on empirical results and equations. For the past several decades, significant efforts have been devoted to improve the fatigue life calculation method for bearings [5–6].

Recently, several researchers developed equations for calculating bearing fatigue life based on different stress–life criteria [7–8]. Espejel et al. [9] developed a model for predicting rolling bearing life, which explicitly separates the survival of the raceway surface from the subsurface fatigue risk of the rolling contact. This model can account for the operating conditions of the bearing (e.g., load and lubrication conditions, as well as bearing geometry). In many applications, rolling bearings are subjected to highly varying loads and high-level vibrations. In such conditions, bearing failures occur significantly earlier than in conventional conditions [10]. An alternative method for estimating bearing fatigue life was developed. This method is applicable for general loading conditions because it accommodates a detailed information of contact pressure and load distribution in the rollers [11]. Therefore, the method can be used to estimate the bearing rated life using a quasi-static bearing model [12–13]. The original fatigue life equation was modified for the life estimation of tapered roller bearings with time-varying characteristics [14].

The load on high-speed trains is complicated and random. For unfixed conditions of the bearing load, such as in high-speed train bearing, Chen et al. [15] developed the 2D bearing life calculation method of metro vehicles, considering load and track conditions. The idea was to convert the complex varying load into an equivalent one by dividing the load and track conditions in calculating the service life; however, the divided load conditions failed to describe the entire load–time varying condition, leading to inaccurate life prediction. Ruitian et al. [16] studied the influence of partial load on the probability life of axle bearing of railway cargos. Juping et al. [17] obtained the design evaluation load of bearing through domestic and foreign standards of rail vehicles to calculate the reliability life; however, in these standards, the load is expressed in a simple equation that is related to the axle load rather than the load spectrum; hence, it cannot predict the bearing life accurately either. Thus, designing accurate models to estimate the service life of rolling-element bearings under varying loads and vibrations is essential.

This section provides a review on the recent development of rolling bearing fatigue life calculation. The remainder of this paper is organized as follows. Section 3 describes two typical bearing life calculation models of the L–P theory and ISO standard. Then, a new fatigue life prediction method based on the actual measured loadings is proposed. Section 4 presents a real engineering experiment of calculating the fatigue life of bearings of high-speed trains axle box using the three methods. Results and discussion of different life calculation methods are also presented. Section 5 concludes the paper.

3. METHODOLOGY

3.1 LIFE CALCULATION METHOD OF L–P THEORY

Lundberg and Palmgren proposed the life prediction model based on subsurface stress. However, this method needs to calculate the contact load between each roller and the raceway. The axle box bearing of high-speed trains is a double-row conical rolling bearing. Multiple variables are involved in determining the stress of a conical rolling bearing. Especially for a double-row conical rolling bearing, variables are related to one another. Therefore, the double-row conical rolling bearing is a complicated system. In engineering, the statics method is also practical for analysis [18].

L–P theory calculates the line contact situation according to load distribution. After obtaining the internal load distribution of the bearing, the equivalent loads of the inner and outer raceways are calculated as follows:

$$P_{\mathrm{i}} = \left(\frac{1}{Z}\sum_{j}^{Z} Q_j^4\right)^{1/4} \quad P_{\mathrm{e}} = \left(\frac{1}{Z}\sum_{j}^{Z} Q_j^{4.5}\right)^{2/9} \tag{1}$$

Then, the fatigue life of the bearing can be obtained as follows:

$$L_{10} = \left(L_{\mathrm{i}}^{-9/8} + L_{\mathrm{e}}^{-9/8}\right)^{-8/9} \tag{2}$$

$$L_{\mathrm{i}} = \left(\frac{C_{\mathrm{i}}}{P_{\mathrm{i}}}\right)^4 \quad L_{\mathrm{e}} = \left(\frac{C_e}{P_{\mathrm{e}}}\right)^4 \tag{3}$$

where L_i and L_e ($\times 10^6$) are the service lives of the inner and outer raceways, respectively; L_{10} ($\times 10^6$) is the service life under a reliability of 90%; and C_i and C_e (N) are the rated dynamic loads of the inner and outer raceways, respectively. For centripetal rolling bearing, the basic rated dynamic load between the raceways and the rollers are expressed as

$$C_{\mathrm{i,e}} = 552k\lambda\frac{(1\mp\gamma)^{29/27}}{(1\pm\gamma)^{1/4}}\left(\frac{D_{\mathrm{we}}}{D_{\mathrm{pw}}}\right)^{2/9} D_{\mathrm{we}}^{27/29} L_{\mathrm{we}}^{7/9} Z^{-1/4} \tag{4}$$

where λ is the stress concentration factor, which is 0.7 according to the empirical data; γ is the structural parameter, $\gamma = D_{we}\cos\alpha/D_{pw}$; D_{we} (mm) is the roller diameter used for the rated load calculation; D_{pw} (mm) is the pitch circle diameter of the bearing; k is the number of columns of the rollers; L_{we} (mm) is the effective length of the roller used for the rated load calculation. The symbols above and below "$\pm$" are suitable for the inner and outer raceways, respectively.

3.2 LIFE CALCULATION METHOD OF ISO STANDARD

L–P theory can well predict the bearing life based on the subsurface origin; however, it needs to first calculate the load distribution inside the bearing, which is a complex process that involves a large amount of calculation. Thus, ISO has presented a simplified algorithm based on the L–P theory. The internal load distribution of the bearing adopts the standard one. The latest ISO standard adopts the bearing life theory of Svenska Kullager Fabriken (SKF) Company. The modified life equation of its bearing is

$$L_{\mathrm{p}} = a_1 a_{\mathrm{ISO}} \left(\frac{C}{P}\right)^{10/3} \tag{5}$$

where a_1 is the modified coefficient of the reliability life; a_{ISO} is the life correction coefficient based on the system method; C (N) is the basic rated dynamic load; and P (N) is the equivalent dynamic load.

3.2.1 Basic Rated Dynamic Load

The radial basic rated dynamic load of the centripetal rolling bearing is

$$C = b_{\mathrm{m}} f_{\mathrm{c}} \left(kL_{\mathrm{we}}\cos\alpha\right)^{7/9} Z^{3/4} D_{\mathrm{we}}^{29/27} \tag{6}$$

where b_m is the rated coefficient of the commonly used high-quality hardened bearing steel and good processing method; f_c is the coefficient related to the geometry, manufacturing precision, and materials of the bearing parts; and α (°) is the nominal contact angle.

3.2.2 Equivalent Dynamic Load

In ISO standard method, the life calculation adopts the ideal load distribution. Thus, the equivalent dynamic load of the centripetal rolling bearing with $\alpha \neq 0°$ under the constant radial and axial loads is

$$P = XF_r + YF_a \tag{7}$$

The values of X and Y are shown in Table I (see section: supplementary material). e is applicable to the limit value of Fa/Fr with different X and Y.

3.2.3 Life Correction Coefficient

a_{ISO} is the correction coefficient based on the system method of life calculation, considering the complicated relationship among the fatigue limit of the bearing steel, lubrication, contamination, and equivalent dynamic load of the bearing. It is the complex function of contamination coefficient e_c, fatigue load limit C_u, equivalent dynamic load P, and viscosity ratio κ.

C_u is defined as the load when the contact point with the largest load of the raceway only reaches the fatigue stress limit. It is generally obtained from the chamber of bearing manufacturing. The ISO standard gives a simplified calculation method to simply estimate the fatigue load limit C_u of the rolling bearing with the basic rated static load C_0.

Life reduction caused by the contaminants in the lubricant membrane can be considered through the contamination coefficient e_c. Its value can be obtained from the table of the standard according to the actual contaminated situation.

Lubricant effectiveness mainly depends on the separation degree of the rolling contact surface. This type of separation condition can be expressed by the viscosity ratio κ. The standard provides two calculation methods of κ. The viscosity ratio is adopted for calculation based on the actual situation of the high-speed train axle box bearing. In other words, the lubrication state on the working surface of the bearing can be expressed by the viscosity ratio (i.e., the ratio of the actual motion viscosity ν to the reference motion viscosity ν_1).

$$\kappa = \frac{\nu}{\nu_1} \tag{8}$$

3.3 NEW LIFE CALCULATION METHOD BASED ON MEASURED LOAD

The time history and running speed information of the axle box spring and tumbler of the high-speed train when actually running is obtained. The spring load is approximated to the radial load of the bearing. The axial load is approximately obtained by the tumbler load [19]. Then, the equivalent load P_i can be calculated. The running speed information of the train is converted into the bearing rotating speed. The bearing rotating speed in the sampling interval is assumed to be constant to obtain the number of rotating turns of the bearing in the sampling interval l_{Pi}. The corresponding bearing life rings L_{Pi} can be calculated using ISO standard or L-P theory. According to Palmgren–Miner linear damage cumulative law [20], the damage degree intensifies with the increase of rotating turns of the inner and outer rings when the rolling bearing is under load. When the damage value $D = 1$, fatigue damage of bearing occurs. The life when the damage occurs under the running condition is the predicted bearing life based on the measured load.

$$L = \frac{\sum_{i=1}^{m} l_{Pi}}{D} = \frac{\sum_{i=1}^{m} l_{Pi}}{\sum_{i=1}^{m} \frac{l_{Pi}}{L_{Pi}}} \tag{9}$$

The parameters required in the equation are discrete data. i is the load points series, and m is the total load points. The corresponding bearing turns l_{Pi} can be calculated as long as high-precision sampling frequency is used to obtain the real-time changing discrete load P_i and real-time rotating speed P_i. The life turns of the bearing L_{Pi} can be calculated using discrete load P_i through the ISO standard or L–P theory method. In summary, analysis process of life calculation method as shown in Fig. 1.

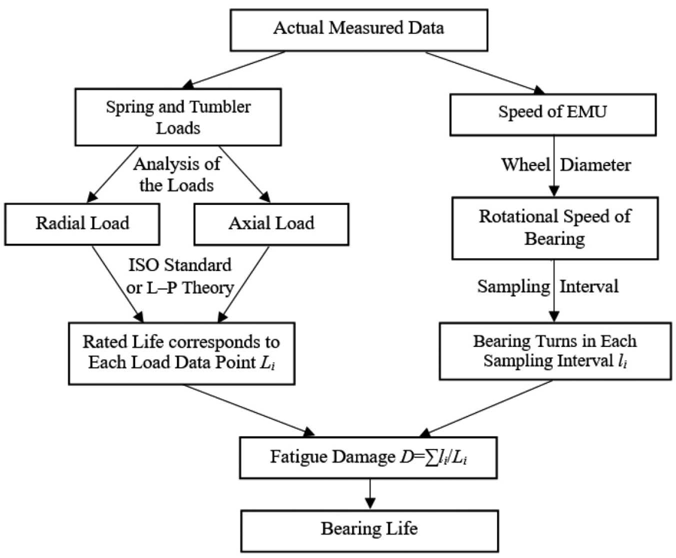

Fig. 1. Analysis Process of Life Calculation Method

3.4 EXPERIMENTAL MEASUREMENT FOR LIFE CALCULATION

For texting the axle box spring and tumbler load directly and accurately, the axle box spring and tumbler of the CRH380AL high-speed train were made into sensors according to the professional force sensor-making in research institute [21]. They were installed on the high-speed train bogie to complete the preparation for the railway line test. After installing the calibrated spring and tumbler force sensors, the installation and wiring were performed on site, as shown in Figs. 2 and 3 (see section: supplementary material). The spring and tumbler measuring positions on this type of high-speed train are shown in Fig. 4.

Fig. 4. Measuring Points of Spring and Tumbler Loads

In the entire testing process, various jamming signals were entered into the data collection system of the test link to interfere with the signal. Then, specific data processing software was used to improve the signal-to-noise ratio. The electrical signal is con-

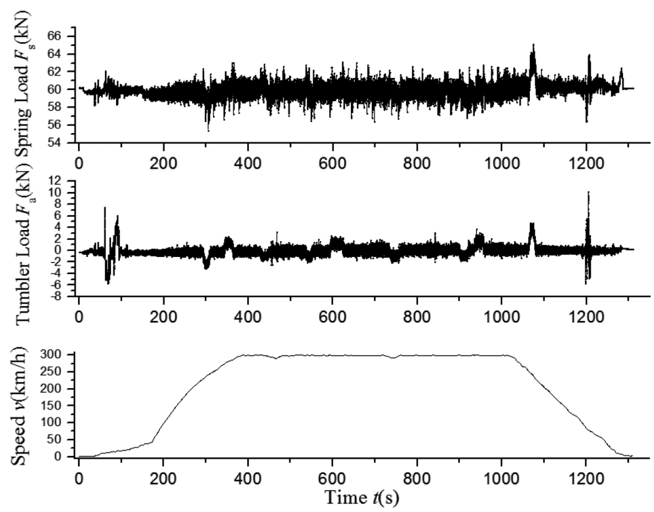

Fig. 5. Variations of Spring Load, Tumbler Load, and Train Running Speed over Time

Parameter	Value
Nominal contact angle α/(°)	9
Rated coefficient of contemporary common high-quality hardened bearing steel and good processing method b_m	1.1
Coefficients related to the geometry, manufacturing precision, and materials of bearing parts f_c	84.3
Column number of the rollers k	2
Effective length of the roller L_{we}/mm	52.8
Roller number of a single bearing Z	19
Diameter of the roller D_{we}/mm	19
Pitch diameter of the roller group D_{pw}/mm	185
Contamination coefficient e_c	0.8
Grease movement viscosity ν/(mm^2/s)	39.6(40 °C) 7.21(100 °C)

Table II. Values of Related Parameter Symbols verted into the load value to obtain the accurate test results. The experiment tested the track of the high-speed train from Beijing to Taiyuan (train number G615). This experiment intercepted a section of a typical line from Baoding to Dingzhou, stretched for 62 km (including the acceleration speed, uniform speed, and slowdown conditions), and obtained the time history of the spring load F_s and the tumbler load F_a of an axle box bearing on the motor bogie. Simultaneously, the Global Positioning System(GPS) was set on the train to collect the running speed signal v. In Fig. 5, the spring and the tumbler loads had great fluctuations at the starting and stopping stages. When the axle box of the high-speed train ran normally, the radial load fluctuated at approximately 60 kN with the maximum amplitude of 6 kN; and the axial load fluctuated at nearly 0 kN with the maximum amplitude of 12 kN. The parameter symbols and values for the test are presented in Table II.

4. BEARING LIFE CALCULATION AND RESULT ANALYSIS

4.1 THEORETICAL CALCULATION OF THE TRADITIONAL LIFE

The basic calculation parameters of the bearing can be obtained through the equations in Section 3.2, the calculation value as presented in Table III.

Parameter	Calculation value (kN)
Basic rated dynamic load C	740.52
Basic rated static load C_0	1488.65
Fatigue load limit C_u	150.95
Rated dynamic load of inner raceway C_i	94.93
Rated dynamic load of outer raceway C_e	124.29

Table III. Calculation Values of Relevant Parameters

The static analysis method is used to calculate the bearing load distribution. Column 1 roller and Column 2 roller are in the axial load direction of the double-row conical rolling bearing. The rollers in the columns are numbered to illustrate the expression of the load distribution. Bearing loads and roller numbers are shown in Fig. 6.

According to the measured data of the axle box spring and tumbler in Section 3.4, the maximum radial load of the axle box bearing is 66 kN, the maximum axial load is 12 kN, the average radial load is 60 kN, and the average axial load is 0 kN. The internal load distribution of the double-row conical rolling bearing is solved under the action on the maximum and average radial and axial loads. The calculation results are shown in Fig. 7.

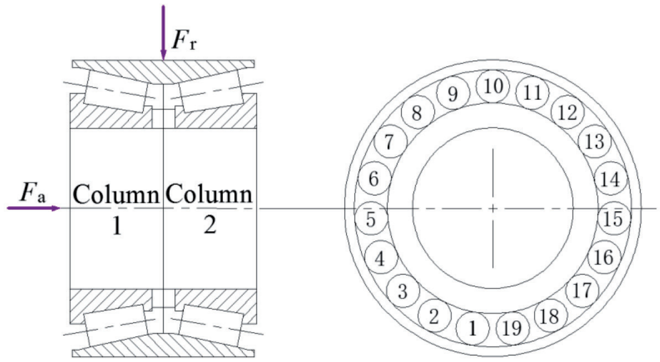

Fig.6. Schematic of the Double Row Conical Roller Bearing Loads and Roller Number

The internal load distribution of the bearing considers the numerical calculation results of the average radial and axial loads. The equivalent dynamic loads of the inner and outer raceways are calculated using Equation (1) to obtain P_i = 4.25 kN and P_e = 4.40 kN. Thus, the life turns of the bearing under 90% the reliability are

$$L_{10}=\left(L_i^{-9/8}+L_e^{-9/8}\right)^{-8/9}=191532.89\times10^6 \tag{10}$$

Based on the wheel diameter of 860 mm, the life turns of the bearing can be converted into the life mileage, as shown as follows:

km $$L_{10}'=860\times10^{-6}\times\pi L_{10}=51746.99\times10^4\ \text{km} \tag{11}$$

According to the ISO standard method, among the mileage data, the average equivalent dynamic load of 65.8 kN can be calculated based on the equivalent dynamic load data in Fig. 7. The working temperature is 70 °C under the normal running speed of 300 km/h. Then, the life turns of the bearing under the reliability of 90% is

$$L_{10p}=a_1a_{ISO}\left(\frac{C}{P}\right)^{10/3}=159714.12\times10^6 \tag{12}$$

The life turns of the bearing is likewise converted into the life mileage of 43151.04 × 10⁴ km.

4.2 BEARING LIFE CALCULATION USING THE PREDICTION METHOD BASED ON MEASURED LOADS

In the ISO standard method, service life is calculated using the ideal load distribution. Then, the equivalent dynamic load can be calculated, as illustrated in Fig. 8 (see section: supplementary material).

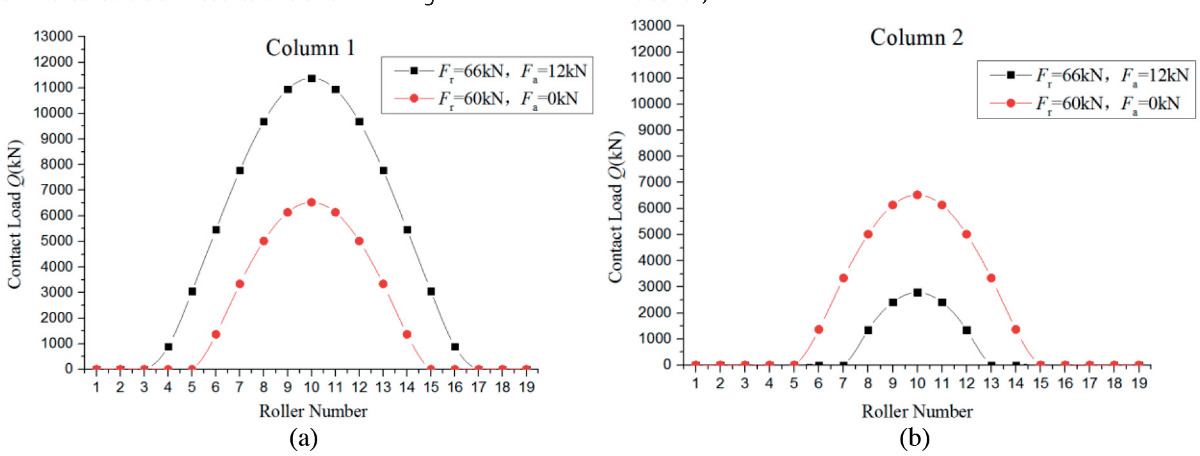

Fig. 7. Load Distribution of the Double Row Conical Roller Bearing. (a) Load Distribution of Column 1 Rollers. (b) Load Distribution of Column 2 Rollers

According to the actual contamination state of the bearing, the contamination coefficient uses $e_c = 0.8$. Then, the life correction coefficient a_{ISO} can be determined by the lubrication condition (expressed by the viscosity ratio κ) on the working surface of the bearing and the equivalent dynamic load *P*. The corresponding relation of the life correction coefficient a_{ISO} is shown in Fig. 9 (see section: supplementary material). The limit range of the life correction coefficient a_{ISO} is 0.1 to 50, and the maximum is 500 times greater than the minimum, indicating its great influence on the rated life of the bearing.

The bearing rotating speed under the running condition can be calculated by the running speed and the wheel diameter; however, no relevant data were obtained on the actual bearing temperature. Nevertheless, the data were selected between the two stations of the train in the high-speed running with short residence time. The working temperature in the data section can be assumed to be unchanged, and the working temperature is 70 °C. The power law empirical relation, which can accurately describe the viscosity–temperature characteristics of the bearing grease [22], is adopted to calculate the viscosity at 70 °C. The relation is expressed as follows:

$$\nu = \mu_{50} / (50/\theta)^{\beta} \tag{13}$$

where ν (mm²/s) is the motion viscosity of the grease; θ (°C) is the bearing temperature; μ_{50} is the viscosity at 50 °C; and β is the viscosity–temperature coefficient.

The viscosity data of 40 °C and 100 °C were substituted into Equation (13). Then, μ_{50} = 26.1542 mm²/s and β = 1.859 were obtained, and the movement viscosity of 13.99 mm²/s under 70 °C/s was calculated. The viscosity ratio was obtained using Equations (8). The viscosity ratio and the bearing rotating speed are illustrated in Fig. 10 (see section: supplementary material).

During the running of the train, the equivalent dynamic load and the viscosity ratio change in real time; thus, the life correction coefficient a_{ISO} also varies over time. The life correction coefficient a_{ISO} can be solved based on the viscosity ratio κ and corresponding equivalent motion load *P*.

The variations of the life correction coefficient a_{ISO} over the running time is shown in Fig. 11 (see section: supplementary material). The value of the life correction coefficient a_{ISO} ranges from 0.1 to 50, indicating a large difference. The value of a_{ISO} is minimal in the low-speed stage compared with the running speed. The life correction coefficient a_{ISO} also becomes gradually large with the increase of the running speed. The value of a_{ISO} in the typical running mileage is usually large, indicating that the running condition of the bearing is generally good.

The ISO standard provides the reliability life correction coefficient. The reliability considers 90%, i.e., $a_1 = 1$. Thus, the correction rated life of the bearing is calculated, as depicted in Fig. 12 (see section: supplementary material).

Based on the L–P theory method, the equivalent dynamic load of inner raceway and outer raceway can be calculated. Then, the rated life of the bearing can also be calculated, as illustrated in Figs. 13 and 14 (see section: supplementary material).

According to the actual wheel diameter of 860 mm, the train speed can be converted to the rotating speed of the axle box bearing. The sampling frequency of the speed signal is 50 Hz; thus, the time interval between two sampling points is Δt = 0.02 s. Given that the rotating speed and the equivalent load are constant within 0.02 s, the rotating turns of the bearing in each data point interval Δt can be converted as follows:

$$l_P = \frac{5}{18} \times \frac{\Delta t \times v}{0.86\pi} = \frac{5}{18} \times \frac{0.02v}{0.86\pi} \tag{14}$$

A total of 65528 data points was collected in the entire mileage, which divided the running time of the train into 65528 segments. The rotating turns of the bearing and the corresponding correction rated life in each segment were solved, as shown in Fig. 15 (see section: supplementary material). Then, the damage value of the axle box bearing of the train in this section can be solved.

$$D_{ISO} = \sum_{i=1}^{65528} \frac{l_{Pi}}{L_{P1i}} = 5.83 \times 10^{-7} \tag{15}$$

$$D_{L\text{-}P} = \sum_{i=1}^{65528} \frac{l_{Pi}}{L_{P2i}} = 1.49 \times 10^{-7} \tag{16}$$

Thus, the predicted life of the bearing under the reliability of 90% in the line is

km $$L_{ISO} = \frac{\sum_{i=1}^{65528} l_{Pi}}{D_{ISO}} \times \pi \times 860 \times 10^{-6} = 5216.71 \times 10^4 \text{ km} \tag{17}$$

km $$L_{L\text{-}P} = \frac{\sum_{i=1}^{65528} l_{Pi}}{D_{L\text{-}P}} \times \pi \times 860 \times 10^{-6} = 20410.97 \times 10^4 \text{ km} \tag{18}$$

4.3 COMPARATIVE ANALYSIS OF THE CALCULATION RESULTS

Table IV lists the bearing life under the reliability of 90% using the four methods.

Method	Life L_{10} (10000 km)
L–P theory	51746.99
ISO standard	43151.04
Proposed method based on L–P theory	20410.97
Proposed method based on ISO standard	5216.71

Table IV. Life Comparison Using the Different Life Calculation Methods

After comparing the life calculation results of L–P theory and the ISO standard method, the latter disregards the accurate load distribution but only provides an approximately standard load distribution, which may underestimate the bearing fatigue life. The predicted bearing life using the new method based on ISO standard is the shortest, because it considers the lubrication state of the train bearing under the low and other speeds while those traditional methods only considers the normal working speed. Under the low-speed condition, the reference motion viscosity is large, leading to a minimal viscosity ratio. However, the viscosity ratio reflects the important parameters of the lubrication state of the bearing, which results in the reduction of the bearing life. On the contrary, the new method based on ISO standard and in combination with the damage fully considers the complex load variations of the bearing in the actual operation. The predicted life result is relatively conservative and safe.

The data adopted in the test experiment were collected in the high-speed improved section where the train runs. Many extreme conditions, limited by the short mileage of the line section, are disgarded; thus, it cannot comprehensively reflect the entire service life. The measured axle box bearing is installed in the first-class compartment rather than the place with the most capacity and worst load conditions; hence, the predicted life is long. The requirements on the high-speed train bearings are considerably restricted. At present, bearing maintenance is only conducted at Level 3 maintenance (1.2 million km) and above. Bearing replacement is performed at Level 5 maintenance (2.4 million km). The calculation results show that the high-speed train bearings have adopted a large safety allowance in response to extreme conditions.

5. CONCLUSION

To predict the bearing life of high-speed trains accurately under a time-dependent varying load, a new life calculation method is proposed using the axle box spring and tumbler sensor based on the real vehicle line test data and combined with Miner's linear damage theory. The following conclusions can be drawn from this study:

(1) From the load distribution of the double-row conical rolling bearing on the axle box under different radial and axial loads, the radial load can increase the contact load of the double-row rollers and the axle load can cause the double-row rollers of the bearing under a partial load.

(2) From the test data, when the axle box of the high-speed train runs normally, its radial and axial loads widely fluctuate. The radial load fluctuates at approximately 60 kN with the maximum amplitude of 6 kN. The maximum fluctuation amplitude of the axial load is 12 kN. Therefore, the service life calculated with the fixed load using the traditional ISO and L–P theory methods is inaccurate.

(3) From the results of the different life calculation methods, the traditional methods of ISO standard and L–P theory disregard the influence of the dynamic varying load of the bearing on the service life and underestimate its actual life. The new method proposed in this paper considers the real-time load and speed variations of the train in the actual running condition. The predicted life of the proposed method based on ISO standard is the shortest, which is approximately one-eighth of that using the ISO standard method; thus, the predicted life using the proposed method is relatively conservative and safe.

The conclusions in this study provide a guideline for designing the axle box bearings of high-speed trains and conducting relevant theoretical studies. However, the measurement of the actual load of the bearing is required when using the method proposed in this study . The axle box bearing load in this study is measured by converting the spring and tumbler into the sensor; but for the motor gearbox bearing and others with complex bearing structure, directly measuring the actual load is difficult, which will affect its application to a certain extent. Load identification methods may be used to obtain the complex bearing loads, and this issue will be considered in future research works.

BIBLIOGRAPHY

[1] Shouqiang K, Danyang M, Yujing W, et al. "Method of assessing the state of a rolling bearing based on the relative compensation distance of multiple-domain features and locally linear". Mechanical Systems and Signal Processing. 2017. Vol. 86. p. 40-57. DOI: http://dx.doi.org/10.1016/j.ymssp.2016.10.006

[2] Amini A, Entezami M, Huang Z, et al. "Wayside detection of faults in railway axle bearings using time spectral kurtosis analysis on high-frequency acoustic emission signals". Advances in Mechanical Engineering. 2016. Vol. 8-11. p. 1-9. DOI: http://dx.doi.org/10.1177/1687814016676000

[3] Lundberg G, Palmgren A. "Dynamic Capacity of Rolling Bearings". Acta Polytechnica, Mechanical Engineering Series, Royal Swedish Academy of Engineering Sciences. 1947. Vol. 1. No. 3, p 1-52.

[4] ISO 281:2007 "Rolling Bearings – Dynamic Load Ratings and Rated life," Second Edition, February 15, 2007.

[5] Arakere N K. "Gigacycle Rolling Contact Fatigue of Bearing Steels: A Review". International Journal of Fatigue. 2016. Vol. 93. p. 238-249. DOI: http://dx.doi.org/10.1016/j.ijfatigue.2016.06.034

[6] Hong S W, Tong V C. "Rolling-Element Bearing Modeling: A Review". International Journal of Precision Engineering and Manufacturing. 2016. Vol. 17-12. p. 1729-1749. DOI: http://dx.doi.org/10.1007/s12541-016-0200-z

[7] Tong V C, Hong S W. "Study on the Stiffness and Fatigue Life of Tapered Roller Bearings with Roller Diameter Error". Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology. In Press. 2016. DOI: http://dx.doi.org/10.1177/1350650116649889

[8] Nagatani H. "Improved Method of Roller Bearing Fatigue Life Prediction Under Edge Loading Conditions". Tribology Transactions. 2010. Vol. 53-5. p. 695-702. DOI: http://dx.doi.org/10.1080/10402001003699593

[9] Morales Espejel G E, Gabelli A. "A Model for Rolling Bearing Life with Surface and Subsurface Survival: Sporadic Surface Damage from Deterministic Indentations". Tribiology International. April 2016. Vol. 96. p. 279-288. DOI: http://dx.doi.org/10.1016/j.triboint.2015.12.036

[10] Jacobs W, Van Hooreweder B, Boonen R, et al. "The Influence of External Dynamic Loads on the Lifetime of Rolling Element Bearings: Experimental Analysis of the Lubricant Film and Surface Wear". Mechanical Systems and Signal Processing. January 2016. Vol. 74. p. 144-164. DOI: http://dx.doi.org/10.1016/j.ymssp.2015.04.033

[11] Tong V C, Hong S W. "Fatigue Life of Tapered Roller Bearing Subject to Angular Misalignment". Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science. February 2016. Vols. 230-2. p. 147-158. DOI: http://dx.doi.org/10.1177/0954406215578706

[12] Kabus S, Hansen M R, Mouritsen O. "A New Quasi-Static Cylindrical Roller Bearing Model to Accurately Consider Non-Hertzian Contact Pressure in Time Domain Simulations". Journal of Tribology. October 2012. Vol. 134-4. Paper No. 041401(1-10). DOI: http://dx.doi.org/10.1115/1.4007219

[13] Kabus S, Hansen M R, Mouritsen O. "A New Quasi-Static Multi-Degree of Freedom Tapered Roller Bearing Model to Accurately Consider Non-Hertzian Contact Pressures in Time-Domain Simulations". Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics. January. 2014. Vol. 228-2. p. 111-125. DOI: http://dx.doi.org/10.1177/1464419313513446

[14] Houpert L, Chevalier F. "Rolling Bearing Stress based Life -Part I: Calculation Model". Journal of Tribology. April 2012. Vol. 134-2. Paper No. 021103. DOI: http://dx.doi.org/10.1115/1.4006135

[15] Chen Y, Hui W, Gang S, et al. "Two-dimensional calculation method for the life of metro axle box bearing". Journal of East China Jiaotong University. 2014. Vol. 5. p. 1-5. DOI: http://dx.doi.org/10.16749/j.cnki.jecjtu.2014.05.004

[16] Ruitian Z, Yongxiang Z, Hui C. "Skewed loading effect on the probabilistic lives of journal bearing of railway wagon". Journal of Mechanical Engineering. 2015. Vol. 51-24. p. 177-183. DOI: http://dx.doi.org/10.3901/jme.2015.24.177

[17] Juping Y, Yongxiang Z, Hui C, et al. "Reasonable load sets for design assessment on journal/axle box bearings of railway vehicles related to probabilistic lives". Journal of Mechanical Engineering, 2015. Vol. 51-18. p. 191-197. DOI: http://dx.doi.org/10.3901/jme.2015.18.191

[18] Jiasen Y, Longquan Z, Sanxing Z, et al. "Analysis on load distribution of high-speed railway axle box bearings". Bearing. 2012. Vol. 10. p. 12-16. DOI: http://dx.doi.org/10.3969/j.issn.1000-3762.2012.10.003

[19] Dekun L, Qiang L, Xi W, et al. "Life Prediction Method for EMU Axle Box Bearings Based on Actual Measured Loadings". Chinese Journal of Mechanical Engineering. November 2016. Vol.52-22. p.45-54. DOI: http://dx.doi.org/10.3901/JME.2016.22.045

[20] Miner M A. "Cumulative damage in fatigue". Journal of Applied Mechanics. 1945. Vol. 12. p. A159–164.

[21] Zunsong R, Shouguang S, Qiang L, et al. "Axle spring load test and dynamic characteristics analysis of high speed EMU". Chinese Journal of Mechanical Engineering. May 2010. Vol. 46-10. p. 109-115. DOI: http://dx.doi.org/10.3901/JME.2010.10.109

[22] Yanjuan Z, Jianmei W, Min C, et al. "Research on viscosity-temperature characteristic of the lubricating oil of oil film bearing". Lubrication Engineering. 2014. Vol. 39-3. p. 75-78. DOI: http://dx.doi.org/10.3969/j.issn.0254-0150.2014.03.017

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SUPPLEMENTARY MATERIAL

http://www.revistadyna.com/documentos/pdfs/_adic/8414-1.pdf