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Research Article

The Cross-Scale Life Prediction for the High-Speed Train Gearbox Shell Based on the Three-Interval Method

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The gearbox shell is a key component of class A of high-speed trains. In engineering applications, the fatigue life prediction of the gearbox shell is a critical issue to be addressed. It is not feasible to obtain fatigue life results for the gearbox shell experimentally because of its long design life, lack of actual failure data, complex structure, high test cost, and material dispersion. Therefore, the cross-scale method was introduced to accurately predict the fatigue life of the gearbox shell. In this study, the entire gearbox shell is divided into two scales of "material structure." Firstly, the S-N curve is plotted within the material layer, based on the data from the rotating bending fatigue test. Secondly, the finite element model of the gearbox shell is established within the structural layer via the simulation platform. The characteristics and random vibration of the established model are analyzed and presented. Additionally, the first ten-order frequency of modal analysis and power spectral density responses of the gearbox are obtained. The fatigue life of the gearbox shell and the safe running distance of the train are calculated by using the three-interval method and the linear cumulative damage rule, respectively, by combining the fatigue analysis from the material layer with the simulation analysis from the structure layer. Finally, to illustrate the application of the proposed method, a group of small-scale test examples is provided. The proposed method can be used in fatigue life prediction more effectively than the single finite element simulation method.

1. Introduction

As a key component of class A of high-speed trains, the gearbox has an important influence on the safe operation of high-speed trains [1]. The gearbox is located at the bottom of the train and is suspended on the moving shaft. During the normal driving of the vehicle, it will be affected by various vibrations and shock loads between the wheels and rails, which may cause cracks in the gearbox body [2]. There are two main kinds of failure forms for the gearbox shell as follows: fatigue damage failure and tensile damage failure. Fatigue damage failure is caused by the reciprocating oscillation of the gearbox when the train is running, while tensile damage failure is due to its static load and external impact on the gearbox. Several factors, such as cracks, airtightness, and high-temperature rise, could cause the gearbox shell to burst [3]. The failure of the gearbox shell on Japan's Shinkansen trains caused two bursts, resulting in outage accidents and serious injuries. Fatigue damage, a damage form of structures or materials under the action of small reciprocating external stress, is one of the typical failure forms of such nonrepairable long-term service structures. The fatigue damage process is slow and difficult to detect. Therefore, the fatigue life prediction of the gearbox shell is critical, which in turn affects the operation safety of high-speed trains.

The gearbox shell of the high-speed train is a typical nonrepairable long-term service structure. Its characteristics include long design life, lack of actual failure data, a complex structure, high test cost, inability to conduct an independent test for a single shell, and material dispersion [3]. A routine test will take too long as the gearbox shell is bulky and has a design life of 20 years. Even if the accelerated test is done, the reliability test for the gearbox will be conducted in a few months. Therefore, the traditional test method is ineffective in obtaining fatigue test data. To overcome the above shortcomings, the CAD model of the gearbox shell is established and used for the fatigue analysis to obtain the failure data of the gearbox shell with the help of the finite element simulation technology [4], such as Guo et al. [4] developed a computation process to predict noise radiation of gearboxes. But the simulation analysis yielded fewer failure data. For the problem of insufficient failure data in complex systems, the existing and most widely used method is statistical inference through probability statistics [5, 6]. However, these methods that use the data from the bottom layer to directly analyze the system-level fatigue life do not consider the perspective of scale effect and cross-scale problems.

The two important factors that affect the performance of a gearbox shell in service are material and structure, of which material is the main factor affecting performance. The scale effect in material science refers to the characteristics of the research object at different scales [7, 8], and cross-scale refers to the difference in the degree of linearity in different directions of objects. In the field of materials, cross-scale research methods have been widely used, and they are gradually being applied in the study of complex systems. To predict the mechanical properties of CFRP, Qi et al. [9] proposed an approach based on cross-scale simulation. Firstly, the ABAQUS software is used to establish the structural representative volume element of unidirectional CFRP at the microscopic level. Secondly, based on the homogenization theory, the mesoscopic simplified RVE of multidirectional CFRP is established according to the dimension of structural RVE. Finally, a series of experiments were conducted. To solve the interactions between the concrete-faced rock fill and the foundation, Chen et al. [10] presented a global concurrent cross-scale nonlinear analysis approach. To examine the multiscale elastic anisotropy of shale, Saleh et al. [11] used a cross-scale, big data-based nanoindentation technique. This was accompanied by compositional analysis such as XRD and cementation agent identification, and microstructural analysis such as optical microscopy and SEM. Thus, material deformation and fracture models are established and integrated into mechanics and finite element models to form cross-scale models. This has paved a new direction for research in solving complex system problems.

The entire gearbox shell can be regarded as a complex system that is divided into the scale domains of "material structure" from a cross-scale perspective. The high-strength aluminum alloy A356 was used to perform the rotating bending fatigue test for the material layer. The Weibull distribution of two parameters, a related fatigue theory, is chosen to describe the fatigue life distribution of that material. The expected value of the stress amplitude is used as the number of cycles in the Weibull distribution. In this study, the S-N curve model of the material layer is established by fitting 11 sets of experimental data under different stress amplitudes. A CAD model of the gearbox shell was established for the structure layer by finite element analysis. The first ten-order frequency modal analysis of the gearbox shell and the equivalent stress figure of the gearbox shell from 1sigma to 3sigma is obtained through modal analysis and random vibration analysis. Combining the fatigue analysis from the material layer with the simulation analysis from the structure layer, the fatigue life of the gearbox shell is predicted using the three-interval method and linear cumulative damage rule. In this study, a set of small-scale sample tests is designed to verify the rationality of this method. The results prove that this method is very superior and can be applied in aerospace, shipbuilding, automobiles, and other industries.

2. S-N Curve Model of Gearbox Shell Material

The S-N curve is a curve that represents the relationship between the fatigue strength of a standard material sample under certain experimental conditions and the number of fatigue cycles (fatigue life) [12, 13]. To determine the S-N curve of the gearbox shell material of a high-speed train [14], standard samples must be cast according to the material composition of the gearbox shell and a rotating bending fatigue test must be conducted.

In the rotating bending fatigue test, the samples used are high-strength aluminum alloy A356, which is the same material as that of the gearbox shell of a high-speed train. Its chemical composition is as follows: Si content is 6.5–7.5%, Mg content is 0.20–0.40%, Fe content is less than 0.20%, Cu content is less than 0.20%, Mn content is less than 0.10%, Zn content is less than 0.10%, Ti content is less than 0.20%, each of the other elements is less than 0.05%, their sum being less than 0.15%, and aluminum is the remaining amount. The standard sample adopted is cylindrical. The total length of the samples is 140 mm, the diameter of the clamping section is 17 mm, and the diameter of the parallel section is 11.5 mm.

There are 26 sets of rotating bending fatigue tests, all of which use samples of the same size. After fitting the experimental data and comparing different distributions, the two-parameter Weibull distribution is found to be most suited to describe the experimental data. Therefore, in this study, the two-parameter Weibull distribution is used to describe the fatigue life distribution of materials [15]. The probabilistic graphical method is used to estimate the parameters of the Weibull distribution. Under the Weibull distribution, the expected value of the stress amplitude is used as the number of cycles of the stress amplitude. The cumulative distribution function of the two-parameter Weibull distribution can be expressed as equation (1), and the probability density function of the distribution can be expressed as equation (2):

$$F(t) = 1 - \exp\left[-\left(\frac{t}{\eta}\right)^{\beta}\right], \tag{1}$$

$$f(t) = \frac{\beta}{\eta}\left(\frac{t}{\eta}\right)^{\beta-1} \exp\left[-\left(\frac{t}{\eta}\right)^{\beta}\right], \tag{2}$$

where β is the shape parameter, and η is the scale parameter.

Logarithms are taken at the same time on both sides of the two-parameter Weibull distribution equation. Setting $y = \ln[\ln((1/1 - F(t)))]$, $x = \text{In}\, t$, $a = \beta$, and $b = -\beta\text{In}\eta$, equation (1) can be transformed into the following form:

$$\begin{gathered} Y = ax + b, \\ \frac{1}{1 - F(t)} = \exp\left(\frac{t}{\eta}\right)^{\beta}, \\ \ln\left[\ln\left(\frac{1}{1 - F(t)}\right)\right] = \beta \ln t - \beta \ln \eta. \end{gathered} \tag{3}$$

The Weibull distribution parameters of fatigue life differ when different stress amplitude conditions are applied. Therefore, this experiment needs to be considered at different stress amplitude levels separately. For example, when the stress amplitude is 220 MPa, there are five sets of experimental data according to the number of cycles N, which are calculated by the value of $F(t)$ in equation (4). Based on the relationship between the number of cycles and the time (54000 cycles per hour), the number of cycles should be converted into time t. Finally, by t and $F(t)$, the values of x and y are obtained. Table 1 lists the results of the Weibull distribution parameter estimation when the stress amplitude is 220 MPa:

$$F(t) = \frac{i - 0.3}{n + 0.4}\ (i = 1, 2, \ldots, n). \tag{4}$$

Linear fitting of the experimental data based on the least-squares method provides the intercept a and slope b of the linear model. When the stress amplitude condition is fixed to 220 MPa, the shape parameter β is 2.94958 and the scale parameter η is 0.85059. On substituting the scale parameters and shape parameters into equation (5), the expected value of the Weibull distribution is 0.759 h, which is converted to the number of cycles N, which is 40986. Similarly, the number of cycles under other stress amplitude conditions could be calculated and converted into logarithmic form to obtain 11 sets of processed experimental data. Table 2 lists the number of cycles under different stress amplitudes:

$$\sum = \eta \cdot \Gamma\left(1 + \frac{1}{\beta}\right). \tag{5}$$

Based on the 11 sets of experimental data in Table 2, the S-N curve of gearbox shell material is fitted by selecting an appropriate S-N curve model. The form of the S-N curve model selected in this study is $S^m N = C$ [16]. The logarithmic form of this model is $LgN = LgC + mLgS$. Let $Y = \text{lg}N$ and $X = \text{lg}S$, so the model can be expressed as $Y = a + bX$.

Table 2 lists the linear fitting of 11 sets of fatigue test data. Table 3 lists the result of curve fitting. The R-square value is 0.97375, which shows that the fitting result is excellent and conforms to the linear relationship. The logarithmic relationship between the number of fatigue cycles of the gearbox shell material and the stress amplitude is expressed as equation (6):

$$\text{lg}N = 17.76904 - 5.58984\text{lg}S. \tag{6}$$

From Figure 1, the power function formula of the S-N curve model is expressed as equation (7):

$$S^{5.58984} N = 10^{17.76904}. \tag{7}$$

3. Finite Element Simulation of Gearbox Shell Structure

3.1. Modal Analysis. Modal analysis is a special method in the study of dynamic characteristics of structures. This method is widely used in the dynamic design of mechanical structures and the fault diagnosis of equipment. In this study, a modal analysis of the gearbox shell structure of a high-speed train (Figure 2) is performed to understand the main modal characteristics of each order of the gearbox shell structure of a high-speed train and predict the actual vibration response generated by each internal or external vibration source within the vulnerable frequency band [17, 18]. To obtain the modal parameters [19, 20], the method of computational modal analysis is adopted due to the long test cycle, test difficulty, and high test cost of the gearbox for bench testing. Initially, the three-dimensional model of the high-speed train gearbox shell is established using the three-dimensional modeling software Pro/E to realize the parameterization of geometric modeling. Then, the simulation software ANSYS Workbench carries out the modal analysis. During the 3D modeling process, the model is simplified and modified reasonably to improve the running speed of the modal analysis.

The design documents of the high-speed train establish the three-dimensional solid model and import it into the finite element analysis software while the material properties, such as Poisson ratio, elastic modulus, density, yield strength, and tensile strength, are set [21]. Table 4 displays the specific values. The Solid186 element, with 8 nodes of hexahedron, was chosen to mesh the gearbox shell structure as the element characteristics make it suitable for structures with irregular shapes and curve model boundaries. The gearbox shell structure is divided into free grids. The grid density is manually set at the upper edge of the shell, and the rib plates on both sides of the shell, the bottom of the shell, and the area between the bearing seats [22, 23]. This ensures not only the accuracy of the modal analysis of the gearbox shell structure but also improves the speed of calculation. After meshing, 25979 grid nodes and 13982 cell bodies are generated.

There are two constraints set for the gearbox structure shell. One of them is to limit all degrees of freedom of the derrick washer at the connection between the gearbox and

TABLE 1: Weibull distribution parameter estimation when the stress amplitude is 220 MPa.

i	N	T	$F(t)$	x	y
1	24000	0.444444	0.129630	−0.81093	−1.97446
2	33000	0.611111	0.314815	−0.49248	−0.97269
3	39000	0.722222	0.500000	−0.32542	−0.36651
4	48000	0.888889	0.685185	−0.11778	0.144767
5	60000	1.111111	0.870370	0.10536	0.714455

TABLE 2: The number of cycles under different stress amplitudes.

	S (MPa)	N	lg$S$	lg$N$
1	250	18000	2.39794	4.255273
2	220	40986	2.342423	4.61066
3	200	123000	2.30103	5.089905
4	170	117180	2.230449	5.308031
5	130	705000	2.113943	5.848189
6	110	4070000	2.041393	6.609594
7	100	5890000	2.000000	6.770115
8	94	2995000	1.973128	6.476397
9	91	5832500	1.959041	6.765855
10	88	7095000	1.944483	6.850952
11	85	10100000	1.929419	7.004321

TABLE 3: The result of curve fitting.

	Value	Standard errors
A	17.76904	0.61402
B	−5.58984	0.28981
R-square	0.97375	

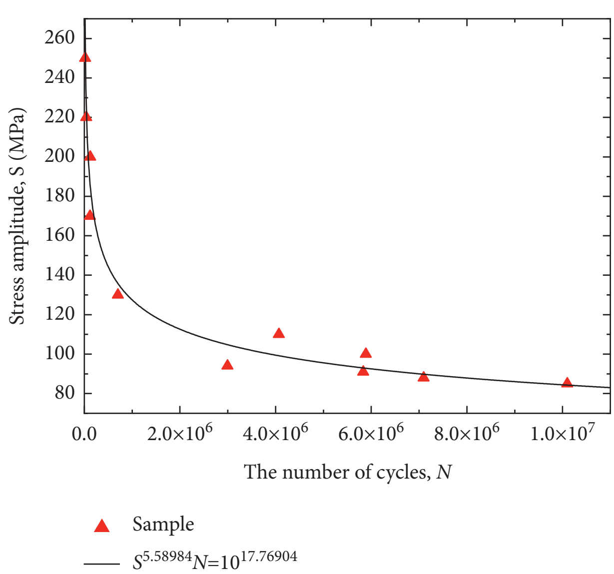

FIGURE 1: S-N model of high strength aluminum alloy A356.

the derrick [24]. As represented in Figure 3, the other constraint limits the large bearing seat of the gearbox to the other 5 degrees of freedom except for the degree of freedom of rotation around the axle. After applying constraints and meshing, the ANSYS Workbench software is used for modal analysis to obtain the first ten modal frequencies of the gearbox, as listed in Table 5. The minimum first-order frequency is 525 Hz. When the high-speed train runs at the

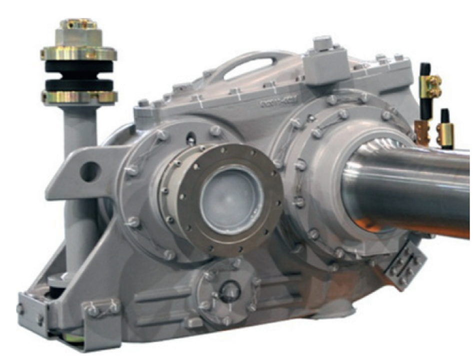

FIGURE 2: The gearbox of a high-speed train.

TABLE 4: The parameters of finite element analysis.

Properties	Parameters
Material	A356
Poisson's ratio	0.33
Elastic modulus (MPa)	0.7×10^5
Density (kg/m3)	2700
Yield strength (σ_s/MPa)	≥ 190
Tensile strength (σ_b/MPa)	≥ 230

speed of 300 km/h, the frequency of the input shaft at the rated speed of the gearbox is 69.033 Hz. The results of the above analysis show that the frequency of the input shaft of the gearbox is much less than the first-order natural frequency, and there will be no resonance with it.

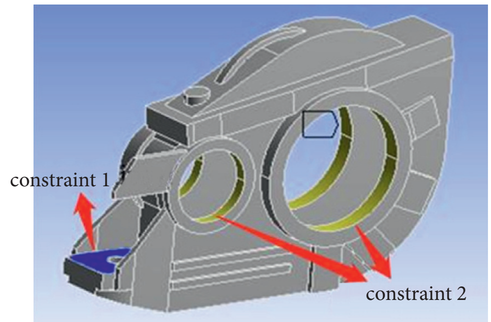

Figure 3: Constraint division of gearbox shell structure.

Table 5: The ten-order frequencies of the gearbox shell.

Frequency	First order	Second order	Third order	Fourth order	Fifth order	Sixth order	Seventh order	Eighth order	Nine order	Tenth order
F (Hz)	532	973	1132	1564	1608	1713	1819	1889	2015	2210

3.2. Random Vibration Analysis. High-speed trains produce vibrations during operation. Such vibrations are mainly caused by unflat track, and unflat track is the main excitation source during high-speed train operation. The vibrations produced by high-speed trains consist of various random vibrations. These vibrations have a significant impact on high-speed train safety. At present, there are two methods to study the reliability of mechanical structures with random vibration as follows: the time-domain analysis method and the frequency-domain analysis method [4]. The time-domain analysis method requires a large amount of measured data to count the cycles of the measured mechanical structure. However, structures such as high-speed train gearbox shells have not yet reached the critical point of fatigue damage. Therefore, it is difficult to obtain more comprehensive measured data. Therefore, in this study, the frequency-domain analysis method is adopted. The frequency-domain analysis method is based on the power spectral density and can be easily analyzed with the help of finite element analysis software.

As is known, the power spectrum data must be measured on high-speed railways. For random vibration analysis [25, 26], the power spectral density of the Qinghai-Tibet railway track is chosen as the input power spectral density to save manpower. Firstly, the results of the modal analysis, which includes the natural frequency and vibration modes, are imported into ANSYS Workbench finite element software. Then, the acceleration power spectral density is selected as the power spectral density. To complete the loading of power spectral density for random vibration analysis, the power spectral density diagram of the Qinghai-Tibet railway track is converted into data points and imported into the tabular data. Finally, as depicted in Figure 4, the output results are set to obtain the power spectral density response of the gearbox shell of the high-speed train, including the equivalent stress figure of the gearbox structure from 1sigma to 3 sigma. According to the results, the 1 sigma equivalent stress is 19.072 MPa, the 2 sigma equivalent stress is 38.144 MPa, and the 3 sigma equivalent stress is 57.217 MPa.

4. Life Prediction of Gearbox Shell Based on Three-Interval Method and Linear Cumulative Damage Rule

The fatigue life of the gearbox shell structure is calculated using the S-N curve model of the gearbox shell material of the high-speed train and the random vibration analysis results of the gearbox shell structure, combined with the three-interval method [27, 28] and the linear cumulative damage rule [29, 30]. As listed in Table 6, based on the S-N curve model of gearbox shell material (9) the number of permitted cycles of the gearbox shell structure under 3sigma equivalent stress is calculated.

As per the design requirements, the gearbox shells ensure that the high-speed train can safely run 2.4 million km at a speed of 300 km/h. Therefore, according to the formula $T = (240 \times 10^4/300)$, the calculated actual operation time of the gearbox on the high-speed train is 8000 h.

Based on the Miner linear cumulative damage rule, the equation for the structural damage of high-speed train gearbox shell [31]is expressed in equation (8):

$$D = \frac{n_{1\sigma}}{N_{1\sigma}} + \frac{n_{2\sigma}}{N_{2\sigma}} + \frac{n_{3\sigma}}{N_{3\sigma}}, \tag{8}$$

where D is the damage of the gearbox shell of the high-speed train.

From experience, it is seen that the Miner linear cumulative damage rule is too conservative for the random vibration analysis of large mechanical structures. As a result, this study adopts the suggestions of Yao and Yao [32] and assumes that the damage D is 1.5. When D equals 1.5, it indicates that the gearbox shell has suffered fatigue damage. When D is less than 1.5, it means that the gearbox shell is not damaged. N is the actual number of cycles of the gearbox structure at or below the stress level. $n_{1\sigma} = 0.681 v_0^+ T$, $n_{2\sigma} = 0.271 v_0^+ T$, and $n_{3\sigma} = 0.0433 v_0^+ T$.

For the random process conforming to the Gaussian distribution, the peak expectation rate is obtained from equation (9). The peak expectation rate is numerically equal to the number of cycles per unit time [33, 34]:

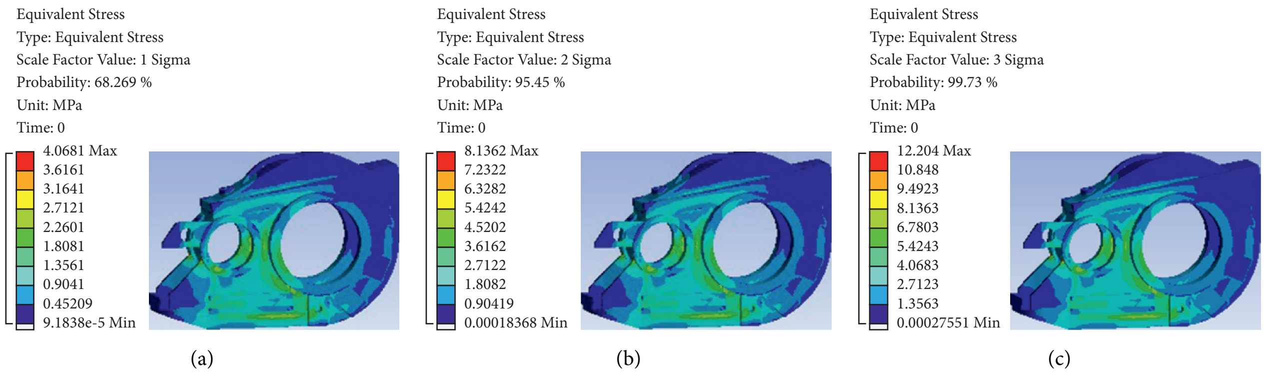

Figure 4: The result of random vibration. (a) 1sigma. (b) 2sigma. (c) 3sigma.

Table 6: The number of permitted cycles of gearbox shell structure under 3sigma equivalent stress.

	Equivalent stress S (MPa)	Number of permitted cycles N
1sigma	19.072	4.091×10^{10}
2sigma	38.144	8.494×10^{8}
3sigma	57.217	8.806×10^{7}

$$v_0^+ = v_p = \sqrt{\frac{\dot{\sigma}}{\dot{\sigma}}}, \quad (9)$$

where v_p is the peak expectation rate of the random process.

For random vibration, σ is the root mean square of stress, and $\dot{\sigma}$ is the root mean square of the first derivative of the stresses at each position of the gearbox shell.

For general engineering problems, the number of cycles per unit time is assumed to be 40:

$$T = \frac{1.5}{v_0^+ \left(\left(0.683/N_{1\sigma}\right) + \left(0.271/N_{2\sigma}\right) + \left(0.0433/N_{3\sigma}\right)\right).} \quad (10)$$

Finally, by transforming equation (8) into equation (10), the fatigue life of the gearbox shell is calculated as 12588 h, and the safe running distance of the high-speed train is 3.776 million km. The calculated values far exceed the 2.4 million km required by the design life of the high-speed train gearbox shell. Therefore, the fatigue life of the gearbox shell obtained by simulation analysis combined with the three-interval method and linear cumulative damage method fulfills the design requirements.

5. Verification of Fatigue Life Prediction of Aluminum Alloy Specimens with Prefabricated Cracks

The current test conditions cannot achieve the fatigue test of the entire gearbox shell due to the large volume and long test cycle of the high-speed train gearbox shell. Therefore, to verify that the fatigue life of the gearbox is 12588 h, this study designs a set of small-scale samples.

Before the test, a set of 8 samples were cast using the high-strength aluminum alloy A356 in the same casting environment as the gearbox shell. At the designated position of the sample, two holes were opened and a crack was prefabricated to ensure that the sample was installed on the universal fatigue testing machine for fatigue testing; the crack can grow in the designated direction. The loading ratio is set at 0.1, the loading force is 3800 N, and the loading frequency is 15 Hz (54000 cycles per hour). The fatigue data of each sample were recorded. Thus, the true number of fatigue cycles of the sample was calculated by analyzing the fatigue test data. After the test, the crack in the middle of the sample extended along the prefabricated crack direction until fatigue failure occurred. Figure 5 shows the sample cracking under prefabricated conditions.

Table 7 lists the fatigue life of eight samples. All eight samples were carried out under the same experimental conditions [35, 36]· To facilitate the verification of subsequent results, the average fatigue life of these eight samples was selected as the real value of the structural fatigue life test. Thus, the real value of the structural fatigue life test is 135958.

In the finite element simulation software ANSYS, a three-dimensional model of the sample structure is established, which is similar to the actual physical experiment. The length and width are both 50 mm, the thickness is 10 mm, and a prefabricated crack is set in the middle. The free mesh division method is adopted, which is the same as that in the gearbox shell of the high-speed train. The mesh density is increased by the manual setting in the crack growth direction, which is the most vulnerable place for fatigue damage. Under the same experimental conditions as the physical experiment, the two holes are subjected to sinusoidal forces F1 and F2 of the same size and opposite direction, with an amplitude of 3800 N and a frequency of 15 Hz. Figure 6 shows the loading by the sinusoidal forces, F1 and F2.

At the simulation level, the equivalent stress of the sample structure is 177.12 MPa after loading. Figure 7 shows the simulation results. Based on the S-N curve model of the gearbox shell material of a high-speed train obtained in Section 2, the number of permitted cycles of the structure of the gearbox shell under 3sigma equivalent stress is shown in Table 8. As the fatigue life of the sample calculated from equation (9) and equation (10) is 2.946 h, the number of fatigue cycles of the sample is 159068.

The comparison results show that the fatigue life of the sample is predicted using the fatigue tool of the finite element simulation software. The same loading conditions are set as in the physical experiment, and the results are shown in Figure 8. The minimum number of cycles of the fatigue life of the structure obtained by the fatigue tool is 177500, and

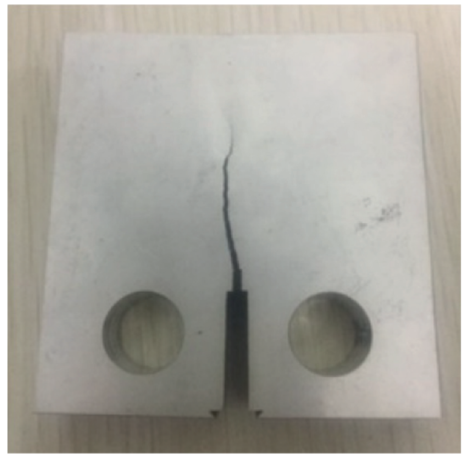

Figure 5: Sample cracking under prefabricated conditions.

Table 7: The fatigue life of 8 samples obtained by fatigue test.

Number	CT1	CT2	CT3	CT4	CT5	CT6	CT7	CT8	Mean
Fatigue life	103101	92245	108976	183269	116605	173060	107920	202489	135958

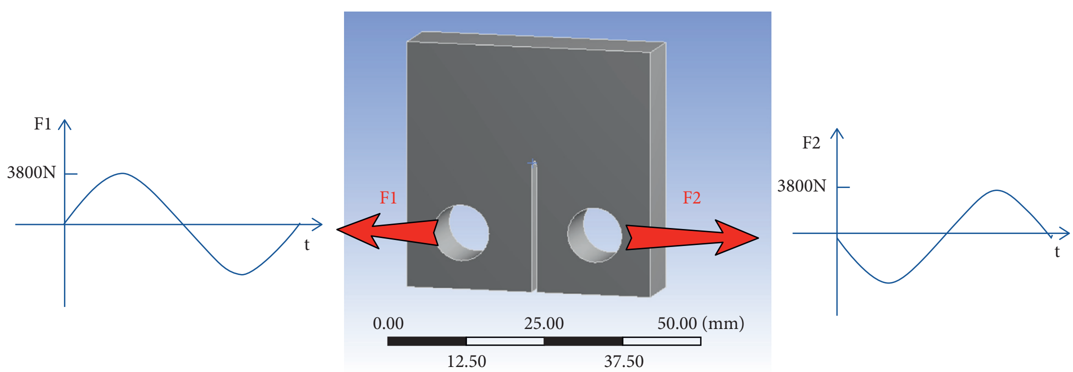

Figure 6: Loading by sinusoidal forces F1 and F2.

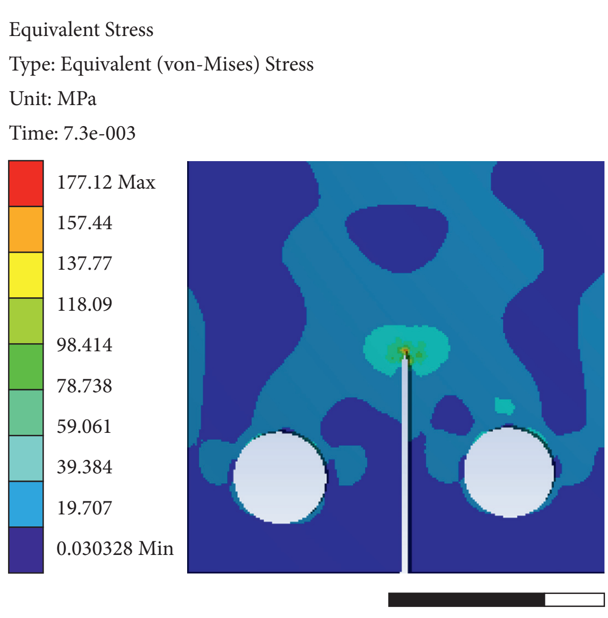

Figure 7: Equivalent stress of sample.

Table 8: The number of permitted cycles of sample structure under 3sigma equivalent stress.

	Equivalent stress S (MPa)	The number of permitted cycles N
1sigma	85.7	9.204×10^6
2sigma	124	1.167×10^6
3sigma	269	1.539×10^4

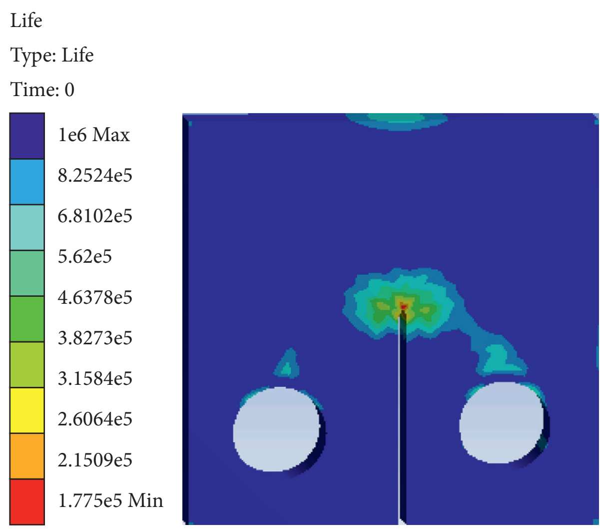

Figure 8: The fatigue life of the structure obtained by the finite element method.

Table 9: Comparison between the cross-scale method and the finite element method.

	Number of cycles N	Absolute error Δ	Relative error δ
Fatigue test	135958	—	—
Cross-scale method	159068	23110	16.99%
Finite element simulation	177500	41542	30.54%

the maximum number of cycles is 10^6. The top of the crack is most prone to fatigue failure. Fatigue damage to the sample occurs when the number of cycles at the top of the crack reaches 177500.

The number of cycles, 135958, which is obtained through the fatigue test, is selected as the real value. Table 9 lists the comparison of the error values between the cross-scale method and the finite element method. The number of fatigue cycles obtained by the cross-scale method is 159068, the absolute error is 23110, and the relative error is 16.99%. In the finite element simulation method, the number of fatigue cycles obtained is 177500, the absolute error is 41542, and the relative error is 30.54%. The absolute error of fatigue cycles obtained by the cross-scale method is 18432 less than the absolute error obtained by the finite element method. The cross-scale method obtains a relative error of fatigue cycles that is approximately twice as small as that obtained by the finite element method. The results show that the error value calculated by the cross-scale method is smaller than that calculated by the finite element software directly, whether the error is relative or absolute. Therefore, the cross-scale method has certain advantages and is better suited to solving the problem of fatigue life. Since the fatigue damage process of the high-speed rail gearbox is a long process, the error value of the above method is acceptable.

6. Conclusion

In this study, research work is carried out on a type of typical nonmaintainable long-term service structures, such as high-speed train gearbox shell, nuclear power plant pressure vessels, and aviation turbine disks. Because the high-speed train gearbox shell has the characteristics of long design life, lack of actual failure data, complex structure, high test cost, disabled to conduct an independent test for a single shell, material dispersion, and so on, it is not suitable to obtain fatigue test data via the traditional test method and the finite element simulation technology.

For this issue, the cross-scale concept from materials science is introduced. The whole gearbox shell is divided into the scale levels of "material structure" and analyzed separately based on its aspects. For the material level, this study carries out 26 sets of rotating bending fatigue tests and uses the two-parameter Weibull distribution to describe the fatigue life distribution of materials. By calculating the number of cycles under different stress amplitude conditions, the S-N curve of the gearbox housing material is linearly fitted using the least-squares method. For the structural level, the three-dimensional model of high-speed train gearbox shell is established by using software Pro/E. After applying constraints and meshing, the modal analysis is carried out by using software ANSYS Workbench to obtain the first ten modal frequencies of the gearbox. By selecting the power spectral density of the Qinghai-Tibet railway track as the input power spectral density of the random vibration analysis, the power spectral density responses of the gearbox are obtained. In this study, the three-interval method is combined with the linear cumulative damage method to achieve an accurate fatigue life prediction of the high-speed train gearbox shell. The fatigue life of the gearbox shell structure is calculated to be 12588 h, and the safe running distance on the high-speed train is 3.776 million km, which far exceeds the design requirement of 2.4 million km. Therefore, the results obtained by using the cross-scale method to predict the life of the high-speed rail gearbox structure meet the design requirements.

A small-scale sample fatigue test is designed in this study to validate the effectiveness of this method. The number of fatigue cycles obtained in the fatigue test is selected as the true value. The absolute error of the number of fatigue cycles obtained by the cross-scale method is 23110, and the relative error is 16.99%, which is far less than the number of structural fatigue cycles obtained by the finite element simulation method. Therefore, the cross-scale method is more suitable for predicting the life of a high-speed train gearbox shell. The cross-scale method could be applied in aerospace, shipbuilding, automobiles, and other industries.

Data Availability

The processed data required to reproduce these findings cannot be shared at this time as the data also form a part of an ongoing study.

Disclosure

Yibo Ai and Haonan Ma are co-first authors.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Authors' Contributions

Yibo Ai and Haonan Ma contributed equally.

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