Vibration Analysis and Noise Prediction of Gear Transmission System

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Abstract. As mechanical equipment develops towards high reliability, high speed and silence, the vibration and noise of gear transmission systems have attracted more and more attention. Aiming at the vibration and noise problems generated by gearboxes during operation, the dynamic model of the gear-shaft-bearing-gearbox coupling system is established based on the dynamics and structural acoustics of the gear transmission system. The dynamic force of the bearing is calculated through advanced whine analysis as the excitation force of the gear structure system. The harmonic response analysis of the gearbox is carried out using the modal superposition method in finite element analysis to solve the vibration response of the gearbox. The acoustic boundary element mesh model of the gearbox is established using the boundary element method. The results of the harmonic response analysis are used as boundary conditions to predict the vibration and noise characteristics of the gearbox. Finally, the gear transmission system test device is established through the FZG gear testing machine, the vibration signal of the gearbox is collected, and its main frequency components are analyzed and compared with the simulation results. The results show that this method can effectively predict the vibration and noise problems of the gear transmission system, and the accuracy of the theoretical simulation is verified, which provides a reliable basis for the noise control and structural optimization of the gear transmission system.

Keywords: Gear transmission system; FEM; Vibration analysis; Structural acoustics; BEM; Noise prediction

1 Introduction

The gear transmission system is an important power and motion transmission device in industrial production. It has the advantages of high transmission efficiency and constant transmission power. It has been widely used in aerospace, automobile, shipbuilding, power equipment, mining machinery, cutting machine tools, petrochemical and other industries. In view of the vibration and noise problems of gear transmission systems during operation, domestic and foreign scholars have conducted a series of studies. Optiz [1] studied the generation mechanism of gear system noise. Gao [2] established a dynamic model of the gear transmission system by the lumped mass method and obtained the vibration response of the gear transmission by numerical solution. Chang [3] established the unit motion equation based on the finite element method and formed a system dynamic model according to the connection relationship. Zhou [4-5] obtained the excitation force of the gearbox through the transmission system dynamic model. But ignoring the separate modeling of the transmission shaft and bearing will lead to inaccurate vibration and noise prediction. He [6] established a rigid-flexible coupling multi-body dynamic model including all components of the gear transmission system and analyzed the radiation sound field of the gearbox. Park [7] reduces the gear system to a shaft-bearing-plate model and uses a circular plate in an infinite baffle to model radiated noise. Wang [8] used the FEM to establish a dynamic model that takes into account shaft flexibility. The above scholars provided a theoretical basis for vibration analysis and noise prediction, but did not systematically connect these theoretical methods. How to accurately establish an analysis model and improve calculation efficiency is the key to realizing the prediction of vibration and noise of gear transmission systems.

This paper uses the dynamic analysis software Romax to establish a model of the gear-shaft-bearing-gearbox coupling system to obtain the simple harmonic excitation force on the gearbox. The gearbox model is imported into the finite element analysis software Ansys Workbench, and the modal analysis is first performed to obtain the natural frequency and modal vibration shape of the gearbox. The exciting force is applied to the bearing seat of the gearbox, and the harmonic response analysis of the gearbox is performed using the modal superposition method to obtain the node displacement of the gearbox vibration. The response data solved by the vibration analysis is imported into acoustic software Simcenter 3D to establish an acoustic boundary element mesh, and a far-field acoustic field mesh is constructed around the gearbox. The surface sound pressure and far-field sound pressure of the gearbox are calculated using the boundary element method. Finally, a gear transmission system test device is established to collect the vibration signal of the gearbox and compare it with the simulation results to verify the feasibility of this prediction method, which provides a reliable basis for noise control and structural optimization of the gear transmission system.

2 Dynamics Simulation

2.1 Dynamic Model of Gear Transmission System

The main reason for the vibration and noise generated by the gear transmission system is the dynamic excitation of gear meshing [9]. Gear transmission system mainly consists of the gear pair, the transmission shaft, the bearing and the gearbox, etc. In the dynamic analysis software Romax, the gear pair, the transmission shaft and the bearing are modeled parametrically. The gearbox is a flexible part relative to the bearing, and the error caused by the force deformation cannot be ignored for the transmission error, which is already very small in magnitude. In order to be more in line with the actual situation, the gearbox needs to be pre-processed by finite element and imported into the dynamic analysis software Romax for correct positioning and matching. As shown in Figure 1, the dynamic model of the gear-shaft-bearing-gearbox coupling system is constructed. Finally, the load conditions required for the gear transmission are defined.

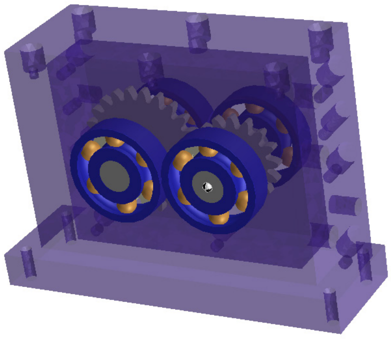

Fig. 1. Dynamic model of gear transmission system.

2.2 Bearing Force Equation of Gear Transmission System

Based on the Hertz contact theory, a rolling bearing model was established and introduced into the gear transmission system to make the research more complete. A 6406 deep groove ball bearing was selected, and its inner ring was fixed to the transmission shaft that performs circular motion. It is assumed that the balls in the rolling bearing are evenly arranged between the inner and outer raceways at equal intervals, and the contact mode between each ball and the inner and outer raceways is pure rolling without slipping [10].

When the jth ball contacts the inner and outer rings of the bearing, the pressure generated is F_j, and the nonlinear bearing force of the bearing is:

$$F_{bx} = \sum_{j=1}^{N_b} F_j cos\theta_j \tag{1}$$

$$F_{by} = \sum_{j=1}^{N_b} F_j sin\theta_j \tag{2}$$

Where N_b is the number of rolling elements in the bearing; θ_j is the angle to which the jth rolling element of the bearing rotates.

2.3 Dynamic Response of Gear Transmission System

After defining the torque and speed, the dynamic response analysis of the gear transmission system is performed. The dynamic response of the bearing is one of the more critical components in the dynamic analysis. Gear transmission error is the main excitation source causing the squeal. Therefore, it is necessary to analyze the dynamic response of the bearing of the gear transmission system under excitation such as transmission error.

The transmission error is analyzed using FFT to extract the harmonics of each order, as shown in Table 1. Since the amplitudes of the second-order and higher harmonics are negligible, they can be disregarded. Therefore, this study focuses only on the first-order harmonic of the gear pair as the excitation source for NVH.

Table 1. Transmission error.

Order	Value /μm
First	14.49
Second	3.20
Third	1.95

Taking the 6406 radial ball bearing at the input end as an example, in Romax, the frequency spectrum of the dynamic force of the bearing in the x, y, and z directions is obtained through the whine analysis calculation, as shown in Figure 2.

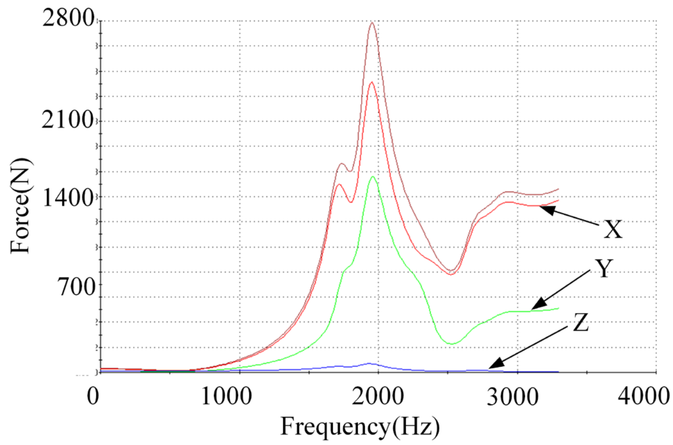

Fig. 2. Bearing dynamic force frequency response curve.

3 Analysis of Vibration Characteristics

3.1 Modal Analysis

To solve for the vibration response of the gearbox, the first step is to perform a modal analysis. The gearbox model is constructed in SolidWorks and then imported into Ansys Workbench. The material of the gearbox is defined as structural steel with an elastic modulus of $E = 2 \times 10^{11}$ Pa, Poisson's ratio $\mu = 0.3$, and density $\rho = 7850$ kg/m³. Fixed constraints are applied to the four bolt holes at the base of the gearbox, and the model is meshed using tetrahedral elements. There are many methods for extracting modes. This paper uses the Block Lanczos method to solve for the first ten natural frequencies of the gearbox, as shown in Table 2.

Table 2. The first ten order of natural frequency.

Order	Frequency/Hz	Order	Frequency/Hz
1	617.76	6	3283.4
2	1321.6	7	3399
3	1625.2	8	3619.2
4	1967.7	9	3839.8
5	2968.4	10	4313.7

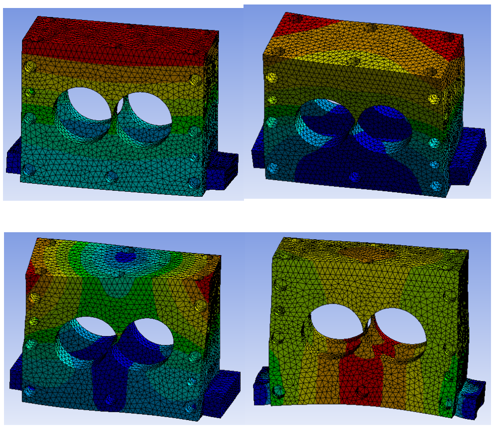

Fig. 3. First four orders of natural frequency.

In Figures 3, the first four mode shapes are shown. It can be seen that the natural frequencies of the gearbox range from 617.76 Hz to 4313.7 Hz, with the primary mode shapes involving bending and torsion. The areas with the greatest deformation are the upper end cover, bearing seats, and fixed supports. There is a noticeable overlap between the gearbox's natural frequencies and the operating frequencies of the gear transmission system. Simply increasing the stiffness of the housing to raise the natural frequencies and avoid the gearbox's operating frequencies is difficult to achieve. A comprehensive approach must be taken to address the issues of resonance and noise in the gearbox.

3.2 Harmonic Response Analysis

Before conducting acoustic radiation simulation, it is necessary to understand the response of the gear transmission system under dynamic excitation and the vibration law of the gear transmission system during operation. Therefore, it is necessary to perform a vibration response analysis on the gearbox. During the operation of the gear pair, although there is an instantaneous meshing impact, the overall vibration process of the gearbox is a periodic steady-state response. Therefore, the harmonic response analysis method is used to perform a vibration response analysis on the gearbox.

For the transmission system dynamics equation:

$$M\ddot{U} + C\dot{U} + KU = F(t) \tag{3}$$

If the node displacement and external load are in simple harmonic form, the displacement U and external load F can be expressed as follows:

$$U = U_{max}e^{i\varphi}e^{i\omega t} \tag{4}$$

$$F = F_{max}e^{i\psi}e^{i\omega t} \tag{5}$$

Where: M is the mass matrix of the system; C is the damping matrix of the system; K is the stiffness matrix of the system; F is the external load of the system; U_{max} and F_{max} are displacement amplitude and load amplitude; φ, ψ are the angular displacement phase and load phase; ω is the circular frequency;

This paper uses the modal superposition method to solve the dynamic differential equation. The principle is to decouple the dynamic equation through coordinate transformation, solve each independent equation separately, and finally superimpose the results of the solution. This method does not need to input the entire system matrix, which greatly improves the efficiency of the solution.

The bearing dynamic force is applied to the gearbox as a dynamic excitation. First, the model is simplified. At the center of the input and output ends, the equivalent mass point of the gear transmission system is established using the mass point mass21 unit. The bearing is simulated using the combin14 unit, and the bearing dynamic force is applied to the equivalent mass point, as shown in Figure 4. The calculation frequency is set from 0 to 3300 Hz.

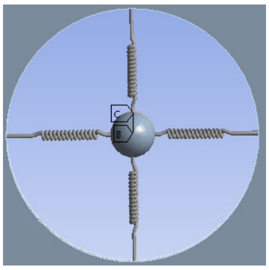

Fig. 4. Spring mass unit.

The vibration displacement of the gearbox surface nodes is extracted, and the following trends are observed: the amplitude of the vibration displacement is larger near the bearing seat, the two narrow side plates on the left and right sides of the gearbox, and the upper cover, and as the frequency increases, the response amplitude at the input end is larger than that at other parts. The peak value of the vibration displacement appears at the meshing frequency and its multiples, and the larger the excitation value, the more intense the corresponding vibration. As shown in Figures 5 and 6, the cloud diagrams of the vibration displacement at one and three times the meshing frequency.

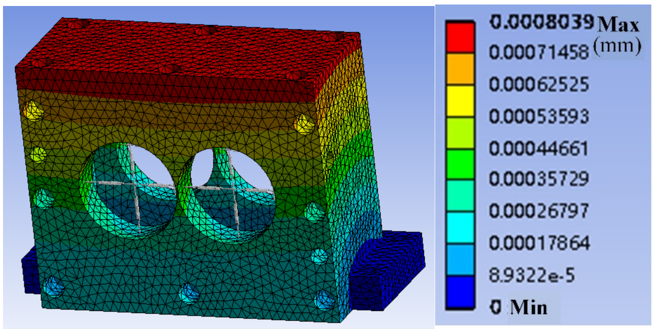

Fig. 5. Vibration displacement map at the meshing frequency.

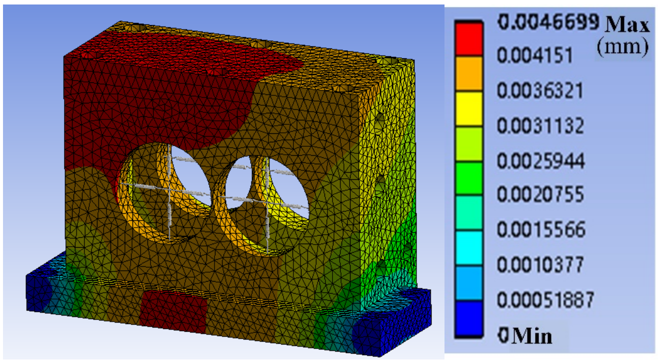

Fig. 6. Vibration displacement map at triple meshing frequency.

4 Noise Prediction

4.1 Acoustic Characteristics Analysis

The gearbox is the vibrating body with the largest contact area with the outside world. The noise generated by its surface vibration accounts for 90%- 95% of the overall transmission energy of the gear transmission system. Therefore, it can be considered that the system radiation noise is mainly generated by the vibration of the gearbox. When calculating the acoustic radiation in infinite space, the finite element method has certain limitations. Therefore, the boundary element method can be used to solve the infinite space radiation problem of gearbox radiation noise [11].

The model file of the harmonic response analysis is imported into the acoustic analysis software Simcenter 3D. The acoustic boundary element grid is divided on the surface of the gearbox, as shown in Figure 7. The displacement, torque and other results obtained from the harmonic response analysis are mapped to the acoustic grid. At the same time, the microphone grids of the far field and the surface are constructed.

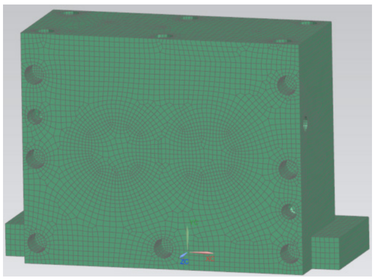

Fig. 7. Acoustic boundary element mesh.

The gear meshing frequency and its multiple frequency are the primary components that excite the gearbox and radiate noise. Therefore, these frequencies are analyzed in detail. Figures 8 show the surface sound pressure distribution of the gearbox at the single and triple meshing frequencies (around 600 Hz and 1800 Hz). The figures reveal that the bearing support, directly excited by the bearing's dynamic force, exhibits significant vibration and a relatively high surface sound pressure level. The gearbox's bottom support, which is constrained by fixed boundary conditions, also shows a relatively high sound pressure level. Comparing different frequency ranges, it is evident that the surface sound pressure peak at the third meshing frequency is higher. Due to high-frequency excitation, the gearbox exhibits a more complex vibration mode, resulting in a wider distribution of high-pressure areas, with the non-support side plate showing a particularly high sound pressure level.

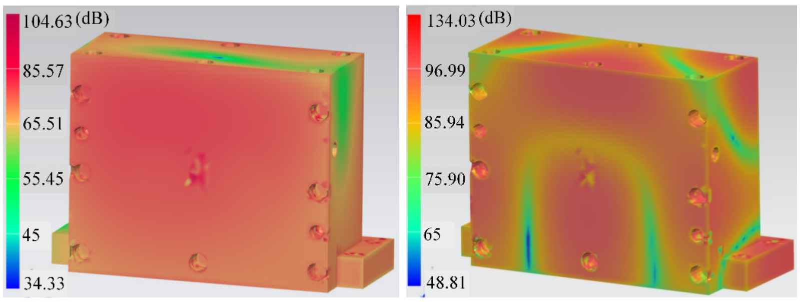

Fig. 8. Surface pressure level map.

The space in which sound pressure exists is referred to as the sound field. A rectangular grid of microphones is positioned around the gearbox to calculate the far-field sound pressure distribution at this location. Figures 9 show the far-field sound pressure distribution of the gearbox at the single and triple meshing frequencies (around 600 Hz and 1800 Hz) of the gear transmission system. The figures indicate that the far-field sound pressure level is higher near the input end, output end, and upper surface of the gearbox. There are noticeable differences in the far-field sound pressure level peaks at different meshing frequencies, with higher peaks observed under high-frequency excitation, which corresponds to the distribution of surface sound pressure and vibration signals. At lower frequencies, sound pressure level changes occur gradually, while at higher frequencies, these changes become more pronounced.

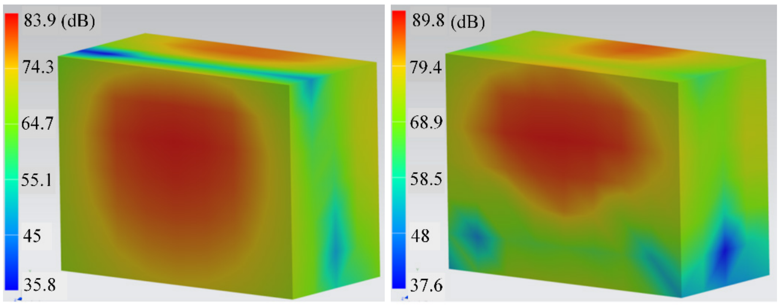

Fig. 9. Far-field pressure level map.

Three observation points at different positions on the surface of the gearbox were selected to solve the sound pressure level frequency response curve, as shown in Figure 10. It can be seen from the figure that the trend of the sound pressure level of the selected observation points changing with frequency is roughly similar, and a peak value appears near the meshing frequency and its multiples.

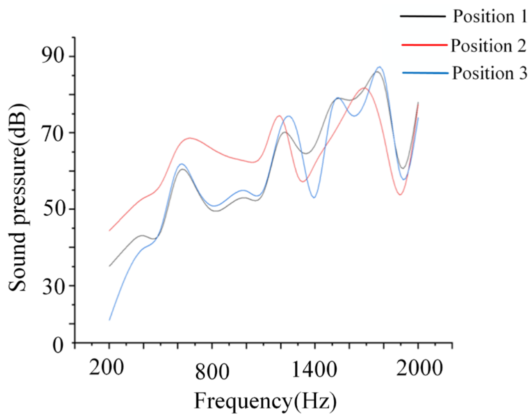

Fig. 10. Sound pressure level at different positions.

4.2 Experimental Verification

In this section, the FZG gear testing machine platform is used, as shown in Figure 11, to carry out dynamic experiments and vibration signal measurement experiments, verify the accuracy of the results of the theoretical model simulation analysis above, and provide a basis for noise prediction for the gear transmission system.

Fig. 11. Test equipment.

This paper uses an IEPE three-axis acceleration sensor, as shown in Figure 12. Two sensors are placed at the input end cover and the upper box cover respectively. When the sensor is working, based on the piezoelectric effect of the internal piezoelectric body, the stress and deformation are converted into voltage output. When the vibration test control frequency is much lower than the natural frequency of the sensor, the sensor output voltage is proportional to the force applied [12].

Fig. 12. Extraction of vibration signals.

Since the experiment uses oil bath lubrication, the vibration amplitude at the gear mesh point cannot be directly measured. As an example, the voltage signal in the y-direction from the sensor placed on the upper cover was extracted, as shown in Figure 13, and a time-domain plot of the voltage signal over time was obtained. To further analyze the frequency components of the vibration, the time-domain signal was transformed into the frequency domain using FFT, as illustrated in Figure 14. The analysis reveals that the primary frequency components with higher peak values in the experimental results are the meshing frequency and its multiple frequencies, which align with previous findings, verifying the validity of the analysis method. Additionally, rotational frequencies and their harmonics are also present in the frequency components. Due to experimental limitations, installation errors during gear assembly may differ from the theoretical values, leading to increased vibration in the gear transmission system and the appearance of other frequency components in the frequency domain map.

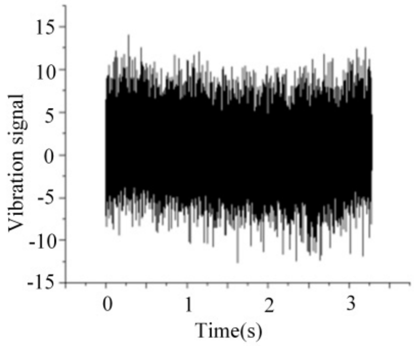

Fig. 13. Time domain map of vibration signal.

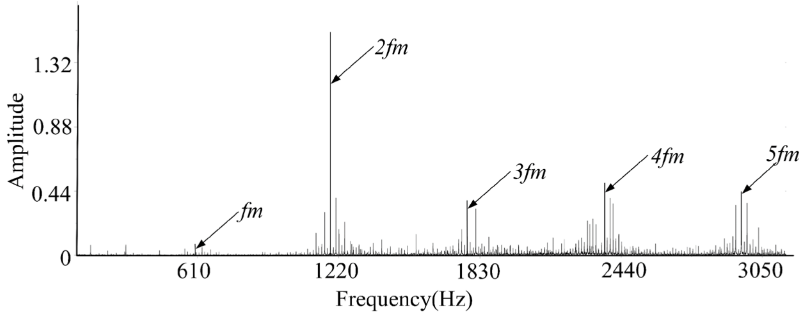

Fig. 14. Frequency domain map of vibration signal.

5 Conclusions

This paper establishes a model of the gear transmission system and conducts vibration analysis and noise prediction, which is summarized as follows:

(1) The dynamic model of the gear-shaft-bearing-gearbox coupling system is established in Romax. The dynamic force of the bearing under dynamic excitation is calculated and used as the simple harmonic excitation force borne by the gearbox.

(2) Through modal and harmonic response analysis in Ansys, the displacements of the gearbox surface nodes are obtained by FEM. In Simcenter 3D, the sound pressure cloud map of the gearbox and sound pressure level frequency response curve are calculated by BEM. It is known that the meshing frequency and its multiple frequencies are the main components that excite the gear transmission system and radiate noise outwards.

(3) The vibration signal of the gearbox was collected using the FZG gear testing machine platform, and the frequency domain diagram of the vibration signal was obtained through FFT. The main frequency components of the vibration signal were analyzed. It can be seen that this method can provide a reliable basis for noise control and structural optimization of the gear transmission system.

References

1. H. Optiz. Dynamic behavior of super and helical gears[J]. JSMESemi- International Syposium papers, Gears,1967.
2. Gao Zaida, Li Shengbo, Fu Shengping, Polyakov Roman. Analysis of Spur Cylindrical Gear Pair Vibration Characteristic Considering Error and Randomness of Tooth Surface Friction [J]. Journal of Mechanical Transmission, 2022, 46(2):8.
3. CHANG Lehao, HE Zhaoxia, LIU Geng. Dynamic modeling of parallel shaft gear transmissions using finite element method [J]. JOURNAL OF VIBRATION AND SHOCK, 2016, 35(20):47-53.
4. Jianxing Zhou, Wenlei Sun. Vibration and Noise Radiation Character-istics of Gear Transmission System[J]. Journal of Low Frequency Noise, Vibration and Active Control, 2014, 33(4):485-502.
5. Jianxing Zhou, Wenlei Sun, Li Cao. Vibration and Noise Characteristics of a Gear Reducer under Different Operation Conditions[J]. Journal of Low Frequency Noise, Vibration and Active Control, 2019, 38(2):574-91.
6. Zeyin He, Tianhong Luo, Xiangyang Xu, Tengj iao Lin. An approach to calculate radiation noise ofgear system[J]. Joumal of Vibroengineering, 2016,18(1):649-660.
7. Park, C.I. Characteristics of noise generated by axial excitation of helical gears in shaft-bearing-plate system[J]. Journal of Mechanical Science and Technology, 2015, 29(4):1571-1579.
8. WANG Shengnan, ADAYI Xieeryazidan, ZHANG Xiangfeng, ZHOU Jianxing. Vibration and Noise Analysis for Two-Stage Gear Reducer Considering Shaft Flexibility[J]. JOURNAL OF XI’AN JIAOTONG UNIVERSITY,2020,54(09):117-127.
9. Beinstingel A, Schabcrt S, Heider M, et a1. Computational prediction of stmcture-bome noise in a two-stage double-helical gearbox using harmonic balance method[J].Mechanical Systemand Signal Processing, 2023,189:110112.
10. Guo Chen, Meijiao Qu. Modeling and analysis of fit clearance between rolling bearing outer ring and housing[J]. Journal of Sound and Vibration,2019,43 8(6):419-440.
11. Meng W H. A new periodic FM-BEM for solving the acoustic transmission problems in periodic media[J]. Engineering Analysis with Boundary Elements,2023,154:54-63.
12. DENG Wen-jing. Research on the Impact of the Process Control of Random Vibration Testing by Piezoelectric Accelerometer Sensitivity[J]. Environmental Technology,2024(2):6-11.