Research Article

Numerical Noise Transfer Analysis of a Flexible Supported Gearbox Based on Impedance Model and Noise Transfer Function

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To investigate the gearbox radiation noise properties under various rotational speeds, a noise prediction method based on impedance model and noise transfer function (NTF) is proposed. One only needs to extract the NTF of the housing once rapid gearbox noise prediction under different working conditions is realized. Taking a flexible supported gearbox as a research object, the external excitation of the housing (the bearing excitation load and isolator excitation load) is calculated through a gear-housing-foundation-coupled impedance model, and the noise transfer function is simulated through the vibroacoustic-coupled boundary element model; then the radiation noise is obtained. Based on this model, the noise transfer analysis of the housing is carried out, different excitation components and NTF components are compared, and the contributions of different excitation components to noise are compared. Results show that the radiation noise of gearbox is mainly excited by the high-speed bearing, while the low-speed bearing and isolator have little influence on noise. At low speed, vertical force, axial force, and moment excitation of bearings all contribute to the radiation noise while at high speed, the gearbox radiation noise is mainly generated by vertical excitation force of bearings.

1. Introduction

Gearboxes are widely used in automobiles, ships, helicopters, etc. due to their advantages of large carrying capacity, high efficiency, and long service life. However, with the development of gear system towards high power, low noise, and light weight, the vibration and noise problems of the housing become more prominent. In order to predict and reduce the radiation noise of the gearbox in the design stage, it is necessary to establish a complete dynamic model considering the coupling effects of each component, build a noise prediction model, which can rapidly calculate noise with different system parameters and under different working conditions, and effectively control the noise from the perspective of vibration and noise transferring.

A lot of researching has been done on the simulation of gearbox radiation noise. The conventional method is to establish a weak coupling model [1] between the transmission system and the housing. Firstly, a dynamic model of the geared transmission system is established to obtain the dynamic bearing reaction load. Secondly, a finite element (FE) model of the housing is established with the dynamic bearing load applied on it to simulate the vibration response of the housing. Finally, a boundary element (BE) model of the housing is established to calculate the radiation noise of the gearbox with the vibration of the housing surface as a boundary condition. Guo et al. [2] established both a FE model and a lumped parameter model of the gearbox to compute the dynamic bearing load considering the housing flexibility. He then built a BE model to predict the radiation noise under this load. Ren et al. [3] proposed a gear-housing-foundation-coupled impedance model, which can realize the fast vibration simulation of flexible supported gear system considering the coupling relationship between transmission system, housing, and isolation system. In order to obtain the dynamic parameters of bearing, Lin et al. [4] identified the bearing stiffness and damping parameters through experimental modal analysis and least square method and built a
coupled FE model of the gear-shaft-bearing-housing system to study the vibration and noise of gearbox. Vannollebeke et al. [5] combined a flexible multibody model and an acoustic model to compute the acoustic response for a wind turbine gearbox and identified the major contributors through a panel contribution analysis. Acoustic transfer vector (ATV) method [6] is adopted to realize the fast prediction of noise. One only needs to calculate the ATV of the housing once, and then we can obtain radiation noise under different working conditions through simulating the housing vibration and multiply the ATV with the surface velocity. Abbès et al. [7] believed that the cavity fluid significantly changes the structural natural frequencies and vibration response and built a coupled acoustic-structural FE model to study the influence of inside fluid on the vibration and noise of the gearbox. In order to extend the BE method to high frequency and very large problems, Gunda and Vijayakar [8] presented a multilevel fast multipole method and applied this method to the noise prediction of vehicle gearbox. Abouel-seoud et al. [9] developed a radiation noise prediction and major noise source identification method and applied this method to the noise prediction of a multilevel fast multipole method. They computed the radiation efficiency and measured the vibration response of each gearbox plate and then adopted the acoustical power theory to calculate the whole gearbox noise.

In order to reduce the radiation noise of the housing, some scholars have studied the influence of system parameters on the noise. Allam et al. [10] built a single-stage helical gear test rig and experimentally investigated the influence of rotational speed and torque on gearbox noise. Brecher et al. [11] divided a two-stage gearbox into source, path, and receiver and experimentally studied mutual interactions between source and receiver by changing the geometry of the intermediate shaft. Results show that the first-stage gear pair makes the most contribution to the excitation and radiation noise. Zhang [12] proposed a transmission system-housing coupled multidynamic model and studied the influence of pressure angle, helix angle, and contact ratio on the whine noise. Zhou et al. [1] analyzed the influence of rotational speed and torque on gearbox radiation noise. The noise increases with the increases of rotational speed in a manner described by Kato formula except for resonance speed. The gear vibration changes from both-sides collision, single-side collision, to normal meshing stage, with the increase of torque. And under heavy load conditions, radiation noise changes with torque in a logarithm form. Guo et al. [2] studied the influence of rolling bearings and modified journal bearings on radiation noise. Barthod et al. [13] designed a perfectly controlled gear rattle noise test rig and experimentally studied the influence of excitation parameters and mechanical gearbox parameters on rattle threshold and rattle noise.

There is also a lot of researching on gearbox noise reduction. Patil et al. [14] summarized the practical techniques used in low noise gearbox design, e.g., high contact ratio gears, tooth modification, and rigid housing. Nayak et al. [15] achieved about 3—3.5 dB noise reductions of the Ashok Leyland gearbox by tooth modification and further reduced about 6—6.5 dB sound pressure by changing the spur gears into helical gears. He et al. [16] built a rigid-flexible model of the transmission system-housing-coupled system to calculate the dynamic bearing reaction load and built a vibroacoustic model to calculate the air-borne noise. He proposed the response surface method to map the structure parameters to the airborne noise and adopted the normal-boundary intersection method to optimize the gearbox noise. Tosun et al. [17] deemed that the gear whine noise depends on the noise transfer path characteristics and investigated the main noise transfer path of gear engagement noise in the interior cabin by operational path analysis. Finally the whine noise is reduced through adding masses on the analyzed dominant paths. Ribs are widely used in the gearbox to enhance its stiffness and reduce noise. Moyne and Tébec [18] divided the functions of ribs into three types, i.e., "vibrating," "obstacle," and "source," and found that the acoustic influence of ribs cannot be ignored. Tanaka [19] found that the first-stage mesh component domains the gearbox noise. He then added ribs near the antinodes of gearbox’s vibration mode to reduce the noise. Zhou et al. [20] proposed the concept of panel acoustic contribution of gearbox and reduced the radiation noise through increasing the plate thickness and adding ribs based on the panel acoustic contribution and mode shapes. Tao et al. [21] proposed a gearbox vibration and noise prediction procedure and also improved noise characteristics by improving gearbox stiffness. Liu et al. [6] combined the ATV, modal acoustic contribution analysis, and panel acoustic contribution analysis to find the maximum acoustic contribution region in a gearbox. And they further combined the acoustic contribution analysis and topology optimization to reduce the vibration and noise of the gearbox. Xiong et al. [22] experimentally studied the airborne gear rattle noise through the whole vehicle transmission loss method and found that structural transfer paths are the main noise transfer paths. They further improved the rattle noise through optimizing the clutch cable, optimizing gear shift bracket, and gear shift cables.

On one hand, there is a lack of effective fast calculation method for gearbox noise prediction under multiworking conditions. Although ATV method can reduce the calculation time of radiation noise under multi working conditions, FE models are still needed to obtain the housing vibration velocity for each working condition. This requires repeating the tedious FE modeling process and consumes a lot of computation time. On the other hand, the current research on the optimization of gearbox radiation noise assumes that the excitation is determined and only the structure of the gearbox is optimized. For low noise design of gearbox, not only the structure of the gearbox can be optimized, but also the excitation can be reduced by optimizing the transmission system parameters. The analysis of gearbox noise transfer properties can provide guidelines for the low noise optimization of transmission system.

Noise transfer function (NTF) has achieved a lot of successful applications in the vehicle NVH (noise, vibration, and harshness) analysis. It is a test-based analysis method that first [23, 24]. With the development of simulation technology, CAE (computer aided engineering) method is applied to the NTF analysis [25, 26]. This method can realize the fast noise calculation of multiple working conditions, the hybrid modeling of theoretical and experimental parameters, and the analysis of noise transfer characteristics.
In this paper, gear system impedance model and NTF method are used to calculate the radiation noise of a flexible supported gearbox under multiple working conditions. On this basis, the noise transfer characteristics of the flexible supported gearbox are analyzed. Different excitation load components and NTF components are compared and analyzed, and the contribution of different excitation components to radiation noise is compared, so as to provide guidelines for low noise design optimization of gearbox.

2. Impedance Model and NTF-Based Radiation Noise Prediction

2.1. Impedance Modeling of Gear System and Excitation Computing of Housing. The first step in predicting the gearbox noise is to establish a dynamic model of the gear system so as to determine the excitation load of the housing. The flexible supported gear system model used in this paper is shown in Figure 1(a), which includes transmission system, housing, vibration isolator, and foundation. As shown in Figure 1(b), the transmission system is a single-stage helical gear transmission system supported by rolling bearings. The speed range of the input shaft is 50∼15000r/min, and the torque on the output shaft is 1200N·m. The basic gear parameters are shown in Table 1.

Firstly, the gear system is divided into subsystems such as gear, shaft, bearing, housing, isolator, and foundation. Then, each subsystem is modeled separately. Finally, the coupled impedance model of the gear system is established by impedance synthesis method [3]. The excitation included in the model is the gear transmission error excitation. The obtained housing excitation load has taken into account the dynamic coupling effect of the whole system.

2.1.1. Gear Subsystem. The gear pair is modeled as a mass-spring-damper element with six degree of freedom (DOF). The dynamic equation of gear subsystem is as follows [3]:

\[ M^{GP} \ddot{x}^{GP}(t) + C^{GP} \dot{x}^{GP}(t) + K^{GP}(t)(x^{GP}(t) - e^{GP}(t)) = f^{GP}(t), \]

where \( x^{GP} \) is the displacement vector of gear nodes, \( M^{GP} \) is the mass matrix of gears, \( C^{GP} = c^{GP}VV^T \) is the mesh damping matrix, \( c^{GP} \) is the mesh damping, \( K^{GP}(t) = k^{GP}VV^T \) is the time-varying mesh stiffness matrix, \( k^{GP}(t) \) is the normal mesh stiffness, \( f^{GP} \) is the reaction load vector of shaft to gear, and \( e^{GP} \) is the comprehensive mesh error vector.

Equation (1) can be approximated into a frequency domain form [3]:

\[ Z^{GP}(\omega)V^{GP}(\omega) = F^{GP}(\omega) + F^{ext}(\omega), \]

where \( Z^{GP} \) is the impedance matrix of the gear pair, \( V^{GP} \) is the velocity vector of the gear pair in frequency domain, \( F^{GP} \) is the external excitation from shaft system, and \( F^{ext} \) is the transmission error excitation in the frequency domain.

2.1.2. Shaft Subsystem. The shaft subsystem is modeled by the generalized FE method. It is divided into a series of shaft segments according to the size of the cross section and the position of gears, bearings, and power input and output points. Each shaft segment is modeled by Timoshenko beam element, which contains two nodes and six DOF per node. After assembling the corresponding matrices of each shaft segment and considering the external forces, the equation of motion can be yielded as follows:

\[ M^{sh} \ddot{x}^{sh}(t) + C^{sh} \dot{x}^{sh}(t) + K^{sh} x^{sh}(t) = f^{sh}(t), \]
where $M^{sh}$ is the mass matrix of the shaft system, $C^{sh}$ is the damping matrix, $K^{sh}$ is the stiffness matrix, $x^{sh}$ is the node displacement vector, and $F^{sh}$ is the external force.

In the frequency domain, the impedance equation of the shaft subsystem can be obtained as follows:

$$Z^{sh}(\omega)V^{sh}(\omega) = F^{sh}(\omega),$$

(4)

where $Z^{sh}(\omega) = j\omega M^{sh} + C^{sh} + K^{sh}/(j\omega)$ is the impedance matrix of the shaft system, $V^{sh}(\omega) = j\omega X^{sh}(\omega)$ is the velocity vector of shaft system in the frequency domain, $X^{sh}(\omega)$ is the frequency domain form of $x^{sh}(t)$, and $F^{sh}$ is the external force vector of shaft system in the frequency domain.

2.1.3. Bearing Subsystem. The bearing is modeled as a spring-damper element, which consists of two nodes. One node is connected to the shaft and the other is connected to the housing. The two nodes form stiffness matrix and can be expressed as

$$K^{Br} = \begin{bmatrix} K_{nr} & -K_{nr} \\ -K_{nr} & K_{mr} \end{bmatrix}$$

(5)

where $K_{mr}$ is the single-node form stiffness matrix with six DOF.

Given the damping characteristics and external force, the impedance equation of the bearing subsystem can be yielded:

$$Z^{Br}(\omega)V^{Br}(\omega) = F^{Br}(\omega),$$

(6)

where $Z^{Br}(\omega) = C^{Br} + K^{Br}/(j\omega)$ is the impedance matrix of bearing subsystem, $C$ is the bearing damping, $V^{Br}$ is the velocity vector of bearing, and $F^{Br}$ is the external force vector of bearing.

2.1.4. Housing Subsystem. A FE model of the housing is built (as shown in Figure 2). The material is steel, the density is 7850 kg/m$^3$, Young’s modulus of 2.07 $\times$ 10$^{11}$ Pa, Poisson’s ratio is 0.3, and the damping ratio is 2%. The 4-node tetrahedral element is used to mesh the housing, and the element size is 5 mm. The FE model consists of about 61,000 nodes and 315,000 elements. Four bearing holes and six bolt holes are coupled to the central nodes through distributed force coupling. Four bearing holes central nodes and six isolator connection nodes constitute 10 external nodes. Block Lanczos method is used to process modal analysis, and the first 500 modes are extracted.

The arbitrary mobility matrix element can be described by modal parameters.

$$Y_{ip}(j\omega) = j\omega \sum_{r=1}^{n} \frac{u_{r}^{*}u_{p}^{*}}{K_{r} - \omega^{2}M_{r} + j\omega C_{r}},$$

(7)

where $u$ is the modal shape, $K_{r}$ is the modal stiffness, $M_{r}$ is the modal mass, $C_{r}$ is the modal damping, and $r$ is the mode order.

The impedance matrix can be yielded through matrix inversion of mobility. The impedance equation can be expressed as

$$Z^{GB}(\omega)V^{GB}(\omega) = F^{GB}(\omega),$$

(8)

where $V^{GB}$ is the velocity vector of the housing, and $F^{GB}$ is the reaction load of bearing and isolator on housing.

2.1.5. Isolator Subsystem. The rubber isolator widely used in engineering practice is adopted. The shape of the isolator is a cylinder with a diameter of 40 mm and a length of 40 mm. The modulus of elasticity is $1 \times 10^{10}$ Pa, density is 1000 kg/m$^3$, Poisson’s ratio is 0.49, and loss factor is 0.1. The isolator is modeled as continuous Timoshenko beam element [3]. The dynamic transfer matrix is obtained by the wave equation, which can then be converted into an impedance matrix.

The impedance equation of the isolator subsystem can be obtained:

$$Z^{ISL}(\omega)V^{ISL}(\omega) = F^{ISL}(\omega),$$

(9)

where $Z^{ISL}$ is the impedance matrix of isolator subsystem, $V^{ISL}$ is the velocity vector of the isolator, and $F^{ISL}$ is the external excitation force vector.

2.1.6. Foundation Subsystem. The FE model of the foundation is shown in Figure 3. The material is steel with a density of 7850 kg/m$^3$, Young’s modulus of 2.07 $\times$ 10$^{11}$ Pa, Poisson’s ratio of 0.3, and a damping ratio of 2%. The model is meshed with 4-node tetrahedral element. The element size is 5 mm. The model contains about 61,000 nodes and 265,000 elements. The six isolator connection nodes are defined as external nodes. The bottom area is constrained. The block Lanczos method is used for modal analysis, and the first 500 modes are extracted. The model consists of six external nodes, and each node has three DOF, so the dimension of the mobility matrix $Y$ is 36 $\times$ 36.

Similar to the housing, the impedance equation of the foundation can be expressed as

$$Z^{BS}(\omega)V^{BS}(\omega) = F^{BS}(\omega),$$

(10)

where $V^{BS}$ is the velocity vector, and $F^{BS}$ is the isolator excitation load.

2.1.7. Impedance Synthesis. When dynamic models of all subsystems are built, impedance synthesis approach [3] is adopted to establish the impedance equation of the whole system. Impedance equation of the coupled system can be directly obtained by assembling impedance matrix elements according to the node number. Assume no external load acting on the interfaces between subsystems. The impedance equation of the coupled system can be yielded as follows:

$$Z(\omega)V(\omega) = F(\omega).$$

(11)

Only the gear transmission error excitation is applied in (11). The impedance matrix $Z$ and the excitation vector $F$ are known, so the vibration velocity can be solved. When the system vibration velocity is obtained, the excitation load of the housing including the dynamic reaction load of the bearing and isolator can be determined by (8).
2.2. Extraction of NTF. The FE model of the housing has been previously built (as shown in Figure 2). The first 500 nodes are analyzed and extracted, and the first six vibration modes are shown in Figure 4. The first mode is the bending deformation of the front and rear panels, the second is the swing deformation of the housing, the third is the combination of the swing and the bending of the front and rear panels, the fourth is the twist deformation of the housing, the fifth is the bending deformation of the front, rear, and top panels, and the sixth is the local bending deformation of the top panel.

The vibroacoustic coupled model of the housing is established in Virtual Lab software. The structure is meshed with finite element shown in Figure 2. The surface is meshed with boundary element shown in Figure 5, the bearing holes and bolt holes are filled in the model, and the mesh size is 8mm. The fluid is air with a density $\rho_0 = 1.2$ kg/m$^3$ and a propagation speed $c = 343.4$ m/s. The bearing holes coupling nodes and bolt holes coupling nodes of the FE model of the housing are defined as load nodes. The ground surface is defined as the symmetry plane, and 19 ISO half-sphere filed points are arranged around the housing.

The coupled FE/indirect BE model can be solved by the following equation [27]:

$$\begin{bmatrix} K_s + j\omega C_s - \omega^2 M_s \end{bmatrix} \omega_i \begin{bmatrix} L_c \end{bmatrix} = \begin{bmatrix} F_s \end{bmatrix},$$

where $K_s$ is the structural stiffness matrix, $C_s$ is the structural damping matrix, $M_s$ is the structural mass matrix, $\omega_i$ is the structural displacement, $L_c$ is the coupling matrix, $F_s$ is the excitation load, $\omega$ is the excitation frequency, and $\omega$ is the index of field point, and each element can be expressed as

$$p_i(\omega) = \sum_{j=1}^{10} \sum_{k=1}^{6} NTF_{ijk}(\omega) F_{jk}(\omega),$$

where $i$ is the index of field point, $j$ is corresponding to four bearing hole coupling nodes and six bolt hole coupling nodes, $k$ is corresponding to six DOF at each direction, $p_i$ is the sound pressure at field point $i$, and $F$ is the external excitation of the housing.

The sound pressure level can be expressed as

$$L_p = 10 \log_{10} \left( \frac{p}{P_0} \right),$$

where $L_p$ is the sound pressure level, $p$ is the effective sound pressure at any field point, and $P_0 = 2 \times 10^{-5}$ Pa is the reference sound pressure.
In order to comprehensively consider the radiation noise of all field points and reduce the influence of the selection of field points on the subsequent analysis results, the radiation noise of each field point is averaged as follows:

\[
L = 10 \log_{10} \left( \frac{1}{n_{fp}} \sum_{i=1}^{n_{fp}} 10^{0.1L_i} \right),
\]

where \( n_{fp} \) is the number of field points, and \( L_i \) is the sound pressure level of field point \( i \).

2.4. Model Validations. In order to verify the NTF method proposed in this paper, it is compared with ATV and modal based vibroacoustic (VA) method [27]. At 3200 r/min, the radiation noise calculated by various methods is shown in Figure 6. NTF method and VA method are based on the same FE/BE model, so the calculation results are completely consistent. However, the ATV method does not take into account the vibroacoustic coupling effect, resulting in slightly different sound pressure. The noise at mesh frequency is significantly higher than the harmonics. The maximum error between NTF method and ATV method is 0.8 dB.

When calculating noise under different working conditions, VA method needs to repeatedly carry out tedious FE modeling and noise calculation. Each FE modeling takes about 20 min and each calculation takes about 15 min. The ATV method only needs to calculate the ATV of the housing once, but it still needs to repeat tedious loading in FE model and vibration and noise calculation for different working conditions. It takes about 10 hours to calculate ATV (600 frequencies), about 20 minutes for each loading, and about 10 minutes for each calculation of vibration and noise. The NTF method only needs to calculate the NTF function of the housing once, and the radiation noise can be directly calculated according to the load under different working conditions. It takes about 10 hours to calculate the NTF (600 frequencies) and less than 1 minute to calculate the noise each time. It can be seen that the calculation time of NTF function and ATV function is basically the same, but NTF method does not need to repeatedly simulate housing vibration like ATV method, so NTF method takes shorter time and is more convenient to use for multworking conditions. The disadvantage of NTF method is that the loading position needs to be defined in advance, so its applicability is not as wide as ATV method.

2.5. Mesh Sensitivity Study. In order to analyze the influence of the FE model mesh size on system vibration, the element size of the housing is changed from 5 mm to 6 mm. The influence of mesh size on the dynamic bearing reaction force and housing vibration is shown in Figure 7.
that the dynamic bearing reaction force is almost unchanged, and the housing vibration only slightly changed at high frequencies. This demonstrates that the mesh size used in the housing FE model is reasonable.

The element size of the foundation FE model has little influence on the housing vibration, so the results are not shown here.

In order to analyze the influence of BE model mesh size on the radiation noise, the element size is changed from 8 mm to 10 mm. The influence of mesh size is shown in Figure 8. It can be seen that the mesh size has limited influence on the radiation noise, which indicates the rationality of the BE model.

3. Numerical Noise Transfer Analysis

3.1. Comparison between Excitation Load. The excitation load of the housing includes four bearing excitation loads and six isolator excitation loads in terms of position and six DOF loads in terms of direction. For convenience of discussion, different excitation load components are classified and compared.

The root mean square (RMS) of the bearing excitation load in different positions is shown in Figure 9. Because the units of force and moment are not consistent, they cannot be directly compared, so the comparison is made for different bearing forces and moments, respectively. Figure 9(a) is the excitation force in translation direction, which is superimposed and synthesized by the RMS of excitation force $F_x$, $F_y$, and $F_z$. Generally speaking, below 2500 $r/min$, the excitation force of bearing 3 and bearing 4 on the low-speed shaft is larger than that of bearing 1 and bearing 2 on the high-speed shaft. Above 2500 $r/min$, the excitation force of high-speed bearing is larger than that of the low-speed bearing, and the excitation force of bearing 2 is greater than that of bearing 1 and bearing 3 is greater than bearing 4. Some resonance speeds and peak values are shown in Table 2.

Figure 9(b) is the excitation moment of the bearing, which is the superposition of the RMS of $M_x$ and $M_y$. It can also be seen that, below 2500 $r/min$, the excitation moment of bearings 3 and 4 on the low-speed shaft is greater than that of bearings 1 and 2 on the high-speed shaft. The opposite is true at high speed. Key resonance values are shown in Table 3.

Figure 10 shows the RMS of the bearing excitation load in different directions. Figure 10(a) shows the comparison of excitation forces in different directions. RMS of excitation forces of the four bearings is used for superposition and synthesis. Generally speaking, the vertical excitation force $F_y$ is the largest and the axial excitation force $F_z$ is the smallest. Figure 10(b) shows the comparison of the excitation moment of the bearings in different directions. RMS of the excitation moment of the four bearings is used for synthesis.
Figure 8: Influence of BE model mesh size on radiation noise.

Figure 9: Bearing excitation load in different positions. (a) Force. (b) Moment.

Table 2: Key data in Figure 9(a).

| Rotational speed (r/min) | Bearing 1 (N) | Bearing 2 (N) | Bearing 3 (N) | Bearing 4 (N) |
|-------------------------|---------------|---------------|---------------|---------------|
| 1050                    | 6.4           | 8.6           | 37.7          | 33.5          |
| 2000                    | 46.9          | 60.2          | 102.6         | 90.2          |
| 3950                    | 138.3         | 176.9         | 52.5          | 18.1          |
| 11750                   | 102.2         | 173.6         | 35.7          | 17.0          |

Table 3: Key data in Figure 9(b).

| Rotational speed (r/min) | Bearing 1 (N-m) | Bearing 2 (N-m) | Bearing 3 (N-m) | Bearing 4 (N-m) |
|-------------------------|-----------------|-----------------|-----------------|-----------------|
| 2100                    | 0.2             | 0.6             | 0.8             | 1.0             |
| 3950                    | 0.6             | 1.8             | 0.1             | 0.2             |
Overall, $M_z$ is larger than $M_y$. Key values in Figure 10(a) and Figure 10(b) are shown in Tables 4 and 5, respectively.

Figure 11 shows the RMS of isolator excitation load in different positions. Figure 11(a) shows the excitation force, which is synthesized by the superposition of $F_x$, $F_y$, and $F_z$. Although the magnitude of the excitation force in different positions can be compared at a specific rotational speed, it is difficult to compare in the whole rotational speed range because there is no prominent component. Vibration of isolators 3 and 4 is relatively larger than other isolators near 6250 r/min and 12550 r/min. Figure 11(b) shows the RMS of isolator excitation moment, which is synthesized by the RMS of $M_x$, $M_y$, and $M_z$. There is no particularly prominent component in the whole rotational speed range.

Figure 12 shows the RMS of isolator excitation load in different directions. Figure 12(a) is the excitation force, which is the superposition of excitation forces of six vibration isolators. It can be seen that the vertical excitation $F_y$ is significantly larger than $F_x$ and $F_z$. Figure 12(b) is the excitation moment. It can be seen that $M_x$ and $M_z$ are significantly larger than $M_y$.

Comparing the bearing excitation load with the isolator excitation load, it can be found that the isolator excitation load is very small and can be ignored.

3.2. Comparison between NTFs. In order to facilitate discussion, NTFs are classified and compared according to the classification of excitation load. They are divided into NTFs at bearing nodes and isolator nodes in different positions and in different directions.

Figure 13 shows NTFs for different bearing nodes. Figure 13(a) is corresponding to the excitation force of the bearing, and the NTFs are averaged by different directions. Because the structure of housing is symmetric, the NTFs of bearings 1 and 2 are exactly the same, and the NTFs of bearings 3 and 4 are exactly the same. At low frequencies, NTFs of bearings 3 and 4 are about 1.1 dB larger than those of bearings 1 and 2. Figure 13(b) shows NTFs corresponding to bearing excitation moment, which are obtained by averaging in different directions. At low frequencies, the NTF of bearings 1 and 2 is about 1.7 dB larger than that of bearings 3 and 4. It can be seen from Figure 13 that different excitation positions of bearings have little influence on the low-frequency radiation noise.

Figure 14 shows NTFs of bearing in different directions. Figure 14(a) is corresponding to bearing excitation force, which is obtained by averaging the NTFs at four bearing nodes. It can be seen that NTF corresponding to axial force is much larger than NTFs in other directions under 1500 Hz. This is because the axial force can more easily excite the bending mode of housing panels, thus making noise larger. With the increase of the excitation frequency, the NTFs in other directions increase while the trend of NTF in the axial direction remains unchanged; this decreases the difference between the NTF in the vertical and axial direction.

| Rotational speed (r/min) | $F_x$ (N) | $F_y$ (N) | $F_z$ (N) |
|-------------------------|-----------|-----------|-----------|
| 2200                    | 63.8      | 127.8     | 22.5      |
| 3950                    | 105.2     | 242.7     | 37.9      |
| 5950                    | 34.9      | 83.8      | 7.4       |
| 11750                   | 92.2      | 220.8     | 14.6      |

| Rotational speed (r/min) | $M_x$ (N·m) | $M_y$ (N·m) |
|-------------------------|-------------|-------------|
| 2200                    | 1.6         | 0.5         |
| 3950                    | 1.8         | 0.8         |
| 11750                   | 1.3         | 0.4         |
moment. It can be seen that, under 1200Hz, the NTF corresponding to $M_x$ and $M_y$ is basically the same but at high frequencies, the NTF corresponding to $M_x$ is significantly larger than $M_y$.

Figure 15 shows NTFs of isolator in different positions. Figure 15(a) is NTFs corresponding to the isolator excitation force, which is obtained by averaging in translational directions. Because the housing has a symmetrical structure, isolators 1 and 2, 3 and 4, and 5 and 6 have the same NTFs. Generally speaking, the NTFs in different positions are almost the same at low frequencies. Only at about 1020 Hz, the NTFs of isolators 3 and 4 are relatively larger than those in other positions. Figure 15(b) shows NTFs corresponding to the isolator excitation moment. Different isolator moment excitation positions have little effect on noise under 2000 Hz.

Figure 16 shows NTFs of isolators in different directions. Figure 16(a) shows NTFs corresponding to the excitation force of isolators, which is obtained by averaging the NTFs of six vibration isolators. It can be seen that NTF in the vertical direction is about 10 dB larger than that in the other two directions in the whole frequency range. Figure 16(b) shows NTFs corresponding to the excitation moment of isolator.
can be seen that NTF in $M_x$ direction is 12.1 dB larger than $M_z$ direction, and $M_x$ direction is 23.2 dB larger than $M_y$ direction.

### 3.3 Comparison between Radiation Noise

According to (15), the total radiation noise is a linear superposition of noise under different excitation loads. Therefore, radiation noise is classified and discussed according to excitation load components in different positions and directions.

Figure 17 shows the comparison between bearing excited noise and isolator excited noise. It can be seen that the radiation noise is mainly generated by the bearing excitation, and the isolator has little influence. The average radiation noise generated by the bearing load is 36.6 dB higher than the noise generated by the isolator. This is because the excitation of the bearing is significantly greater than that of the isolator. Since radiation noise caused by isolator excitation is small, only the bearing excited noise will be discussed later.

Figure 18 shows the comparison of radiation noise generated by bearing excitation in different positions. On the whole, radiation noise generated by bearings 1 and 2 on the high-speed shaft is larger than that generated by bearings 3 and 4 on the low-speed shaft, which is 12.3 dB higher on average. This is because the excitation of bearings 1 and 2 is greater than that of bearings 3 and 4. However, the noise generated by bearings 3 and 4 cannot be ignored at some
special rotational speed. For example, at 1100 r/min the noise of bearings 1 to 4 is 47.1 dB, 44.7 dB, 52.2 dB, and 54.2 dB, respectively. In this case, noise excited by the low-speed bearings is larger than that by the high-speed bearings.

Figure 19 shows the radiation noise corresponding to bearing excitation in different directions. Generally speaking, the noise is mainly excited by $F_y$, $F_z$, and $M_x$ below 4500 r/min. Above 4500 r/min, the noise generated by $F_y$ is significantly higher than other directions, and the noise caused by $F_z$ is significantly reduced. This is because the axial NTF is much larger than the vertical NTF at low speed and the difference between them decreases at high speed, but the vertical excitation force is much larger than the axial force.

### 4. Experimental Validations

In order to verify the noise prediction method proposed in this paper, a back-to-back gearbox vibration and noise test rig is built. The test gearbox is flexibly mounted with a single-stage vibration isolation system that consists of six isolators, as is shown in Figure 20. The basic parameters of the herringbone gears are shown in Table 6. The torque on the high-speed shaft is 956 N·m, and the rotational speed changes from 600 r/min to 2400 r/min.

Because the exact impedance of the isolator is difficult to obtain, the impedance parameters of the isolator are transformed directly from the lumped parameters and
only the vertical DOF is considered in the simulation model.

The MI-8014 data acquisition instrument is adopted. The microphone is 130F20 with a sensitivity of 45 mV/Pa. The noise measuring point is located 1 m above the gearbox. The comparison between test results and calculation results is shown in Figure 21. The overall trends of the analytical results and the experimental results are consistent, although there are local differences. The average calculated noise is 3.0 dB lower than the test value, and the error is less than 10 dB under most working conditions. The experimental values show obvious resonance around 960 r/min and 2040 r/min, with amplitude of 79.6 dB and 86.7 dB, respectively. The corresponding peak values in the analytical model are 73.8 dB and 82.0 dB. The
reason for these errors is that the test conditions are more complex than the simulation model. In the simulation model, only the test gearbox is considered, while the slave gearbox and the electric motor will also induce noise. Although the interference noise is relative small and the slave gearbox and electric motor are far away from the microphone, it will inevitably affect the results.

5. Conclusions

In this paper, a fast gearbox vibration and noise prediction method under multiple working conditions based on impedance model and NTF is proposed. And numerical noise transfer analysis is carried out for a flexible supported gearbox. The conclusion is as follows:

1. The gear system coupling impedance model is suitable for the rapid calculation of housing excitation load considering the coupling relationship between the transmission system, housing and isolation system. The NTF method is suitable for multiworking conditions noise prediction when the load is given. The combination of impedance model and NTF provides a fast gearbox vibration and noise prediction method under multiworking conditions.

2. The high-speed bearing is the main contribution of gearbox radiation noise compared with the low-speed bearing and isolator.

3. At low speed, vertical excitation, axial excitation, and moment \( M_z \) of the bearing all contribute to radiation noise, while at high speed, gearbox noise is mainly excited by the vertical force.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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