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Original Article

Keywords: Radiated noise, Acoustic contribution, Topology optimization, Noise reduction structures, Experimental verification

Posted Date: August 10th, 2021

DOI: <https://doi.org/10.21203/rs.3.rs-764224/v1>

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Title page

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ORIGINAL ARTICLE

Optimal Design and Experimental Verification of Low Radiation Noise of Gearbox

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Received June xx, 201x; revised February xx, 201x; accepted March xx, 201x

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Abstract: In this paper, the radiation noise prediction, low-noise optimization design and experiment of gearbox are studied. Firstly, taking a single-stage herringbone gearbox as the research object, a finite element/boundary element model is established. Considering the vibration excitation of gear system, the radiation noise of single-stage gearbox is predicted based on modal acoustic transfer vector (MATV) method. Then, the maximum field point of radiated noise is determined, and the acoustic transfer vector (ATV) analysis and modal acoustic contribution (MAC) analysis are carried out to determine the region which contributes greatly to the radiated noise of the field point. Through the panel acoustic contribution (PAC) analysis, the optimization region is selected. Then, aiming at reducing the normal speed in the optimization region, topology optimization is carried out. According to the topology optimization results, noise reduction structures are added to the gearbox, and a low-noise optimization model of the gearbox is established. Finally, by measuring the radiated noise of the gearbox before and after optimization under a given working condition, the validity of the radiated noise prediction method and the low-noise optimization design method are verified by comparing the simulation and experimental data.

Keywords: Radiated noise • Acoustic contribution • Topology optimization • Noise reduction structures • Experimental verification

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1 Introduction

The vibration of the gear system in the gearbox will transmit the excitation to the box body through the bearing seat, which will not only make the gearbox generate radiation noise to the outside and affect the cabin comfort, but also further transmit the vibration to the hull through the bolt of the box foot, posing a threat to the safety of the ship. Therefore, it is the key to reduce the noise pollution of gearbox by choosing appropriate noise reduction methods based on accurate prediction of radiated noise. At present, the noise reduction measures adopted by ships at home and abroad for gearbox mainly include [1]: structural improvement design of gearbox, laying of damping materials, optimization of pedestal admittance and installation form, active control, etc. At present, aiming at reducing vibration and noise, there are two methods to guide the improvement of gearbox structure, one is acoustic contribution analysis, the other is structural optimization.

The main idea of acoustic contribution analysis is to determine the position that contributes a lot to the radiated noise of the field point based on acoustic contribution analysis, which is the main region of structural improvement, and complete the structural improvement of this region to reduce the sound pressure value of this field point. Panel contribution analysis was first applied to noise reduction in vehicles [2], and has been applied to vibration reduction and noise reduction of gearboxes in recent years [3-4]. ZHOU et al [5], determined the optimization region through panel acoustic contribution (PAC) analysis and optimized the design of this region, which reduced the

peak value of radiated noise sound pressure level by 12dB(A) after improvement. LIU [6] through real car data acquisition, comprehensive simulation calculation and the establishment of sound quality prediction model, the acoustic mode of indoor acoustic cavity is obtained, and the contribution of body panel based on sound pressure level and sound quality is analyzed and compared. Through the analysis of modal acoustic contribution (MAC), we can find out the mode that contributes a lot to the radiation noise of a certain field point, and improve the structure of the region where bending mode appears, which can also reduce the radiation noise of this field point [7]. WANG et al [8], combined with PAC and MAC analysis, put forward a method that can quickly and accurately determine the effective improvement region, and the structural improvement of this region can significantly reduce the radiated noise of the gearbox. Aiming at the problem of radiated noise in automobile transmission housing, ZONG [9] et al, proposed a method to find out the specific location of the shell that need to be improved intuitively and accurately by using MAC analysis and PAC analysis and the improved shell structure can be effectively improved the noise characteristics of the shell.

Structural optimization mainly includes size optimization, morphology optimization and topology optimization. Topology optimization allows designers to solve more design freedom, and can get better results than other optimization methods [10], so this method is widely used at present. Homogenization method [11], variable density method [12] and level set method [13] are the main methods to solve topology optimization. The variable density method is commonly used. Park et al [14], taking an air compressor as an example, determined the modal order which has the greatest influence on the noise peak value through experiments, and optimized the topology with the maximum natural frequency corresponding to this modal as the goal, and the radiated noise was obviously reduced after optimization.

In the above two methods of acoustic contribution analysis and topology optimization, the goal of topology optimization is clear and the constraints are accurate, but it is difficult to select the design domain, so it obtains good optimization results while bringing a large amount of calculation; The acoustic contribution analysis can determine a certain region which contributes a lot to radiation noise, although its calculation scale is small, but it cannot get an accurate improvement method. To sum up, if the design domain can be selected through contribution analysis, and then the accurate optimization results can be obtained through structural optimization, then the

advantages of both can be applied. There are also some researches in this field, SHU et al [15], determined the regions with large contribution through PAC analysis, and maximized the first-order natural frequency through topology optimization and morphology optimization, which significantly reduced the radiated noise after optimization. LI et al [16], aimed at improving the third-order natural frequency and reducing the noise emissivity in this region, and obtained the optimal wall thickness scheme to significantly reduce the noise in the optimized region. ZHANG et al [17] took the body wall panel noise contribution and modal contribution as the comprehensive evaluation index to establish a multi-objective optimization function of the dynamic mechanical properties of the body panel, and the topology optimization technique was used to obtain the optimal layout of the damping material.

In this paper, the radiation noise prediction, low-noise optimization design and experiment of gearbox are studied. In section 2, the definition of ATV, MATV, MAC, PAC and topology optimization are introduced. In section 3, taking a single-stage herringbone gearbox as the research object, the finite element/boundary element model of gearbox is established, and its radiated noise is predicted under the excitation of a certain working condition. In section 4, through ATV analysis and MAC analysis and the PAC analysis is carried out to screen out the panel region which contributes greatly to the radiated noise of the target field. Then, according to the topology optimization results, by adding the noise reduction structures, low-noise optimization model of the gearbox is established. In section 5, the radiated noise of the gearbox before and after optimization is measured in the semi-anechoic chamber. The results show that the low-noise optimization design method can effectively reduce the radiated noise of the target optimization field points.

2 Theoretical Basis

2.1 ATV and MATV

The acoustic Equation. can be regarded as linear under the condition of small pressure disturbance, and a linear relationship can be established between the sound pressure at a certain field point in the sound field and the vibration at the structural surface, so that the sound pressure at any position $\mathbf{r}$ in the sound field at frequency ω is as shown in Eq. (1) [18]

$$p(\mathbf{r}, \omega) = \{ATV(\mathbf{r}, \omega)\}^T \{\mathbf{v}_n(\omega)\} \quad (1)$$

where $\{ATV(\mathbf{r}, \omega)\}$ is acoustic transfer vector, $\mathbf{v}_n$ is vibration velocity in the normal direction of the structural

surface, and ω is the angular frequency.

In this way, the relationship between the sound pressure at a certain point of the sound field and the vibration velocity of the structure surface can be established through ATV. Therefore, the physical meaning of ATV can be understood as the sound pressure value caused by unit speed of a element or node at a certain field point at a specific frequency.

The main factors affecting ATV include [18]: (a) the shape of the closed boundary, which is the surface shape of the structure; (b) the value of Green's function, so the position of field point is one of the main factors affecting ATV; (c) physical parameters of acoustic medium (including density, wave velocity, etc.); (d) calculation and analysis frequency of radiated noise.

In addition, through the linear superposition of mode shape, the displacement response of structural vibration can be expressed as

$$\{u(\omega)\} = \Omega \{MRSP(\omega)\} \quad (2)$$

where $\{u\}$ is displacement of structure, Ω is matrix formed by structural modal vector, $\{MRSP(\omega)\}$ is vector formed by modal participation factor.

The displacement vector of the structure $\{u\}$ is projected on the normal direction of the structure surface, and the vibration velocity of the structure in the normal direction be expressed as

$$\{v_n(\omega)\} = j\omega \Omega_n \{MRSP(\omega)\} \quad (3)$$

where Ω_n is the matrix is composed of the component vectors of the vibration modes in the normal direction of the structural surface.

Substituting Eq. (3) into Eq. (1) to get the sound pressure at any point in the sound field as follows

$$\begin{aligned} p(r, \omega) &= \{ATV(r, \omega)\}^T \cdot i\omega \cdot \Omega_n \cdot \{MRSP(\omega)\} \\ &= \{MATV(r, \omega)\}^T \{MRSP(\omega)\} \end{aligned} \quad (4)$$

where $\{MATV(r, \omega)\}^T$ is modal acoustic transfer vector, and its expression is,

$$\{MATV(r, \omega)\}^T = \{ATV(r, \omega)\}^T \cdot i\omega \cdot \Omega_n \quad (5)$$

The physical meaning of $\{MATV(r, \omega)\}$ can be understood as the sound pressure value at a certain point in the sound field by the single modal response at a specific frequency.

2.2 MAC

The Eq. (3) can be expanded to obtain the normal vibration velocity of structure [19].

$$v_n(\omega) = i\omega \sum_{j=1}^N Q_j(\omega) \phi_{nj} \quad (6)$$

where ϕ_{nj} is the normal component of j th structural

modal shape on the surface; N is the total order of structural mode of participation calculation.

Substituting Eq. (6) to Eq. (1), the sound pressure at any point in the sound field in the frequency domain can be obtained as shown in Eq. (7).

$$\begin{aligned} p(r, \omega) &= ATV(r, \omega)^T i\omega \sum_{j=1}^N Q_j(\omega) \phi_{nj} \\ &= \sum_{j=1}^N i\omega Q_j(\omega) ATV(r, \omega)^T \phi_{nj} \\ &= \sum_{j=1}^N p_{sj}(r, \omega) \end{aligned} \quad (7)$$

where $p_{sj}(r, \omega)$ is sound pressure produced by j th structure mode.

It can be known from Eq. (7) that the total sound pressure at any field point in the sound field is obtained by superimposing the sound pressures caused by various modes of the structure. $D_{sj}(r, \omega)$ as shown in Eq. (8) is the acoustic contribution of j th structure mode.

$$D_{sj}(r, \omega) = \frac{|p_{sj}(r, \omega)| \cos(\theta_p - \theta_{pj})}{|p(r, \omega)|} \quad (8)$$

where θ_p and θ_{pj} are the phases of $p(r, \omega)$ and $p_{sj}(r, \omega)$ respectively.

The physical meaning of MAC is the proportion of sound pressure produced by j th structure mode in the total sound pressure. If the acoustic contribution of a certain mode of the structure is greater, it means that the influence of this mode on the total sound pressure is greater. When the acoustic contribution $D_{sj}(r, \omega)$ of a certain structure mode is larger, it shows that a certain structure has a larger mode participation factor $Q_j(\omega)$. According to Eq. (6), the structure has larger normal vibration velocity under this mode.

2.3 PAC

By expanding $ATV(r, \omega)$ and ϕ_{nj} , Eq. (7) can be converted into the form shown by Eq. (9).

$$\begin{aligned} p(r, \omega) &= \sum_{j=1}^N i\omega Q_j(\omega) \sum_{k=1}^m ATV_k(r, \omega) \phi_{nj} \\ &= \sum_{k=1}^m \sum_{j=1}^N i\omega Q_j(\omega) ATV_k(r, \omega) \phi_{nj} \\ &= \sum_{k=1}^m ATV_k(r, \omega) v_{nk}(\omega) \\ &= \sum_{k=1}^m p_k(r, \omega) \end{aligned} \quad (9)$$

where m is the total number of nodes, k is the node number; $ATV_k(r, \omega)$ is the ATV of the k th node; ϕ_{nj}

is the normal modal displacement of the k th node in the j th mode, and $p_k(r, \omega)$ is the sound pressure caused by the vibration of the k th node.

Assuming that a panel structure c is composed of L nodes, the sound pressure $p_c(r, \omega)$ caused by vibration of the panel structure is shown in Eq. (10).

$$p_c(r, \omega) = \sum_{k=1}^L p_k(r, \omega) \quad (10)$$

Panel acoustic contribution $D_c(r, \omega)$ of this structure is shown in Eq. (11).

$$D_c(r, \omega) = \frac{|p_c(r, \omega)| \cos(\theta_p - \theta_c)}{|p(r, \omega)|} \quad (11)$$

where θ_c is the phase of $p_c(r, \omega)$.

The physical meaning of PAC is the proportion of sound pressure generated by vibration of the panel in the total sound pressure under a certain excitation. The greater the acoustic contribution of a panel, the greater the influence of the normal vibration of the panel on the total sound pressure.

2.4 Structural topology optimization model

Generally, the location of observation points, the density and wave velocity of acoustic medium, and the calculation and analysis frequency will not change in the radiation noise calculation model. Therefore, even if the surface structure of a certain gearbox is slightly changed, the size and distribution of ATV will not be affected by the change. At this time, the main factor that has great influence on radiation noise is the normal vibration velocity of the gearbox structure surface. To sum up, in the region where the distribution of ATV on the structure surface is large, the effective sound pressure of the corresponding field point can be reduced only by reducing the normal vibration speed of the region with the largest contribution. It is better to apply the topology optimization method to solve this kind of target, so this paper applies the topology optimization equation to reduce the normal vibration velocity in this region, and the corresponding topology optimization model [20] is shown in Eq. (12).

$$\text{Optimization target: } \min_{\rho_e} \|v_{ns}(\omega)\|$$

Constraint conditions:

$$V_s \leq 30\%V \quad (12)$$

$$\|v_{nk}(\omega)\| \leq \eta \|v_{n0}(\omega)\| (k = 1, 2, 3 \dots s-1, s+1 \dots N) (\eta < 1)$$

$$\text{Design space: } 0.01 \leq \rho_e \leq 1$$

where $v_{ns}(\omega)$ represents the normal vibration velocity of node s in the finite element model of gearbox, which is located in the region with the largest acoustic contribution

to the target field point, and the normal vibration velocity of this node in the original structure is greater than that of all other nodes in the region with the largest acoustic contribution; $v_{nk}(\omega)$ represents the normal vibration velocity at node k in the finite element model of gearbox, which is located in the region with the largest acoustic contribution to the target field point; η indicates the speed limiting coefficient, which is used to control the normal vibration speed of each node in the region with the largest acoustic contribution; $v_{n0}(\omega)$ is the minimum normal vibration velocity in the region with the largest acoustic contribution to the target field point, which is constant during topology optimization; N represents the total number of nodes in the region with the largest acoustic contribution to the target field point; ρ_e is the volume density, which is a design variable in the topology optimization model.

3 Radiation Noise Prediction

3.1 Radiation noise prediction process

The process diagram of gearbox radiated noise analysis is shown in Figure 1. In order to make the study simple, the gear system is decoupled from the gearbox, and the dynamic load of the bearing obtained by solving the dynamic equation of the gear system is used as the excitation source of the gearbox to solve its radiated noise. Firstly, the complex solid modeling of gearbox is simplified by 3D modeling software. The simplified solid modeling is imported into finite element analysis software to obtain finite element model for modal analysis, and the natural frequency and main mode shape of gearbox system are obtained. Secondly, on the basis of modal analysis, the dynamic load file of bearing is read in to analyse the vibration response of gearbox, and the modal participation factor is obtained. Then, based on the finite element model, boundary element model and sound field mesh model is established for calculating ATV. Finally, the sound response is calculated based on MATV method to obtain the sound pressure frequency response function of field point.

Figure 1 Radiated noise analysis process of gearbox

3.2 Modal analysis

This paper adopts a simplified model of gearbox, and its basic dimensions are: length $\times$ width $\times$ height = 714mm $\times$ 372mm $\times$ 681mm, as shown in Figure 2. Some small features in the structure are removed by proper simplification in the structural modeling. The gear transfer system is single-stage herringbone gear transfer, and the gear system stimulates the box body through four sliding bearing seats. The box body is made of cast steel, with Young's modulus of 210 GPa, Poisson's ratio of 0.3 and mass density of 7800kg/m³. The four-node tetrahedral element Solid285 is used to mesh the structure with a mesh size of 10mm, and a total of 90482 nodes and 368771 elements are generated.

Figure 2 Solid model of gearbox

In the process of establishing the finite element model, the bearing hole center node and the upper and lower center nodes at the bolt hole of the machine foot are established, and the nodes are assigned with MASS21 mass elements with real constants MASSX, MASSY and MASSZ, which are all 1×10^{-6} . The rigid coupling relationship is established between the bearing hole center node and the bearing hole inner wall surface node. When the bearing dynamic load is applied, the excitation can be applied to the box. A COMBIN14 spring element with

longitudinal direction stiffness of 2×10^9 N/m is established between the upper and lower nodes of the pedestal bolt hole to simulate the pedestal bolt, which restricts the six degrees of freedom of the nodes of the spring element and truly simulates the installation state of the pedestal. The gearbox and the base are connected by a vibration isolator, which is simplified as a spring unit with a spring stiffness of 6.67×10^6 N/m. The finite element model obtained after gearbox pretreatment is shown in Figure 3.

Figure 3 Finite element model of gearbox

The modal analysis of finite element model of gearbox is carried out by Block Lanczos method. The working condition of the gearbox is 1600r/min, and the torque is 1066N m, therefore the peak frequency of the dynamic load of the bearing on the gearbox is 613.33Hz, its 6-times frequency is 3680Hz. Considering that the 6-times frequency (3680Hz) is less than the 100th natural frequency (4032Hz), so the natural frequency and mode shape of the first 100th order are calculated to meet the requirements of the subsequent mode-based forced response calculation. The first 20 orders of natural frequencies of gearbox are shown in Table 1.

Table 1 The first 20 Natural Frequencies of Gearbox

Order	Frequency/Hz	Order	Frequency/Hz
1	89.669	11	1322.4
2	132.76	12	1368.8
3	151.00	13	1560.6
4	907.76	14	1721.3
5	917.01	15	1755.2
6	988.63	16	1759.7
7	1047.9	17	1788.4
8	1073.9	18	1829.3
9	1230.9	19	1840.6
10	1270.9	20	1856.0

3.3 Solving ATV

The acoustic boundary element model of gearbox is based on the finite element model. The boundary element needs a face mesh(2D mesh), not a solid mesh(3D mesh).Therefore, the surface elements of the finite element solid mesh are firstly extracted, and then shell elements are used to patch bearing holes and bolt holes.In this way, the whole gearbox becomes a closed body, and it is convenient to calculate the radiated noise of the external sound field of the gearbox. Finally, the surface elements are extracted again and the mesh is coarsened to get a regular boundary element model. The working condition of the gearbox is 1600r/min, and the torque is 1066N·m. Considering that the peak frequency of the dynamic load of the bearing on the gearbox is 613.33Hz, its 6-times frequency is 3680Hz. Therefore according to Eq. 12 [18], the BEM grid size is set to 15mm, and its upper limit frequency is 3777.78Hz. Figure 4 shows the BEM model as the basis of acoustic calculation.

$$L \leq \frac{c}{6f} \quad (12)$$

where L is the boundary element grid size; c refers to the propagation speed of sound in the fluid medium, where the sound speed is 340m/s, f is the highest frequency of calculation.

Figure 4 BEM model

Figure 5 Sound field model

On the basis of boundary element model, considering the actual installation position of the gearbox, the reflecting surface and box sound field are determined. The sound speed is 340m/s and the air density is 1.225kg/m³. Firstly, inserting reflecting plane to simulate the ground. Then, defining box sound field mesh. Finally, taking 1m from the upper side, front side, left side and right side of the sound field boundary as the sound field measuring points for radiated noise, as shown in Figure 5. In this paper, the ATV is introduced to improve the calculation efficiency, in which different frequencies are calculated by asynchronous distance. Finally, the ATV of the gearbox boundary element model at different frequencies are obtained.

3.4 Gearbox radiation noise prediction

On the basis of the modal analysis result file, the dynamic load of the bearing is applied to the center node of the bearing hole, the modal superposition method is used to calculate the forced response based on the modal, and the modal participation factor is obtained. Because MATV method is used to calculate radiation noise, modal participation factor is solved by finite element model, while ATV is solved by boundary element model, so it is necessary to transfer modal data from finite element model to boundary element model. The MATV acoustic response solution is calculated based on ATV, modal data, and modal participation factor, and finally the frequency response function at the field point is obtained.

The effective sound pressure level of radiated noise at each field point is shown in Table 2 by solving the sound pressure frequency response function of the field point under the working conditions of rotating speed of 1600 r/min and load torque of 1066N·m, and the sound pressure level of radiated noise at mesh frequency and mesh frequency doubling of each field point is shown in Table 3. According to Table 2, the effective sound pressure level of field point 1 is the largest, so this field point is selected as the optimization target field point. According to Table 3, the radiated noise sound pressure level at the mesh frequency (613.33Hz) is greater than that at other doubling frequency, so the mesh frequency (613.33Hz) is selected as the optimized frequency.

Table 2 Effective sound pressure level of radiated noise at each field point (dB(A))

	Field point 1	Field point 2	Field point 3	Field point 4
Effective sound pressure level	95.19	92.61	85.79	84.23

Table 3 Radiation noise sound pressure level of each field point at meshing frequency and frequency doubling (dB(A))

Frequency	Field point 1	Field point 2	Field point 3	Field point 4
Meshing frequency	97.28	94.86	87.90	86.37
2-times mesh frequency	69.89	69.34	62.42	72.92
3-times mesh frequency	67.7	77.48	68.17	71.34

4 Low-noise Optimization Design

4.1 Optimization design process

The low-noise optimization design process of the gearbox is shown in Figure 6. The first step is to determine optimized target field point and frequency. The second step is to determine the region with large ATV value through ATV analysis. In the third step, the mode with large acoustic contribution is determined by MAC analysis, and the region with large normal vibration speed is determined by analyzing the corresponding main vibration mode. In the fourth step, the panel is divided in the region where the ATV and normal vibration velocity are both large, and the panel with the largest acoustic contribution is determined by PAC analysis. The region with the largest PAC is the region with the largest acoustic contribution. In the fifth step, a topology optimization model is established with the goal of reducing the normal vibration speed in this region. In the sixth step, according to the topology optimization results, noise reduction structures are added to the panel, and a gearbox low-noise optimization model is established.

Figure 6 low-noise optimization design process

4.2 ATV analysis

After the radiation noise calculation is completed, the ATV distribution result corresponding to the field point 1 at 920Hz can be obtained as shown in Figure 7. It can be seen that the ATV to the top surface of the gearbox is far larger than other surfaces, so the first screening result of optimization region is the top panel.

Figure 7 The distribution of ATV for field point 1 at 920 Hz

4.3 MAC analysis

After the ATV analysis is completed, the regions which may have larger acoustic contribution can be found by MAC analysis. Based on the analysis of the first 100 order modal results, the first 9th order mode with larger absolute contribution to the sound pressure level at field point 1 are extracted, as shown in Figure 8. As can be

seen from Figure 8, the contribution of the fifth and seventh modes to the modal acoustics of the field point 1 is greater, reaching 40.01% and 4.66% respectively, indicating that the total sound pressure ratio of these two modes at the field point 1 is the largest. The MAC of the other modes are all lower than 4%, or even negative. According to the results, the 5th and 7th modal shape diagram are extracted, as shown in Figure 9. It can be seen from Figure 9 that the top surface region and bearing pedestal region have a relatively large normal vibration velocity in theory.

Figure 8 Results of MAC analysis of field point 1

Combined with the ATV distribution nephogram in Figure 7 and the modal shape diagram in Figure 9, it can be seen that the acoustic contribution of the top surface of the gearbox and bearing pedestal region to the field point 1 are much greater than that of the other regions of the box. Therefore, the second screening results of optimization region are the top surface region and the bearing pedestal region.

Figure 9 5th and 7th order modal shape diagram

4.4 PAC analysis

The panel with maximum acoustic contribution can be found by PAC analysis. Therefore, according to the analysis results of ATV and MAC, two panels are divided as shown in Figure 10. The results of PAC analysis are shown in Figure 11. It can be seen from Figure 11 that the acoustic contribution of the panel 1 to the field point 1 is much greater than that of the panel 2. Therefore, the third screening result of optimization region is panel 1.

Figure 10 Panel division results

Figure 11 Results of PAC analysis

4.5 Structural topology optimization

According to the analysis of ATV, MAC and PAC, the region with the largest acoustic contribution to field point 1 is panel 1. Therefore panel 1 is selected as the topology optimization region. The gearbox topology optimization model is established with reducing the normal vibration speed in the optimization region as the goal and constraint condition, as shown in Figure 12. The topology

optimization result cloud diagram is obtained by solving the above model, as shown in Figure 13.

Figure 12 Noise reduction topology optimization model of single-stage gearbox for field point 1

Figure 13 Results of noise reduction topology optimization of single-stage gearbox for field point 1

4.6 Structure improvement design for noise reduction

According to the topology optimization results, the gearbox is designed by adding noise reduction structures. The higher the bulk density of materials, the more important the materials are. In structural improvement design, priority should be given to adding materials in this part of the region. According to the engineering experience, the thickness of the additional structures is 2.5 times that of the panel 1. Therefore, four optimization models are designed respectively, as shown in Figure 14. Therefore, the additional structures thickness of optimization model 1 and optimization model 2 is 20mm. Optimization model 1 considers all red regions in density nephogram, and optimization model 2 only considers the central red region of density nephogram compared with optimization model 1. Compared with the optimization model 1, the optimization model 3 considers the red, yellow and green regions in the density nephogram, and sets up stepped structures with a thickness of 7mm, and the additional structures of the two models have the same quality. Compared with the optimization model 2, the optimization model 4 considers the red, yellow and green regions in the density nephogram, and sets stepped

structures with the thickness of 7mm, and the additional structures of the two models have the same quality.

Optimization model 1

optimization model 2

Optimization model 3

optimization model 4

Figure 14 Optimal design model of noise reduction for single-stage gearbox

The radiation noise of four optimization models is calculated, and the effective sound pressure level of radiation noise at each field point is obtained, which is compared with the original model of gearbox, as shown in Table 4.

Table 4 Effective sound pressure level of radiated noise at each field point before and after topology optimization design of single-stage gearbox (dB(A))

	Field point 1	Field point 2	Field point 3	Field point 4	Average value
Original model	95.19	92.61	85.79	84.23	89.46
Optimization model 1	81.82	86.14	80.51	80.96	82.36
Optimization model 2	81.64	86.45	80.63	80.12	82.21
Optimization model 3	84.04	86.76	81.76	81.05	83.40
Optimization model 4	84.47	86.27	81.48	81.24	83.37

It can be seen from Table 4 that compared with the original model, the effective sound pressure levels of radiated noise in the four fields of the optimized models 1,

2, 3 and 4 are all reduced, which indicates the effectiveness of the low-noise optimization design of the gearbox.

Compared with the original model, the optimized model 1 is reduced by 12.37 dB(A), 6.47 dB(A), 5.28 dB(A) and 2.27 dB(A) respectively, and the average value of effective sound pressure level of radiated noise at each site is reduced by 7.10 dB(A). Compared with the original gearbox model, the optimized model 2 is reduced by 13.55 dB(A), 6.16 dB(A), 5.16 dB(A) and 4.11 dB(A) respectively, and the average value of effective sound pressure level of radiated noise at each field is reduced by 7.25 dB(A). Compared with the optimization model 1, the difference of radiation noise results of the optimization model 2 is less than 1 dB(A), which shows that the noise reduction effect can be achieved only by adding noise reduction structures in the center of density nephogram. Therefore, optimization model 2 is selected in subsequent test verification.

Compared with the original model, the optimized model 3 is reduced by 11.15 dB(A), 5.85 dB(A), 4.03 dB(A) and 3.18 dB(A) respectively, and the mean value of effective sound pressure level of radiated noise at each site is reduced by 6.05 dB(A). Compared with the original gearbox model, the optimized model 4 is reduced by 10.72 dB(A), 6.34 dB(A), 4.31 dB(A) and 2.99 dB(A), respectively, and the average effective sound pressure level of radiated noise at each field is reduced by 6.09 dB(A). Compared with the optimization model 3, the difference of radiation noise results of the optimization model 4 is less than 1 dB(A), which shows that the noise reduction effect can be achieved only by adding noise reduction structures in the center of density nephogram. Therefore, optimization model 4 is selected in subsequent test verification.

Based on the results of the topology optimization, there are approximately four locations where the noise reduction structures need to be added. However, according to the simulation results, the noise reduction effect can be achieved only by adding noise reduction structures in the center of the density nephogram. The reason is that the other three positions are located around the plate surface 1, close to the fixing bolt, and its vibration is limited. Therefore, its acoustic contribution is also less than that of the central position.

5 Experimental Verification

5.1 Introduction of test bench

In this paper, the mechanical power flow closed test bench system is used to test the radiation noise of the test gearbox. The test bench is mainly composed of the test gearbox and the accompanying test gearbox, both of which are single-stage gearboxes driven by herringbone gear. The two gearboxes are connected by high-speed torsion shaft, high-speed rigid shaft, low-speed torsion shaft and low-speed rigid shaft, which are connected by rigid coupling and diaphragm coupling respectively. A loading coupling is added between the high-speed shaft and the accompanying test gearbox, and the driving motor is responsible for compensating the friction loss of the whole transfer system.

Figure 15 The mechanical power flow closed test bench

- 1—Test gearbox
- 2,4,6—Diaphragm coupling
- 3—Low-speed shaft support seat
- 5—Torque tachometer
- 7—Low-speed torsion shaft
- 8—Accompanying test gearbox
- 9—Water pump
- 10—Oil pump
- 11—Frequency converter
- 12—Motor
- 13—Motor support seat
- 14, 18—Basic platform
- 15—Accompanying test gearbox base
- 16—Load coupling
- 17—High-speed torsion shaft
- 19—Rigid coupling
- 20—High speed shaft support seat
- 21—Test gearbox base

The test gearbox is located in the semi-anechoic chamber. Four sound field test points are arranged around the test gearbox, and an acoustic sensor of type 130F20 is installed at the field point. The position of the sound field

test point is consistent with that of the simulation model, as shown in Figure 16.

Figure 16 Sound field points layout of noise measuring

By loading the coupling, a torque of 1066N·m is applied; starting the motor, adjusting the motor to low speed, and gradually adjusting the motor speed to 2000r/min after the test gearbox runs smoothly. Then The sound pressure signals is collected by using the MI-8014 data acquisition system of ECON Technologies Company. Finally the data is processed by using off-line analysis software

5.2 Experimental verification of radiation noise prediction method

The measured and simulated values of the effective sound pressure level of the test gearbox at four field points are shown in Table 5, where the speed is 1600r/min and the torque is 1066N·m. It can be seen from Table 5 that the differences between simulated values and measured values of field point 1, field point 2, field point 3 and field point 4 are 2.04dB (A), 2.65dB (A), 1.5dB (A) and 1.27dB (A), respectively, which do not exceed 3dB (A). The validity of the radiated noise prediction model and method is proved.

Table 5 Comparison of simulated and measured values of effective sound pressure level of radiated noise at four field points in test gearbox (dB (A))

Field point	Field point 1	Field point 2	Field point 3	Field point 4	Average value
Simulated values	95.19	92.61	85.79	84.23	89.46
Measured values	93.15	89.96	83.94	82.96	87.50
Distance	2.04	2.65	1.85	1.27	1.96

5.3 Experimental analysis of low-noise optimization design method

When the speed is 1600r/min and the torque is 1066N·m, the measured values of the effective sound pressure level of radiated noise at four field points before and after optimization of the test gearbox are shown in Table 6. It can be seen from Table 6 that after the low-noise optimization design of the test gearbox, the measured values of the optimization model 2 and 4 decreases by 9.41dB (A) and 7.28dB (A) respectively compared with the original model at field point 1, and the average measured value of each field point decreases by 2.64dB (A) and 2.54dB (A) respectively.

The radiated noise of the gearbox before and after optimization is measured in a semi-anechoic chamber by using the mechanical power flow closed test bench. By comparing the simulation results with the measurement results, the effectiveness of the low-noise optimization design method is verified.

Table 6 Comparison of measured values of effective sound pressure level of radiated noise at four field points before and after optimization of test gearbox (dB (A))

	Field point 1	Field point 2	Field point 3	Field point 4	Average value
Original model	93.15	89.96	83.94	82.96	87.50
Optimization model 2	83.74	88.35	82.73	84.62	84.86
Optimization model 4	85.87	88.15	82.78	83.07	84.97

The measured values of A-weighted 1/3 octave sound pressure level of radiated noise at each field point, as shown in Figure 17. It can be seen from the Figure 17 that the radiated noise of field point 1 decreases obviously in the 630Hz band, which includes the meshing frequency (613.33Hz). The measured values of radiated noise in other frequency bands change little because the target frequency of low-noise optimization is the meshing frequency (613.33Hz). The measured value of radiated noise of field points 2, 3 and 4 change little, because the low-noise optimization target field point is field point 1. This shows that the low-noise structure optimization design method proposed in this paper is effective.

Figure 17 The measured value of A-weighted 1/3 octave sound pressure level of radiated noise at each field point

6 Conclusions

(1) Considering the dynamic excitation of the gear system, the A-weighted radiated noise sound pressure level of each field point of the gearbox is calculated based on the MATV method. By comparing the simulation value with the measurement value, the validity of the radiated noise prediction method is verified.

(2) According to the results of topology optimization, four kinds of noise reduction structures are added to the gearbox, and four kinds of gearbox low-noise optimization models are established. According to the noise simulation results and experimental verification, the validity of low-noise optimization design method of gearbox is verified. The noise reduction effect can be achieved only by adding noise reduction structures in the center of the density nephogram.

7 Declaration

Availability of data and materials

All data generated or analysed during this study are included in this published article.

Competing interests

The authors declare that they have no competing interests.

Funding

Supported by National Key R&D Program of China

(Grant No. 2018YFB2001501) and the Key Program of National Natural Science Foundation of China (Grant No. 51535009).

Authors' contributions

The author's contributions are as follows: LL, KK and ZX proposed the radiated noise prediction method and gearbox low-noise optimization method. LL, KK wrote the manuscript. KK, JX and JG were in charge of the entire experiment. LL and GL checked and improved the manuscript in writing. All authors read and approved the final manuscript.

Acknowledgements

Not applicable.

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