Research on vibration and acoustic radiation of planetary gearbox housing

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ABSTRACT

In this paper, an analysis model of planetary gearbox housing is constructed based on finite element method/boundary element method (FEM/BEM). Its vibration and acoustic radiation characteristics are investigated. Firstly, the finite element model is established using ABAQUS. The main factors affecting its dynamic characteristics are observed through modal analysis. Then impact of main structural parameters on transmission characteristics is investigated. Secondly, the acoustic radiation analysis model is established using VA-One on the basis of vibration characteristics analysis. Acoustic pressure nephogram,noise spectrum, noise radiation and modal contribution of housing surface are obtained through the numerical analysis. The effects of main structure parameters on noise radiation characteristics are observed. It is shown that the rigidity of the front and back plate is weaker than the circumferential rigidity of the housing. However, reinforcing the local thickness of the front and back plate cannot improve the dynamic characteristics due to the effect of shafting in the gearbox housing. Finally, acoustic protection of the housing is carried out. It can be observed that changing the loss factor can effectively reduce the noise of the housing structure. This investigation can provide technical support on the scheme design of the planetary gearbox housing.

Keywords: planetary gearbox; vibration characteristics; acoustic radiation characteristics

1. INTRODUCTION

A gear reduction unit[1] is one of the equipments critical to dynamical systems of ships. The propeller can gain revolving speed through a gear reduction unit, and then it can improve the propulsive efficiency. The reduction gear system[2-3] is also important for the optimized layout of all kinds of combined power propulsion ways and their propelling machinery. Gear products have vibration and acoustic noise problems[4] of varying degrees. At the same time, large marine reduction gear[5] is becoming of high-speed and heavy-duty, high efficiency, high stability, lightweight design, high reliability, less vibration, low noise and short design and manufacture period, which draw higher demand[6-8] in the dynamical transmission characteristics and design and manufacture capacity of gearing-down systems, therefore it is necessary to optimize and improve the design[9] of the reduction case structure and material.

On the basis of modal analysis theory, Qin Jie[10] and others allow for different boundary conditions' effect on boxes' vibration characteristic, and they verify the transmission boxes' vibration modes under different constraint conditions through experiment research. By means of theoretical analysis and experiment, Wang Ji[11] and others have studied modal analysis theory's practical application on the vibration performance of transfer gearboxes. As for noise prediction, Boundary Element Method (BEM) can be used to study the radiation noise produced by boxes within a certain range, and each part of the box's contribution to the whole radiation noise. Additionally, it can be used to predict radiation noise of the box in mid-low and medium-high frequency. The research results show that this method not only has good accuracy, it is practical, convenient, and of powerful adaptability.

On the basis of finite element method (FEM), this paper has established the box's computational and analytical model, had a numerical analysis of the vibration characteristics, analyzed each part's influence on the box's dynamic performance, got the natural frequency and their corresponding principal modes, and through calculating transfer functions, it has worked out each part's response to input load. On that basis, this paper has built up radiation noise prediction model of the box structure through BEM, calculated the radiation noise produced by the box surface and each part, calculated the

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sound pressure and nephogram, and then analyzed main structure parameter's influence on radiation noise characteristics. At last, through before-and-after comparison of acoustics protection, this paper has put forward noise reduction measures on account of altering loss factors of the box structure.

2. ESTABLISHMENT OF ANALYTICAL MODEL

This paper adopts FEM to analyze the vibration performance of the box's full-band, and adopts BEM to analyze the acoustic radiation characteristics. After determining the computing method, this paper firstly establishes the FEM analytical model in Abaqus. The radial length of the box is 1.20m, and the circumferential diameter is 1.22m. The box thickness is 6mm. The plate thickness is 10mm. The reinforcing rib thickness is 6mm. The box is made from steel. The ektexine has reinforcing rib, which is to strengthen the box. The material attributes are: E (Elastic modulus) =2.1e11Pa, μ (Poisson's ratio) =0.3, ρ (Density) =7800Kg/m^3, the structural damping is 0.2%. And then FEM model will be imported into VA one to establish the BEM analytical model. The actually received and delivered energy FE subsystems within the whole box structural model are 515, which are all plate and shells systems. The finished box FEM/BEM models are shown in Figure 1.

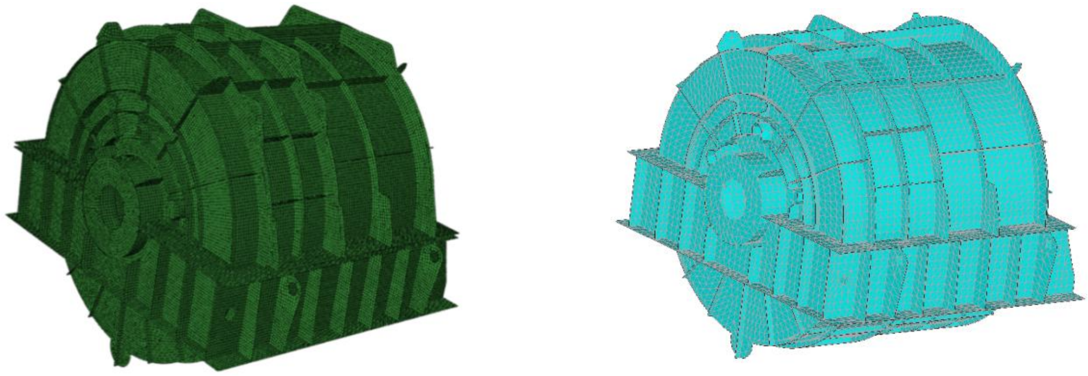


Figure 1 – FEM/BEM models of gearbox housing

This paper analyzes the vibration performance based on model analysis theory. The box structure of planetary gear reducer is a system with infinite multi-degree of freedom. When analyzing the box structure, the research mechanical structure is usually seen as a system consists of particle, rigid body, elastomer, and damper. What's more, it will be discretized into finite mutual elastic connecting rigid bodies. Thus the system of infinite multi-degree of freedom turns into a system of finite multi-degree of freedom.

Multi-degree of freedom system undamped free vibration's equation of motion can be expressed with matrix form as follows:

$$[M]\{\ddot{q}\}+[K]\{q\}=\{0\} \tag{1}$$

The solution of the equation is

$$\{q(t)\}=\{A\}\sin(\omega_n t+\varphi) \tag{2}$$

ω_n in the equation represents the natural frequency of the undamped free vibration;

$\{A\}$ represents the amplitude vector;

φ represents the phase angle.

Substitute the above solution into the equation, and then you can get

$$([K]-\omega_n^2[M])\{A\}=\{0\} \tag{3}$$

This is a system of homogeneous linear algebraic equations. If it wants the untrivial solution, $\{A\}$ cannot be $\{0\}$, and the determinant of coefficient must be zero. So it can be

$$\left|[K]-\omega_n^2[M]\right|=0 \tag{4}$$

This is the frequency equation of the system. As for numerous degree of freedom systems, this equation is nth-degree equation of ω_n^2. It has numerous roots ($\omega_1^2, \omega_2^2, \ldots, \omega_n^2$). Since the frequency ($\omega_1$, $\omega_2, \ldots, \omega_n$) has nothing to do with the starting conditions, but depends on the natural characteristics $[M]$ and $[K]$, so it is called the natural frequency of the system. They sort alphabetically by size. The least is the first natural frequency ω_1, and the biggest is the nth inherent frequency ω_n.

In modal analysis, Lanczozs is an effective method to solve large-scale eigenvalue problems. It not only ensures a certain precision, it is relatively economic. As for other methods, when there is inherent frequency or there are several inherent frequencies in the system, matrix iteration's rate of convergence will become too slow. However, Lanczozs can effectively overcome this difficulty.

Through Lanczos, p (number) minimum eigenvalues and the corresponding eigenvectors can be got. The generalized characteristic equation is as follows:

$$[K][\phi]=[\Lambda][M][\phi] \tag{5}$$

Among them, $[\phi]=\left[\{\phi_1\}\{\phi_2\}\{\phi_3\}\cdots\{\phi_P\}\right]$, $[\Lambda]=diag\left(\lambda_1,\lambda_2\cdots\lambda_p\right)$. The eigenvector satisfies the following orthogonality conditions:

$$[\phi]^T[K][\phi]=[\Lambda],[\phi]^T[M][\phi]=[I] \tag{6}$$

Lanczos combines inverse iteration and Riz orthogonalization at the same time, and it repeats until the required precision is reached.

3. VIBRATION CHARACTERISTIC ANALYSIS OF THE BOX

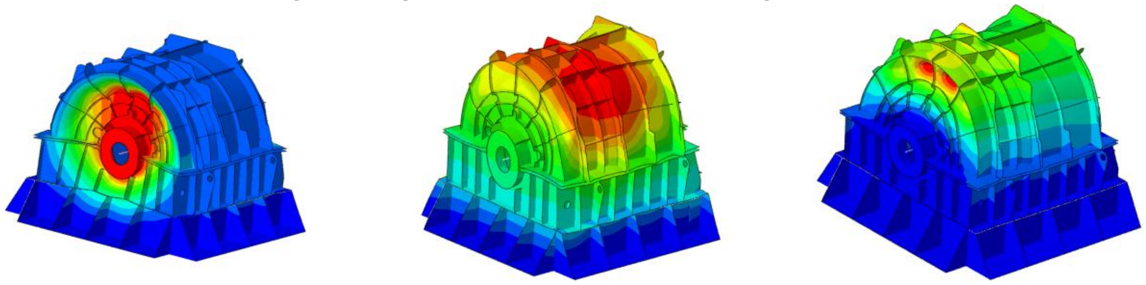
During the vibration characteristic analysis process, boundary conditions will affect the box's inherent characteristics. In consideration of the actual installation conditions, this paper builds up the base model conjoint to the box structure, and connects it with the box structure. On the basis of all the starting conditions determined, this chapter will analyze the vibration performance characteristic of the box.

Firstly, through modal analysis, this paper has got the natural frequencies and vibration modes of all structure parts, and then finds out the parts where the vibration amplitude is comparatively larger. Secondly, this paper makes a comparison between the excitation frequency and the natural frequency, and tries to make external excitation frequency avoid the box's natural frequency. It imposes three-dimensional simply supported boundary conditions on the reduction case base, carries on modal analysis, and then extracts the first six orders of principal modes from the results. The first six orders' natural frequencies and their modes of vibration are shown in Table 1.

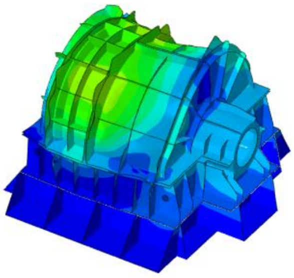
Table 1 – The first six orders' natural frequencies

Orders of mode	1	2	3	4	5	6
Frequency（Hz）	64.5	98.6	122.3	153.8	159.6	167.4

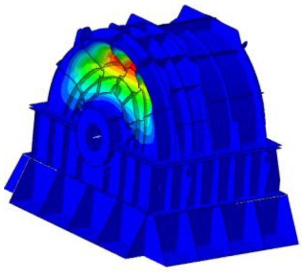
The modes of vibration got through calculation are shown in Figure 2.



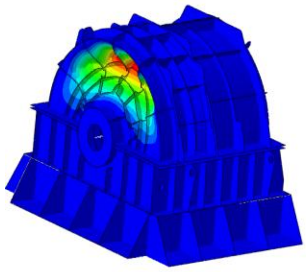
(a) First-order vibration mode (f=64.5Hz) (b) Second-order vibration mode (f=98.6Hz) (c) Third-order vibration mode (f=122.3Hz)



(d) Fourth-order vibration mode (f=153.8Hz)



(e) Fifth-order vibration mode (f=159.6Hz)



(f) Sixth-order vibration mode (f=167.4Hz)

Figure 2 – The first six orders' modes of vibration

From the above modal analysis results, it can be seen that the first few low order modes of the box mainly represent the vibration of the front and back end plate, and the upper box. The vibration of the lower box and the supporting parts conjoint with the base is not obvious. Thus it can be seen that the rigidity of these parts are weaker than that of the lower box and the supporting parts. On this basis that the structural material and structural style will not be changed, the thickness of the upper box, the front and back end plate will be changed to enhance the structural rigidity of the weak parts (to thicken the upper box, the front and back end plate with 10mm). And then this paper will recalculate the dynamic performance, and get new principal mode and its corresponding natural frequency. The natural frequencies' comparison of the thickness before and after is shown in Table 2.

Table 2 – The natural frequencies' comparison of the thickness before and after

Orders of mode	1	2	3	4	5	6
Frequency before the change（Hz）	64.5	98.6	122.3	153.8	159.6	167.4
Frequency after the change（Hz）	65.4	99.3	130.2	164.4	167.7	174.2

Table 2 shows that when the material and structure of the box has not been changed, only changing partial thickness of the box will have influence on the natural frequency of the reduction case, but the effect is not obvious. With the increase of frequency, the natural frequency intensity of the box and the mode intensity will increase, no matter the partial thickness has changed or not. In middle-high frequency band, the inherent frequency and the mode are very close. This fact indicates that changing partial thickness has little influence on the vibration performance. So in order to avoid the occurrence of resonance, the frequency of the input load in actual work should try to avoid the natural frequency.

For further analyzing the vibration performance characteristic of the box, we need to know each part's response to the input load. In order to provide theoretical support for optimizing the design and research of the structural characteristics, this paper measures each part's response of the box to load by transfer function, The input load per unit and acceleration measuring point are shown in Table 3.

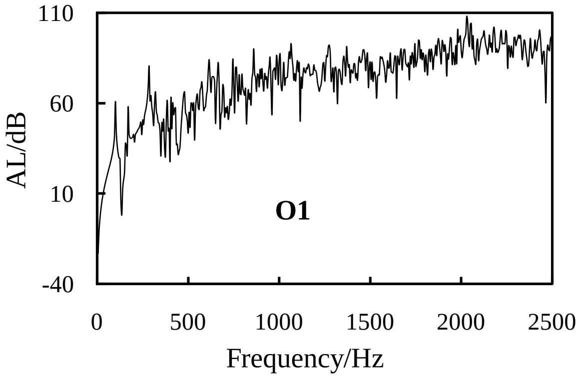
Table 3 – The input load per unit and acceleration measuring points

serial number	input load points	serial number	measuring points
I1	bearing of inlet end	O1	left at machine stand of inlet end
I2	bearing of outlet end	O2	right at machine stand of inlet end
I3	input load at upper box	O3	left at machine stand of outlet end
I4	input load at under box	O4	right at machine stand of outlet end
I5	left at support structure	O5	machine stand of outlet end
I6	right at support structure	O6	inlet end

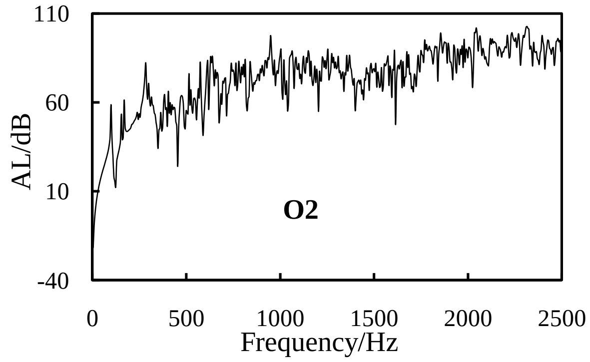
The transfer characteristics of each testing point of box structure machine leg to input load are shown in Figure 3. In order to facilitate comparison, the acceleration level (AL) after treatment is considered as evaluation index of vibration properties. The calculation formula for transforming acceleration value into acceleration level is:

$$L_{a_i} = 20\lg \frac{a_i}{a_0} \tag{7}$$

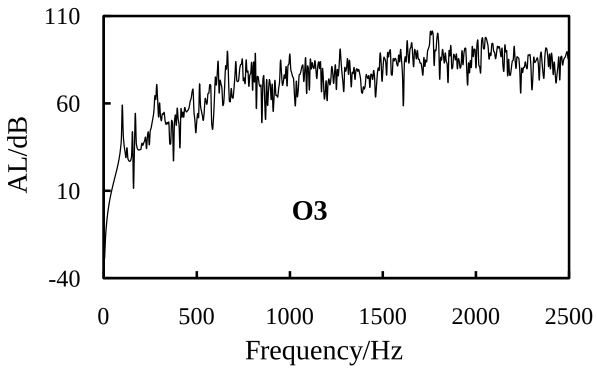
Among them, a_0 is the acceleration reference value, taking $a_0 = 1\times 10^{-6} m/s^2$.



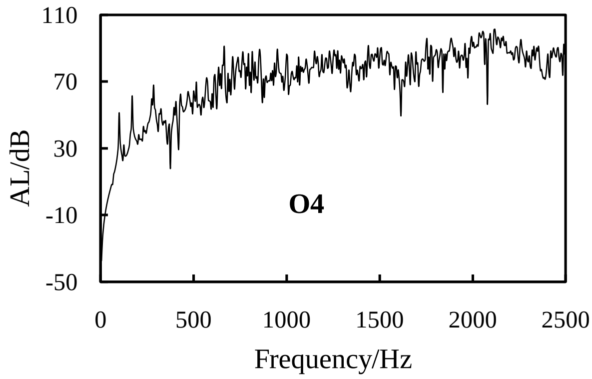
(a) Transfer function of measuring point 1



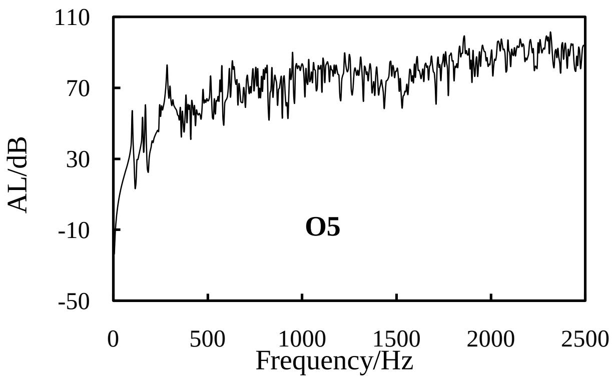
(b) Transfer function of measuring point 2



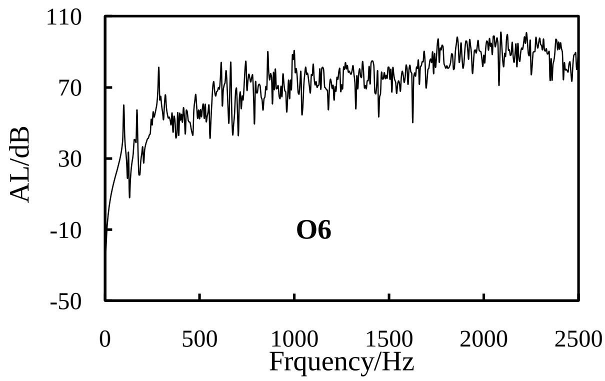
(c) Transfer function of measuring point 3



(d) Transfer function of measuring point 4



(e) Transfer function of measuring point 5



(f) Transfer function of measuring point 6

Figure 3 – The transfer function of each testing point

As shown in Figure 3, the tendency of acceleration frequency-response function for each testing point to the input load is basically the same. At low-frequency stage, with the increase of frequency, the acceleration response of the testing point will also increase with the advent of wave crest at the same time. The study has found that the corresponding frequencies of acceleration peak value are based on the natural frequencies of the box structure. Therefore, the response values are bigger and wave crests are more intensive at high-frequency stage, corresponding to the intensive box structure in high frequency mode. At high-frequency stage, the acceleration value of each testing point increases inconspicuously with the increase of excitation frequency. It is mainly because the high-frequency vibration is mainly embodied in the local vibration, rather than the overall vibration.

4. ACOUSTIC RADIATION CHARACTERISTICS ANALYSIS OF THE BOX

In order to analyze the box sound radiation characteristics, this paper first establish a BEM Fluid five meters away from the box, set the environment around the box as air; then establish a Generate Cube Recovery Date Mesh one meter outside plating layer, which is used to collect the radiation noise of the box at one meter; Finally fix up ten data collectors (Sensors) within each surface of the cube to collect the sound pressure value of each part at that point, and manually connect the BEM Fluid and the boundary element mesh of the subsystem, as shown in Figure 4.

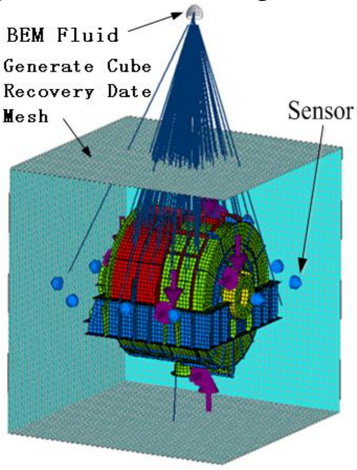
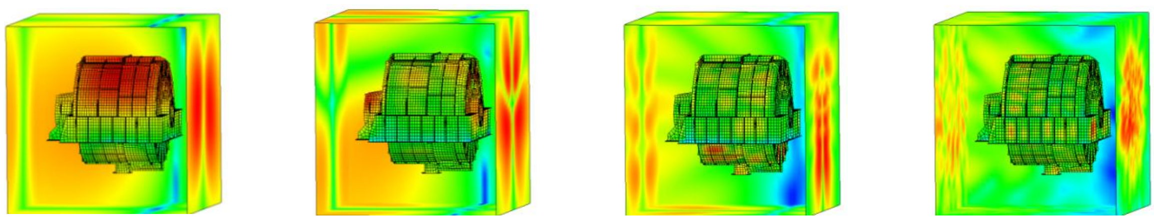


Figure 4 – BEM Fluid and the boundary element mesh of the subsystem

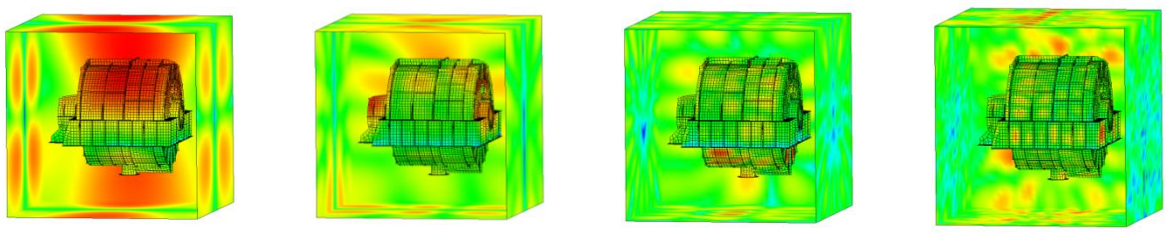
Considering the actual situation of box structure, the equivalent loss factor is 0.002. To facilitate comparison of the radiation noise of each part, and according to the characteristics of each part of the box, the whole box is divided into four parts, respectively: front end plate, upper box, lower box and back end plate. For the correct analysis on sound radiation characteristics of box structure, under the condition of without considering acoustic protection, noise prediction is carried out for the box. In order to provide theoretical support for the optimization for box noise reduction, the proportion of the radiation noise of each part accounting for the total noise is analyzed.

The sound pressure nephogram of the box's each part under typical excitation is shown in Figure 5.



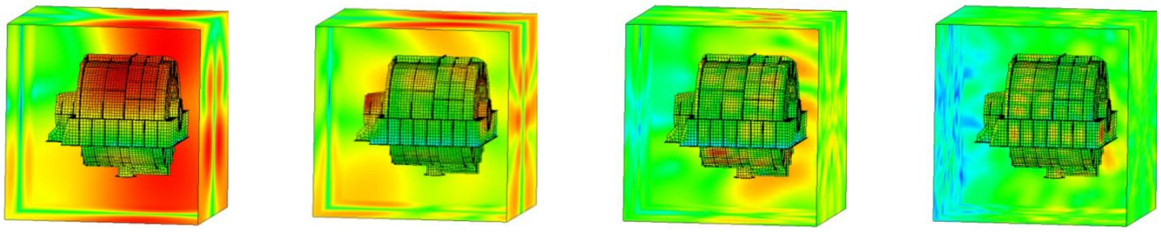
f=200Hz *f*=400Hz *f*=800Hz *f*=1600Hz

(a) The sound pressure nephogram of the front end plate under typical excitation



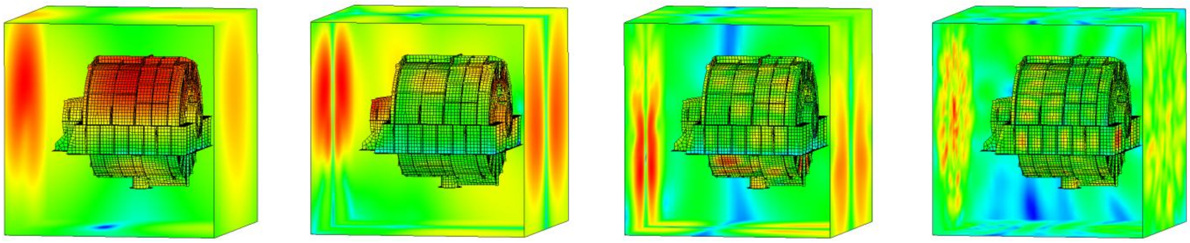
f=200Hz *f*=400Hz *f*=800Hz *f*=1600Hz

(b) The sound pressure nephogram of the under box under typical excitation



f=200Hz *f*=400Hz *f*=800Hz *f*=1600Hz

(c) The sound pressure nephogram of the upper box under typical excitation



f=200Hz *f*=400Hz *f*=800Hz *f*=1600Hz

(d) The sound pressure nephogram of the back end plate under typical excitation

Figure 5 – The sound pressure nephogram of the box's each part under typical excitation

After getting the sound pressure nephogram around the box, extract the sound pressure value at each measuring points, obtain each part's sound pressure level comparison chart under different frequencies, as shown in Figure 6.

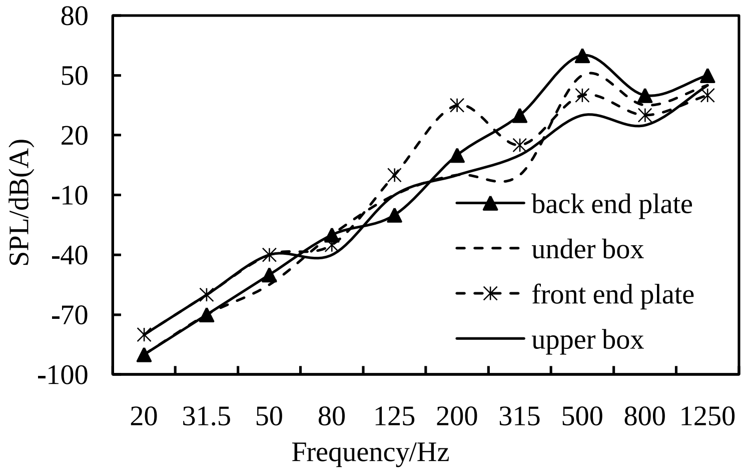


Figure 6 – Each part's sound pressure level comparison chart under different frequencies

It can be concluded that through Figure 5 and Figure 6: at low-frequency stage (less than 200 Hz), the radiated sound pressure values of the front end plate and upper box are relatively larger; at the middle and high frequency stage (more than 200 Hz), the radiated sound pressure values of back end plate and lower box increase gradually and surpass the values of the fore stand plate and upper box. Summarize the sound pressure level of each measuring point; further calculate the total sound pressure level of each part, as shown in Figure 7.

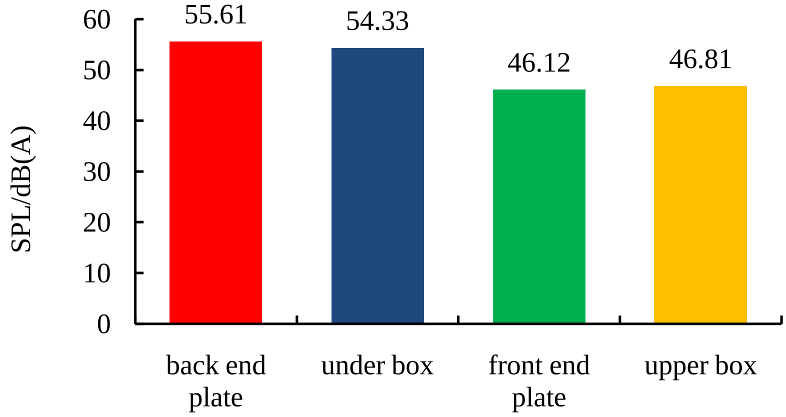


Figure 7 – The total sound pressure level of each part

As shown in Figure 7, each part’s total contribution amount of the radiated sound pressure values is: back end plate > lower box> front end plate > upper box. This shows that before acoustic protection, the radiated sound pressure values of back end plate and lower box are higher, which are the dominant components of the noise. Therefore, there is a need to focus on controlling them during noise reduction.

In order to improve the sound radiation characteristics of the box and reduce the vibration noise, acoustic protective methods can be employed for noise reduction processing; Firstly, lay damping layer on the outer wall of box; secondly, fix damping layer with sound insulating layer; finally, lay sound absorption layer outside the sound insulating layer. When the constrained damping structure is under bending deformation, the relatively soft damping layer stretches, deforms and causes shear deformation. Therefore, it can consume more vibration energy to achieve the goal of noise reduction. After laying damping layer structure, its dynamic characteristics change mainly reflects on the change of the structural loss factor. Therefore, under the condition of the fixed structure, the main means of vibration noise are to improve the compound loss factor of the box structure.

After acoustic protection treatment, the equivalent loss factor of constraint damping structure at calculating-frequency is 0.011. Each part’s sound pressure level comparison chart under different frequencies after acoustic protection is shown in Figure 8.

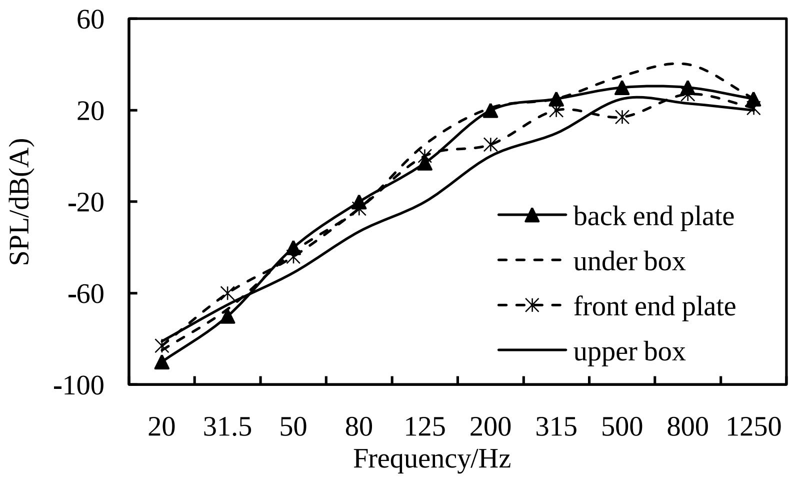


Figure 8 – Each part’s sound pressure level comparison after acoustic protection

Through Figure 8, at low-frequency stage, the radiated sound pressure values of front end plate and upper box are bigger than those of the other parts. At middle and high frequency stage, the radiated sound pressure values of the back end plate and lower plate begin to increase and even surpass those of the fore stand plate and upper box gradually. At the same time, we can also see that the radiation noise increases with the increase of frequency after acoustic protection, which is the same with the radiation noise regularity before acoustic protection. The total sound pressure level after acoustic protection is shown in Figure 9.

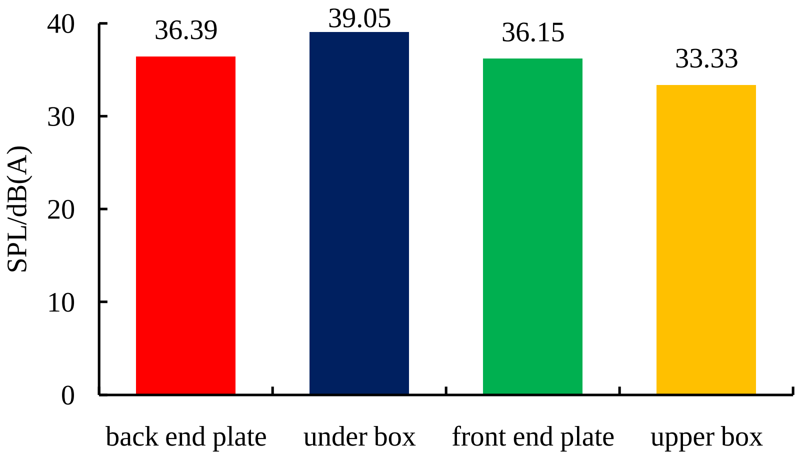


Figure 9 – The total sound pressure level after acoustic protection

Figure 9 shows that after acoustic protection, the total sound pressure level of the lower box surpasses that of the back end plate, which becomes the dominant component in this noise. Before and after the acoustic protection, the total radiated sound pressure level can reduce more than 5 dB, which meets the requirements of the vibration noise reduction design. Through the analysis above, the sound pressure level of box decreases with increment of its structural loss factor. Therefore, considering the feasibility and economic condition, we should try to increase its structure loss factor to achieve the best effect of optimization.

5. CONCLUSION

Based on FEM, this paper established reduction box finite element analysis model and carried on the numerical analysis to its vibration characteristic. By modal analysis, the natural frequency and vibration mode were found and the transfer characteristics on unit load were analyzed; on the basis of vibration characteristic analysis, the sound radiation analysis model was built based on boundary element method. Each part of the box's contribution to the radiation noise was analyzed. The noise reduction optimization measures of the box were studied and compared. The conclusions are as follow:

1. Under the condition of input unit load, there is no overall vibration but the local vibration in the box; the tendency of transfer function from input load to each machine leg is similar. The acceleration peak values are all the natural frequency of box structure, corresponding to the box in every modal.

2. On the basis of modal analysis, the change of the local thickness has little influence on the box's natural frequency and vibration mode and does not improve the dynamic characteristics of box; In order to avoid resonance, the frequency of input load in practical work should try to avoid its natural frequency.

3. Before acoustic treatment, the radiated sound pressure values of back end plate and lower box are higher, which are the dominant components of the noise; by the method of constrained damping layer and acoustic absorption for noise reduction processing increase box structure loss factor to 0.011, which has obvious optimization for box radiated noise. Considering economy, noise reduction effect and feasibility, the sound absorption and insulation layer should be thickened appropriately and the loss factor should be increased for the best noise reduction optimization measure.

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