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Optimal Design of Traction Gear Modification of High-Speed EMU Based on Radial Basis Function Neural Network

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ABSTRACT The dynamic characteristics of the traction gear transmission system have a great influence on the safety, comfort, and reliability of EMU (electric multiple units). Combining the methods of theoretical analysis, numerical simulation, and optimization design theory, establishing a parameterized gear modification model. Meanwhile, designing reasonable shape modification schemes and parameters. The dynamic characteristics, vibration response characteristics, and acoustic response characteristics of gear meshing of CRH380A high-speed EMU under continuous traction conditions are analyzed. The corresponding relationship between gear modification parameters and gear transmission radiation noise is approximated by finite element simulation data and RBF (radial basis function) neural network. Using a multi-island genetic algorithm to optimize gear modification parameters to minimize gear transmission noise, further seeking to meet the low-noise modification design of high-speed train traction helical gear transmission system under continuous operating conditions method.

INDEX TERMS Traction gear of EMU, RBF neural network, multi-island genetic algorithm, optimal design of modification.

I. INTRODUCTION

The traction transmission system is the core of the power source of high-speed EMU, located directly below the passenger vehicle. The vibration noise derived from gear alternation meshing in the high-speed cases, not only degrades system performance, affects the comfort of passengers, but also even endangers the safety of EMU trains running.

The traction gears transmission system of EMU is essentially a nonlinear multi-degree of freedom system in which time-varying parameters and transmission clearance coexist. Its structural noise is mainly dynamic response results from the joint effect of the internal excitation and external excitation in the gears meshing process. The internal excitation is the characteristic of gear transmission for distinguishing other mechanical systems, also is the source of the gear transmission system noise. Scholars at home and abroad have adopted various methods to modify the shape of gears to reduce gear running noise and vibration, which plays an important reference value for the design of gear modification of EMU. Yoon *et al.* [1] used a two-dimensional finite element method to estimate the roll angles for different loading conditions and proposed an optimal modification method from root to tip. To reduce the operating noise and vibration of spiral bevel gears, Mu *et al.* [2] proposed a novel method based on functional design. According to the designed higher-order transmission error (HTE) and contact path, it is obtained by correcting the conjugate tooth surface obtained from the paired gear Pinion target tooth surface. The tool parameters, initial machine settings, and polynomial coefficients that assist the tooth surface correction movement are used to establish a mathematical model for the pinion generator for the design parameters. An optimization model is developed to solve the polynomial coefficients. The results

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show that the meshing quality of HTE spirals bevel gears is better than gears designed using parabolic transmission error (PTE). Samani *et al.* [3] studied the vibration of nonlinear spiral bevel gears through an innovative tooth surface trimming method. To design a spiral bevel gear with HTE, a new method of nonlinear vibration was studied. This method is carried out by a PTE. The results show that the meshing quality of spiral bevel gears is better than that of ordinary gears. Nie *et al.* [4] took a certain type of gearbox as the research object, at the same time, used MASTA software to analyze the strength of the gear and modify the design. To improve tooth contact strength, flexural strength, and the tooth root flank anti-bonding capacity, reduce vibration transmission errors and the engagement gear tooth face load distribution optimized to reduce noise. Ni *et al.* [5] proposed a new parabolic correction design method based on the concept of bevel gear generation. The results show that the parabolic correction can reduce the maximum contact force and root stress of the pinion under the same load. Ramadani *et al.* [6] proposed a new method to reduce gear vibration and weight by changing the gear body structure. The main purpose is to reduce the vibration and noise emissions of spur gears. This method uses topological optimization software based on finite elements to configure and optimize the grid structure. The results prove that the honeycomb lattice structure and polymer matrix can significantly reduce the gear vibration. Tang *et al.* [7] took the traction helical gear of CRH380A as an example to analyze its meshing characteristics. A modification plan is proposed, and the comparison of gear contact stress distribution before and after modification shows that the modification plan can effectively reduce gear meshing impact and transmission noise. Wang *et al.* [8] established a rigid-flexible coupling dynamic model of wind turbine gearboxes, used a mesh model based on strips to represent the meshing of gear pairs. The vibration acceleration amplitude and structural noise of the wind turbine gearbox are reduced through proper tooth shape correction. Fan *et al.* [9] took a high-speed EMU transmission gear system as an example and used KISSsoft to calculate the deformation of the gear support shaft during the transmission process. The influence of shafting deformation on the meshing performance of the gear pair is compensated by the correction method. The contact spots, load distribution of the tooth line, and transmission error of the gear pair before and after modification is compared to ensuring that the transmission system can work more smoothly and reliably. Tang *et al.* [10] predicted radiation noise based on an acoustic radiation analysis model constructed by acoustic BEM and Helmholtz boundary integral equations. The research results can provide a theoretical basis for the optimization design of the low noise gear in EMU. Li *et al.* [11] used marine gearbox as the research object, established a planetary gear transmission model of the gearbox using Romax as well as studied gear modification. According to the transmission error, load distribution per unit length, and flash temperature as the modification amount, through the comparative analysis of the gear transmission performance before and after the transformation optimization, the reliability and service life of the optimized planetary gear transmission are improved, the vibration and noise problems are also improved. Xie *et al.* [12] used RomaxDesign software, to carry out transmission error analysis, gear contact analysis, tooth root stress analysis, and optimization of gears before and after modification based on gear modification theory. The results show that the gear transmission will be smoother, the noise will be smaller, as well as the tooth root stress of the gear will be reduced after optimization.

In the design of gear modification, most gear manufacturers still use traditional empirical formulas or static elastic deformation of gear teeth to estimate the parameters of gear modification, the accuracy is relatively low. However, some scholars have optimized the gear modification parameters through intelligent optimization algorithms and achieved good results. Zhang *et al.* [13] used the variation on planetary gear vibration acceleration as the objective function and optimized the amount of gear modification using a genetic algorithm. Zhu [14] used the KISSsoft software to limit and screen the boundary conditions of the design scheme when designing the gear parameters of an EMU gearbox and then selected the best strength scheme from them. The transmission performance indexes such as gear strength and vibration are improved by the modification contact analysis. Jia *et al.* [15] proposed a helical gear tooth surface modification design method, combined with load contact analysis (LTCA) and intelligent optimization algorithm to establish an optimization model, with the load transmission error amplitude as the optimization goal to obtain the final tooth surface modification shape parameters. Xiong *et al.* [16] adopted comprehensive modification measures combining tooth-shaped drum shape modification and tooth-angled slope shape modification. A genetic algorithm was selected to optimize the comprehensive modification of tooth direction based on Romax Designer software. Finally, the corresponding optimal modification scheme is obtained. Yang *et al.* [17] proposed a gear machining optimization algorithm based on the Kriging model and genetic algorithm for the problems of a large amount of calculation, low accuracy, and complicated operation in the gear machining optimization process. Taking straight spur gear transmission as the research object, the modification of gears is optimized.

Traditional empirical formulas and other estimation methods are difficult to meet the high-speed and high-precision design requirements of high-speed EMU traction gear transmission. Using a computer simulation method based on finite element analysis to replace the real experiment, constructing the gear modification parameter optimization model and solving it through the optimization algorithm is an effective way to realize the optimal design of gear modification parameters. However, most of the existing research focuses on gear vibration acceleration or dynamic transmission error as the optimization goal, and there are few optimization methods to reduce gear transmission noise as the direct goal.

The main reason is the direct relationship between the gear modification parameters and the radiation noise size. The mapping relationship is difficult to establish; the second is that the scope of the study is relatively complex, the system dynamics must be extended to the boundary element sound field response to the system. Furthermore, the two different properties of dynamics and acoustics should be optimized in steps. Combining the methods of theoretical analysis, numerical simulation, and optimization design theory, establishing a parameterized gear modification model. Meanwhile, designing reasonable shape modification schemes and parameters. The dynamic characteristics, vibration response characteristics, and acoustic response characteristics of gear meshing of CRH380A high-speed EMU under continuous traction conditions are analyzed. The corresponding relationship between gear modification parameters and gear transmission radiation noise is approximated by finite element simulation data and RBF neural network. Using a multi-island genetic algorithm to optimize gear modification parameters to minimize gear transmission noise, further seeking to meet the low-noise modification design of high-speed train traction helical gear transmission system under continuous operating conditions method.

II. PARAMETRIC DESIGN OF MODIFIED GEAR

Gear parameterization modelling has a mature design flow. The parametric solid model of the helical gear transmission system was constructed using PRO / E software, based on the constraint relationship between the geometric parameters of the traction gear set of the high-speed EMU and the involute parameter equation in the Cartesian coordinates system. Combining with the involute equation, the rapid parameterized redesign of the modification gear is realized by the selection of modification methods and the parameter design of tooth profile modification.

A. 3D MODEL CONSTRUCTION OF GEAR PAIR

The basic geometric parameters of traction gear pair in high-speed train CRH380A are shown in Table 1.

Define the relational expressions of the basic parameters in Table 1, drawing the basic circle system and a spiral of the gear under the driving of the dimensions of these relational expressions. According to the parametric equation of the involute curve in the Cartesian coordinate system, the involute curve of the gear is created. The generated involute mirror images and the basic circle system of the gear form the basic outline of the gear teeth, then the gear model is generated through the characteristic operations of mixed scanning and array. Since we mainly study the noise of the gear pair transmission system, to reduce the workload of the simulation calculation and ignore the influence of the transmission shaft, bearing and other components, only the simplified model of the gear pair is taken as the research object.

TABLE 1. Basic geometric parameters of gear pair.

Name	Driving gear	Driven gear
Teeth z	29	69
Modulus m_n(mm)	7	7
Pressure angle α_n(°)	26	26
Helix angle β(°)	20	20
Tooth face width B	70	70
The coefficient of addendum h_a*	1	1
The coefficient of tip gap c*	0.25	0.25
Modification coefficient X_n	0	-0.284588

B. THE CHOICE OF MODIFICATION PLAN

The methods of tooth profile modification include tooth tip modification, tooth root modification, and full tooth profile modification of the tooth crown and root simultaneously. Because the modification of the tooth root easily affects the strength of the gear pair, thereby reducing the safety and reliability of the EMU. Besides, from a geometric point of view, the effect of modifying the tooth root is equivalent to the modification of the tooth tip of the meshing tooth, so it is often chosen to modify the tooth tip only. At the same time, considering the technical difficulty, cost, and effect of the gear modification of EMU, only the driving gear can be modified without the driven gear. Therefore, only the top modification of driving gear is selected as the modification scheme.

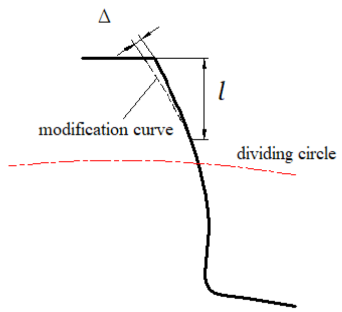
C. PARAMETER DESIGN OF TOOTH PROFILE MODIFICATION

The tooth profile modification design mainly includes three parameters: modification amount Δ, modification length l and modification curve. For the modification amount, it is usually determined according to the elastic deformation of the gear teeth under the rated torque. Using ANSYS Workbench software to conduct static contact analysis on the gear pair, the maximum deformation of the tooth profile of the driving gear is 0.02mm. In order to explore the optimal shape modification parameters, the range of the amount of modification is set to 0.001mm-0.02mm. Recommended by Soviet Cylindrical Gear Standar $l = 0.45mn$(mn is the modulus of the gear). Alecstokes recommends $l = 0.5mn$, ISO standard recommends $l = 0.5mn$.

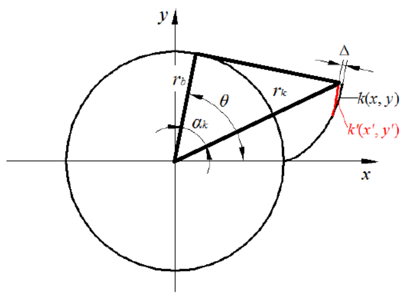
Therefore, the modification length $l = 0.5mn$ is selected as the lower limit of the modification length, and $l = 1mn$ as the upper limit of the modification length, the corresponding modification length is 3.5mm-7mm. Regarding the modification curve, existing research shows that the use of Walker curves to modify the tooth profile can effectively avoid the occurrence of sudden stress changes. So the Walker curve is used to modify the shape.

D. INVOLUTE EQUATION OF MODIFIED GEAR

As shown in Fig. 1, k(x,y) is the point on the gear involute line, k'(x′,y′) is the corresponding point on the modification



(a) The schematic diagram of tooth profile modification.



(b) The schematic diagram of gear Involute.

FIGURE 1. The schematic diagram of involute modification.

curve, the amount of modification is Δ. The rolling angle and pressure angles corresponding to point k are θ and α_{k}, respectively. Then the conversion relationship between k point and k' point coordinates is as formula (1).

$$\begin{cases} x' = x - \dfrac{\Delta}{\sin \alpha_k} \\ y' = y + \dfrac{\Delta}{\cos \alpha_k} \end{cases} \tag{1}$$

The involute equation of the unmodified gear is defined as:

$$\begin{cases} x = r_b \cos(\theta) + r_b\theta \sin(\theta) \\ y = r_b \sin(\theta) - r_b\theta \cos(\theta) \end{cases} \tag{2}$$

The involute equation of the gear modified by Walker curve is expressed as:

$$\begin{cases} x' = r_b \cos(\theta) + r_b\theta \sin(\theta) - \dfrac{\Delta}{\sin\theta} \\ \quad \times (1 - \dfrac{r_b \times \theta_{\max}}{l} + \dfrac{r_b \times \theta}{l})^{1.5} \\ y' = r_b \sin(\theta) - r_b\theta \cos(\theta) + \dfrac{\Delta}{\cos\theta} \\ \quad \times (1 - \dfrac{r_b \times \theta_{\max}}{l} + \dfrac{r_b \times \theta}{l})^{1.5} \end{cases} \tag{3}$$

Here: Δ is the shape modification amount and l is the shape modification length. When the rolling angle $\theta = \theta_{\max}$, the radius of the involute at this point is the radius of the addendum circle, Therefore, $\theta_{\max} = \frac{\sqrt{r_a^2 - r_b^2}}{r_b}$, r_a is the radius of the addendum circle.

In the pinion model constructed by PRO / E parameterization, add the amount of modification Δ and the length of modification l to the basic parameters of the gear. Then use (3) to replace the involute equation of the gear. By inputting different modification amount Δ and modification length l, the parametric modifier design of the gear can be carried out quickly.

III. SAMPLE SELECTION BASED ON OPTIMAL LATIN HYPERCUBE

The optimal Latin hypercube is used to realize the sampling selection within the range of gear modification parameters. Latin hypercube sampling is the latest development of sampling technology and can be used for stratified sampling of multidimensional sample space. Its basic working principle is as follows:

(1) Determine the number of samples to be drawn is N;

(2) Divide each input into N columns with equal probability, $\mathrm{x}_{i0} < \mathrm{x}_{i1} < \mathrm{x}_{i2} < \cdots < \mathrm{x}_{in} < \cdots < \mathrm{x}_{iN}$, and $p(\mathrm{x}_{in} < x < \mathrm{x}_{i(n+1)}) = \frac{1}{N}$;

(3) Only one sample is taken for each column and they are combined with a vector. The position of the drawn sample of the column is random.

Through the DOE module of ISIGHT software, according to the preliminary calculation of the modification parameters of Section II-C, the range of shape modification amount is set to 0.001mm-0.02mm, and the value of the modification length is 3.5mm-7mm, run the optimal Latin hypercube sampling method to generate 35 sets of parameter values. Fig. 2 shows the distribution of the generated modification parameter combinations in the sample space. We can see that the data selected by the optimal Latin super-stand method can be evenly distributed over the sample space.

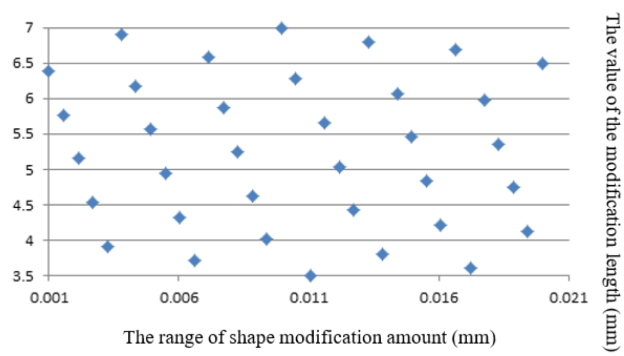


FIGURE 2. Sample data distribution.

IV. PREDICTION MODEL OF GEAR NOISE BASED ON RADIAL BASIS FUNCTION NEURAL NETWORK

A parametrized gear pair model is generated by driving the combination of modifier parameters extracted from the optimal Latin hypercube. According to the flow of dynamic meshing force analysis, modal analysis, harmonic response analysis, and acoustic boundary element analysis. The magnitude of radiated noise is multiple sets of data output. The mapping relationship between the modification parameter combination and the gear transmission noise is established by the RBF neural network training, thereby establishing the prediction model of the gear transmission noise.

A. ANALYSIS OF DYNAMIC MESHING FORCE OF GEARS

The constraint conditions set that the main and driven gears are only allowed to rotate. The speed and torque of the gear drive system of the CRH380A high-speed EMU under continuous operating conditions are 434rad/s and 841.5N·m, respectively. The set simulation model is shown in Fig. 3.

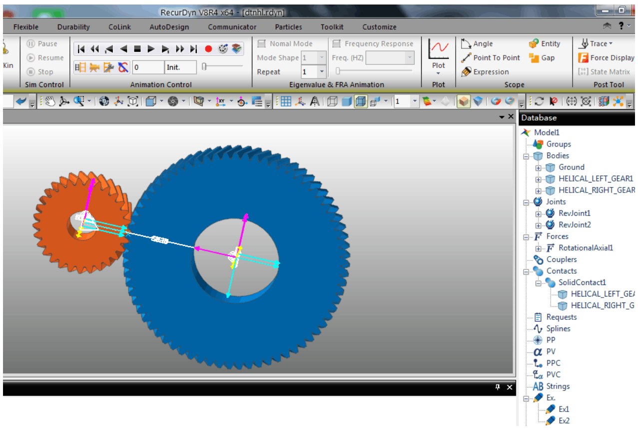


FIGURE 3. Dynamic simulation model of gear pair.

The meshing impact force (dynamic excitation force) generated by gear teeth during gear transmission is the main cause of gear vibration and noise. The multi-body dynamics analysis software RecurDyn is used to analyze and solve the gear transmission meshing force characteristics.

Due to the dynamic meshing force of the transmission gear being mainly caused by the meshing rigidity of the gear, according to Newton's law, the dynamic meshing force can be expressed as:

$$\begin{aligned} F(t) &= k(t)f(R_p\theta_p - R_g\theta_g - e(t)) \\ &= (T_p - I_p\ddot{\theta}_p + R_p c_m(R_p\dot{\theta}_p - R_g\dot{\theta}_g - e(\dot{t})))/R_p \end{aligned} \tag{4}$$

The built model of gear pair in PRO/E is imported into software RecurDyn and the dynamic meshing force of gear pair can be solved. The solution mode is selected as dynamics. The time-domain of the solution is set as 0s-5s, the time domain step is set as 0.0001s, and the frequency domain is set as 0Hz-3500Hz. According to (4), the dynamic meshing force is solved by the nonlinear finite element simulation analysis.

Considering the steady-state process of the system, the time range of the time domain diagram is selected as 3s-4s (Fig. 4). The frequency-domain diagram of meshing to force is plotted by FFT (fast Fourier transform) (Fig. 5).

B. MODAL ANALYSIS OF GEAR TRANSMISSION SYSTEM

The vibration characteristics of the mechanism caused by external force have relation not only to external excitation but also to its natural frequency and natural mode of vibrating. Modal analysis can solve the natural frequency of each order of the transmission gear pair in a certain frequency range as well as its vibration mode characteristics. On this basis, the vibration response of the gear pair under various excitations in a certain frequency range can be predicted.

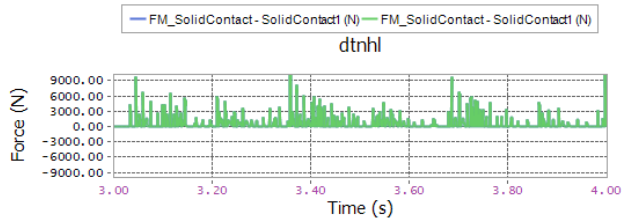


FIGURE 4. Time domain diagram of meshing force.

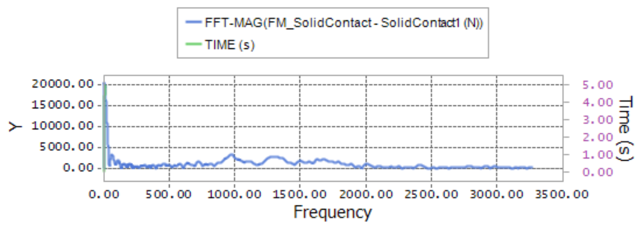


FIGURE 5. Frequency domain diagram of meshing force.

According to the dynamic equation of the undamped free vibration system in modal theory, by Fourier transformation, it can be obtained as follows:

$$[K]\{\Phi i\} = \omega i^2[M]\{\Phi i\} \tag{5}$$

Here: M is systematic mass matrix, K is stiffness matrix, Φi is time-invariant eigenvector. The eigenvalue ωi is systemic natural frequency.

The modal analysis of the system is realized by the ANSYS Workbench module. Select the mode order to be solved and set as the first 6 modes of the gear pair. According to experience, when the gear pair under the impact of the dynamic load vibration generation, usually in a previous order modal resonate more dangerous, serious damage to the gear dynamic performance, and in the high order frequency due to the complexity of the effect of mutual coupling between frequency, frequency and intensity are very serious, it is difficult to accurately identify the modal parameters, combined with little impact on the dynamic performance of box body is neglected modes, therefore, extract only the first six order modal analysis of gear pair. The finite element structures were solved by (5), and the results are shown in Table 2. The first vibrating model is shown in Fig. 6, the others are omitted.

C. HARMONIC RESPONSE ANAIYSIS OF GEAR TRANSMISSION SYSTEM

Because the dynamic meshing force of the gear pair is similar to the sine wave law, the harmonic response of the transmission gear pair should also be analyzed to analyze the vibration

TABLE 2. The first six order natural frequency of gear pair.

Order	1	2	3	4	5	6
Frequency (Hz)	507.34	769.1	882.65	1457.6	2186.7	2603

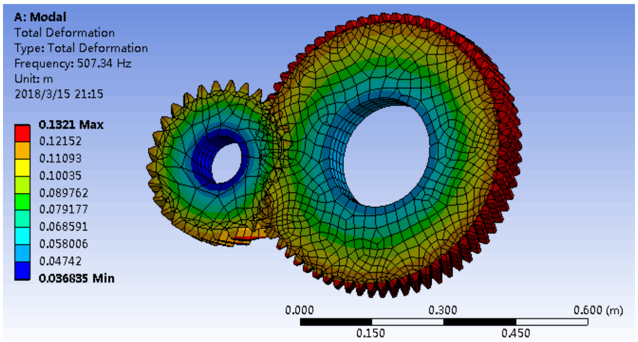


FIGURE 6. The first order natural mode of vibration of gear pair.

response of the gear transmission system under the excitation of meshing impulse. The response values of harmonic response analysis include vibration displacement, vibration speed, and vibration acceleration. The analysis should focus on observing the peaks in the frequency response curve and the frequencies corresponding to the peaks to guide the designer to avoid fatigue and resonance. The forced vibration equation of the undamped system is:

$$[M]\{\mathrm{a}\} + [K]\{\mathrm{s}\} = \{\mathrm{F}\} \tag{6}$$

Here: s is vibration displacement, a is vibration acceleration.

The excitation load of harmonic response can be expressed as:

$$\mathrm{F_i} = \mathrm{F_{i\,max}} \sin(\omega t + \theta_i) \tag{7}$$

Here: $\mathrm{F_{i\,max}}$ is load amplitude, ω is the load frequency, θ is load phase angle.

The structural harmonic response can be analysed by using the workbench. According to (6), the vibration response characteristics are calculated and the frequency response curve is output (Fig. 7).

D. NUMERICAL ANALYSIS OF GEAR PAIR

To solve the acoustic radiation problem with an unbounded sound field, if the whole model is meshed by the finite element method, the calculation is often too large to complete the solution. However, by the boundary element method of Helmholtz boundary integral equation, only the surface mesh of the structure is extracted to complete the acoustic radiation calculation. The expression is as follows:

$$\mathrm{p} = [A_v(w)]^T v_n(w) \tag{8}$$

Here: p is sound pressure, $A_v(w)$ is acoustic transfer vector, $v_n(w)$ is the normal velocity of vibration on structural surface unit.

Then, the normal velocity v_n and sound pressures p at any point with a distance of r to the origin in the acoustic radiation field of the gear transmission system are satisfied as:

$$p(r_a) = \sum_{i=1}^{n_e} N_i^e(r_a) \cdot a_{pi}, \quad r_a \in \Omega_{ae} \tag{9}$$

$$v_n(r_a) = \sum_{i=1}^{n_e} N_i^e(r_a) \cdot a_{vi}, \quad r_a \in \Omega_{ae} \tag{10}$$

Here: v_n is the number of nodes on the Ω_{ae} unit, a_{pi} is sound pressure on the boundary element nodes, a_{vi} is normal velocity on the unit nodes, N_i^e is shape function of unit.

Due to the wide range of human hearing, it is inconvenient to directly use sound intensity or sound pressure to indicate the strength of sound. For comparison purposes, sound pressure is measured by a logarithmic scale. The acoustic pressure level is defined as 20 times the logarithm of the ratio of the effective sound pressure to the reference acoustic pressure, i.e.,

$$L_P = 20log\frac{P_e}{P_r} \tag{11}$$

Here: $\mathrm{P_e}$ is the measured sound pressure, $\mathrm{P_r}$ is the reference acoustic pressure, and usually $\mathrm{P_r} = 2 \times 10^{-5}Pa$, it is the lowest sound pressure value that can be perceived 1 kHz sound in the air by human ears.

The acoustic pressure level is a non-dimensional number, and it is generally in decibels (dB) for easy use and good understanding.

The obtained results of the vibration response from the analysis in the first 3 steps are imported into the noise vibration analysis software LMS, after that the noise can be simulated and predicted [18].

In Virtual.Lab, acoustic boundary element method is chosen. According to (9), the acoustic pressure level nephogram of the radiation noise for gear transmission of the ISO standard spherical sound field is obtained as Fig. 8.

The distance between each point on the ISO spherical sound fields and the center of the gear pair is about 1m. Considering that the passenger positions are directly above the gear pair, select the appropriate 4 points as marked in Fig. 9 on the sound field, and check the acoustic pressure level at these points.

The RMS value represents the root mean square of the acoustic pressure level corresponding to the whole scanning frequency at four selected points, which is the noise level heard by the human ear at these four positions. It can be seen that detecting sound has a difference in orientation, thus the level of radiated noise will not be judged by the noise at a specific point.

Further, the sound power level will be solved for reflecting the radiated energy of the sound source per unit time. Its expression is shown as follows:

$$W = \xi c_0 S \tag{12}$$

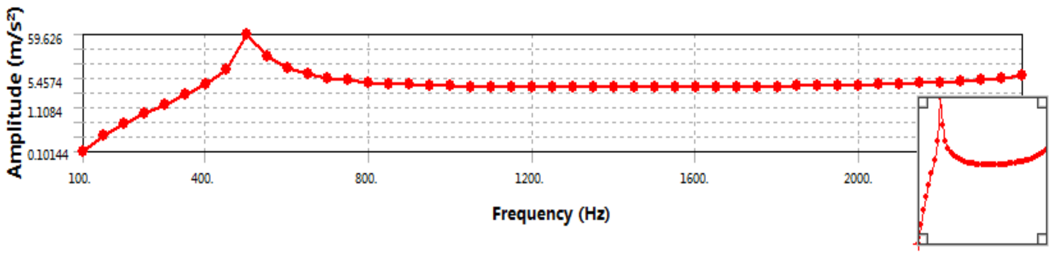


FIGURE 7. Frequency response curve of gear pair.

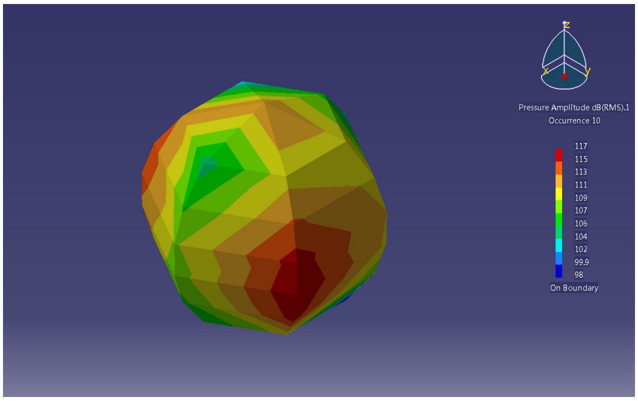


FIGURE 8. Acoustic pressure level nephogram.

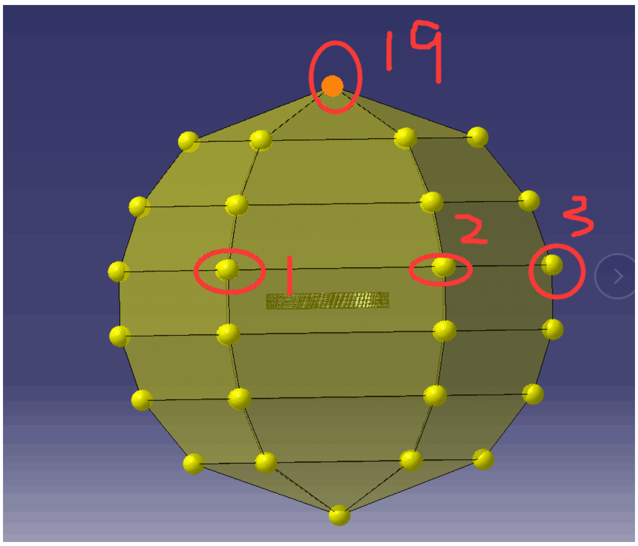


FIGURE 9. Schematic diagram of selected points.

Here: c_0 is the speed of sound propagation, S is the area of acoustic diffusion direction, ξ is sound energy density, and can be calculated as (13):

$$\xi = \frac{E}{V_0} = \frac{1}{2}\rho_0\left(v^2 + \frac{1}{2\rho_0 c_0^2}p^2\right) \tag{13}$$

Also for comparative purposes, the acoustic power level is defined as 10 times the logarithm of the ratio of the acoustic power to the baseline value, i.e.,

$$L_\omega = 10log\frac{W}{W_0} \tag{14}$$

Here: the baseline value of acoustical power W_0 is 10^{-12} W.

Through the acoustic pressure level curve (Fig. 8), according to (12) to (14), the sound power level curve and its root mean square (RMS) can be obtained. The noise simulation value of each point is shown in Table 3.

TABLE 3. Noise simulation table.

	Point 1	Point 2	Point 3	Point 19
Acoustic pressure level（dB）	92.54	99.46	94.76	96.2
Sound power level RMS（dB）	114.44			

E. BUILDING NOISE PREDICTION MODEL OF GEAR TRANSMISSION SYSTEM

The sampled 35 sets of modification amount and modification length parameter values were substituted into the PRO/E parameterized modification model to generate 35 modified gear transmission models. These models are simulated according to the simulation process of solving dynamic meshing force analysis, modal analysis, harmonic response analysis, and acoustic boundary element analysis.

The noise data obtained from the simulation are collected to form the corresponding relationship between the modification parameters and the noise, as shown in Table 4.

The first 25 sets of data in the table are selected for RBF neural network training, and the remaining 10 sets are used for comparison with the prediction results of the RBF neural network.

Compared with the general neural network, the running speed of the RBF neural network has a great advantage. The reason lies in the difference between the operation principle of the middle neuron and the general neural network.

The main steps to build a noise prediction model based on RBF neural network using Isight software are as follows:

(1) Call the Approximation module in the Isight software, select the RBF neural network, read the first 25 sets of data in the above table, set the modification amount and modification

TABLE 4. Table of correspondence between modification parameters and noise.

No	Modification amount Δ (mm)	Modification length l (mm)	Sound power level RMS (dB)
1	0.01882	4.01	115.62
2	0.01941	5.25	109.31
3	0.01412	6.38	112.27
4	0.01529	7	104.65
5	0.00259	4.12	106.86
6	0.01471	4.53	106.94
7	0.01588	5.15	108.79
8	0.00529	4.84	102.67
9	0.01706	5.76	110.53
10	0.01235	5.04	110.61
11	0.00765	5.87	112.55
12	0.01	5.56	109.69
13	0.01059	6.28	107.76
14	0.00235	3.5	109.31
15	0.00647	5.46	109.68
16	0.00528	6.49	110.19
17	0.01765	6.59	108.81
18	0.01294	3.91	104.74
19	0.00571	6.69	108.84
20	0.01118	4.43	110.99
21	0.00941	3.81	108.03
22	0.01176	6.9	105.82
23	0.01647	3.6	113.21
24	0.00882	4.94	103
25	0.00588	3.71	112.35
26	0.00393	5.57	118.91
27	0.00334	5.35	104.75
28	0.01824	4.63	119.29
29	0.00824	6.79	107.08
30	0.00412	4.22	120.81
31	0.00706	6.18	111.01
32	0.00176	4.74	103.31
33	0.0185	6.07	109.74
34	0.01353	5.66	111.3
35	0.00765	4.32	105.65

length as input parameters, and the sound power level RMS as an output parameter.

(2) Set the smoothing filter value to control the smoothness of the fitted surface to eliminate noise data, and set the maximum number of iterations for each response to fit the RBF model. To ensure the accuracy of model training, set the value of the smoothing filter to 0.05 and the maximum number of initialization iterations to 50.

(3) Set the error analysis options and select the cross-validation method.

(4) After all the settings are completed, the prediction model is initialized and built. The running results of the model are as follows:

Through the above RBF neural network training, the mapping relationship between the transmission gears pair modification parameters and the transmission noise size is constructed:

$$Z = f(\Delta, l) \tag{15}$$

Here: Z represents the root mean square value of the sound power level of the scanning frequency band corresponding to the radiated noise of the gear transmission system, Δ represents the amount of modification and l represents the modification length.

Input the modification parameter values of the last ten groups of data for comparison into the RBF neural network prediction model, and compare the output data with the simulation data, as shown in Table 5.

TABLE 5. Comparison of acoustic simulation values and predicted values.

No	Δ (mm)	l (mm)	Sound power level RMS simulation value (dB)	Sound power level RMS prediction value (dB)	Prediction error
26	0.00393	5.57	118.91	118.04	-0.732%
27	0.00334	5.35	104.75	104.99	0.229%
28	0.01824	4.63	119.29	120.03	0.620%
29	0.00824	6.79	107.08	106.55	-0.495%
30	0.00412	4.22	120.81	121.25	0.364%
31	0.00706	6.18	111.01	111.5	0.441%
32	0.00176	4.74	103.31	103.71	0.387%
33	0.0185	6.07	109.74	110.12	0.346%
34	0.01353	5.66	111.3	110.21	-0.979%
35	0.00765	4.32	105.65	106.55	0.852%

We can see that for the same modification amount and modification length, the error between the output of the noise prediction model based on the RBF neural network and the simulation output is less than 1%, indicating that the constructed prediction model can more accurately achieve noise size prediction.

V. MODEL OPTIMIZATION BASED ON MULTI-ISLAND GENETIC ALGORITHM

The standard genetic algorithm may not be able to converge to the global optimal value due to the local optimal solution when solving the optimization problem. The multi-island genetic algorithm [19] is used to optimize the solution of the RBF neural network modification parameter noise prediction model. This algorithm can ensure the diversity of genetic populations through migration and essence retention, and help the algorithm get out of the local optimal solution. Avoid premature convergence and ensure a globally optimal solution.

According to the RBF neural network noise prediction model, a noise reduction optimization model is constructed:

$$\min\ Z = f(\Delta, l)$$
$$s.t.\ \begin{cases} 0.001 \le \Delta \le 0.02 \\ 3.5 \le l \le 7 \end{cases} \tag{16}$$

Here: Z represents the root mean square value of the sound power level of the scanning frequency band corresponding to the radiated noise of the gear transmission system, Δ represents the amount of modification, and l represents the modification length, the constraint is the value range of the shape modification parameters.

The multi-island genetic algorithm is introduced to solve the noise reduction optimization model. In the Isight software, it is very convenient to call the optimization module to optimize the prediction model of the RBF neural network.

Set the number of islands in the multi-island genetic algorithm to 10, the size of the subgroup to 10, the genetic algebra to 100, the crossover rate to 0.9, the mutation rate to 0.01, the migration rate to 0.1, the migration interval to 2, and the number of elites per immigration to 1, the parameter of roulette size selected by roulette accounts for 0.5.

Set the upper and lower limits of the input variable and the binary code length, select the 16-bit code length, the search span of each input variable can reach 65535, to ensure sufficient search accuracy. The goal of sett optimization is to minimize the RMS value of the sound power level.

The multi-island genetic algorithm is an efficient and parallel intelligent global search algorithm, which can automatically acquire and accumulate information in the search space during the search process, and adjust the search method based on this to obtain the optimal output. After all options are set, running the multi-island genetic algorithm to solve the optimization model. Isight will record the entire search process in the form of a number table and chart. The optimal solution of the modification parameter combination in this example is shown in Table 6.

TABLE 6. The optimal solution of modification parameter combination.

Modification amount Δ (mm)	Modification length l (mm)	Sound power level RMS prediction value (dB)
0.0051308	4.8296	102.64

VI. EFFECT ANALYSIS OF GEAR SHAPE MODIFICATION DESIGN

Regenerating the transmission gear pair model with the obtained excellent repair parameters, besides analyzing it according to the simulation process of dynamic meshing

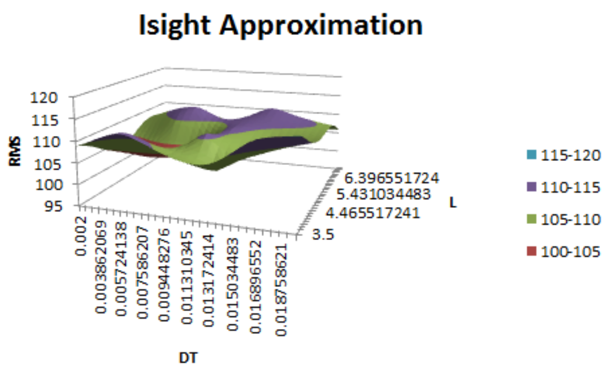


FIGURE 10. RBF neural network prediction model.

force analysis, modal analysis, harmonic response analysis, and acoustic boundary element analysis, sound power level simulation results are shown in Fig. 11.

The sound power level obtained by the simulation analysis is 102.52dB, which is very close to the prediction results in Table 6, and the effectiveness of the optimization results of the noise prediction model is tested.

To facilitate comparison, 4 points at the same position are selected on the ISO standard spherical sound field of the gear pair noise simulation after modification, then the sound pressure level curve is observed. As shown in Fig. 12. The results of sound power level and sound pressure level of the gear pair before and after modification are compared as shown in Table 7.

TABLE 7. Comparison of acoustic pressure level and sound power level at various points before and after modification.

Comparison item	Before modification (dB)	After modification (dB)	Variation (dB)	Rate of change
Acoustic pressure level (point 1)	92.54	83.55	-8.99	-9.71%
Acoustic pressure level (point 2)	99.46	85.36	-14.1	-14.18%
Acoustic pressure level (point 3)	94.76	84.83	-9.93	-10.48%
Acoustic pressure level (point 19)	96.2	82.54	-13.66	-14.20%
Sound power level	114.44	102.52	-11.92	-10.42%

It can be seen from Table 7 that the noise pressure level of the modified gear transmission system at different positions of the ISO standard spherical sound fields is reduced by about 9-14dB, and the sound power level is reduced by 11.92dB. Considering that the simulation environment designed in this paper is different from the actual situation. In the simulation environment, gear vibration noise is directly transmitted through the air, there are no sound insulation and noise reduction measures in the transmission path from the gear pair to the taken point. The amount of noise heard inside should be

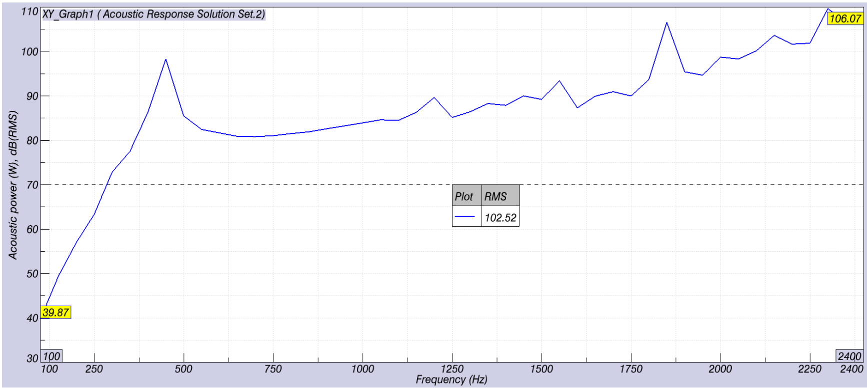


FIGURE 11. Sound power level curve after modification.

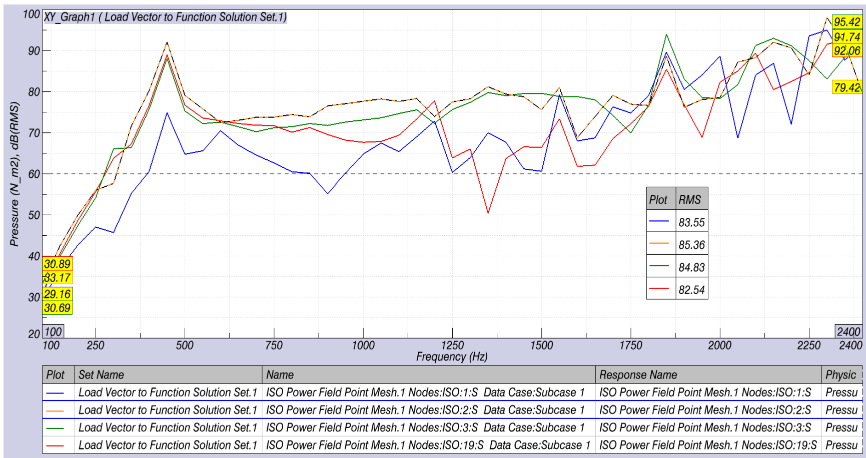


FIGURE 12. Acoustic pressure level curve of each point after modification.

lower. Therefore, selecting the optimal modification parameters can effectively reduce the noise of gear pair transmission and improve the comfort of passengers.

VII. CONCLUSION

The parametric modification model of a gear transmission system is built by Pro / E. Through the analysis process of dynamic meshing force analysis, modal analysis, harmonic response analysis, and acoustic boundary element analysis, the radiated noise of gear pairs with different modification parameters is obtained. These simulation data are trained to construct a transmission gear noise prediction model based on RBF neural network. Taking the reduction of gear transmission noise as the goal, the multi-island genetic algorithm is introduced to solve the model, and the optimal modification parameter combination is obtained, which is verified by comparison with the simulation results. The main conclusions are as follows:

(1) The parametric design of the repair gear was carried out based on the parametric model of traction gear pair for high-speed EMU, including the selection of the repair mode, the parameter design of the tooth profile repair, and the involute equation of the repair gear. In the pinion model constructed by PRO / E parameterization, the involute equation of the modified gear is used to add the modification amount Δ and the modification length l to the basic parameters of the gear. By inputting different modification amount Δ and modification length l, we can quickly parameterize the modification design of the gear.

(2) Through the analysis of the noise of the gear transmission system, this process includes the analysis of the dynamic meshing force of the gear, the modal analysis of the gear transmission system, the analysis of the harmonic response of the gear transmission system, and the numerical analysis of the noise of the gear pair. It provides a simulation data source for noise prediction of the gear transmission system. Running optimal Latin hypercube sampling to generate several sets of parameter values, meanwhile, using Isight software to build a noise prediction model based on RBF neural network. The results show that the output error of the noise prediction model and simulation output based on the RBF neural network is less than 1% for the same modification amount and modification length, which shows that the prediction model can accurately predict the noise size.

(3) The multi-island genetic algorithm is used to optimize and solve the RBF neural network modification parameter noise prediction model. Furthermore, the optimal modification parameters are verified by simulation. The acoustic pressure level of the modified gear transmission system at different positions of the ISO standard spherical sound fields is reduced by about 9-14dB, and the sound power level is reduced by 11.92dB, a decrease of 10.42%. The research results show that the gear modification optimization targets model and method can effectively reduce the transmission noise of the high-speed EMU gear transmission system from the root cause.

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