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Contact Mechanics Analysis and Optimization of Shape Modification of Electric Vehicle Gearbox

Qingyong Zhang[a,b*], Yaru Wang[a], Weiping Lin[a], Yongjun Luo[a], Xingjian Wu[a]

[a]*School of Mechanical and Automotive Engineering, Fujian University of Technology, Fuzhou 350118, China*
[b]*Fujian Provincial Key Laboratory of Automotive Electronics and Electric Drive, Fujian University of Technology, Fuzhou 350118, China*

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Abstract

The performance of transmission gears has a great influence on the overall reliability, NVH performance and transmission efficiency of a gearbox. When the gear strength requirement is met, the gear transmission with low vibration and low noise is designed. In view of the NVH problem of automobile transmissions, the third pair of gear pairs of an electric vehicle gearbox is taken as the research object. Based on the nonlinear dynamic model theory of the gear transmission system, the gear profile modification and tooth orientation are comprehensively used to select a reasonable shape modification scheme. The stiffness matrix, mass matrix and position information of the shell extracted from the finite element software are combined with dynamic model of MASTA and the gear pair is optimized for microscopic shaping. Observing the simulation results, it can be seen that the gear pair transmission error and the maximum contact stress after the shape modification are significantly reduced compared with those before the shape modification, and the contact stress map distribution is more uniform. Therefore, selecting a suitable modification scheme can significantly improve the gear meshing effect. The gear transmission error is reduced, and the stress concentration of the gear teeth caused by the assembly error and the elastic deformation is improved, thereby reducing the vibration and noise generated by the gear transmission process.

Keywords: Gear; micro-modification; optimization; transmission error; contact stress.

1. Introduction

As the core component of the automobile gearbox, the smooth running of the gear will have a great impact on the overall reliability NVH (Noise Vibration and Harshness) performance and transmission efficiency of a gearbox. With the development of modern industry, the gear transmission is gradually developing in the direction of high efficiency, precision and intelligence, and the requirements for the working performance of the gear are getting higher and higher. When the gear strength requirement is met, the gear transmission with low vibration and low noise becomes the goal of the study of gear dynamics research. Correct design of the parameters and accuracy of the gear can reduce its dynamic meshing force, and proper dynamic shaping can also reduce the impact. Based on the dynamic model and analysis of the gear transmission system, the overall simulation of the gearbox is carried out to obtain the NVH response of the gearbox, which helps to reduce its vibration and optimize its design. At this stage, the NVH performance of the gearbox is improved mainly by improving the structure, optimizing the process and the control method, gear shaping, and so on. And the gear shaping is an effective method of the lowest cost. Gear shaping is widely used in traditional vehicle transmissions, but there is currently no mature solution for electric vehicles [1-3].

The methods of tooth profile modification are mainly divided into the following five methods: empirical formula method, differential geometry method, elastic mechanics method, function method and finite element method [4, 5].

The empirical formula method considers various factors that affect the deformation of gears under different working conditions, and gives the corresponding empirical formula to determine the amount of modification [6]. Yang Tingli et al. synthetically considered the factors, such as the manufacturing error and elastic deformation of gear teeth, and finally determined the optimum modification amount of gear teeth according to the corresponding formula [7].

Elasticity mechanics method is to use the theory of elasticity to analyze the forces acting on the gears in meshing, and calculate the modification amount needed for the elastic deformation of the tooth surface [8]. Cheng Yikang et al. used the finite element method to analyze the contact strength of the tooth surface under the condition of different tooth tip modification [9].

The function method is to determine the curve equation of the modified section by establishing the shape

* Corresponding author e-mail: zhqy@fjut.edu.cn.

increment function of the tooth top section, the middle section of the tooth profile and the root section of the tooth profile, or by using the curve transition method to determine the modification amount. Singh and Comparin studied the undamped vibration phenomenon of multi-degree-of-freedom systems, and obtained its approximate solution by harmonic balance method and function description method [10].

Finite element method (FEM) is a popular modification method in modern times. It is a discrete numerical simulation method. The basic idea of the finite element method is to discretize the solution object and divide the solution domain of the problem into a series of units. Each unit is connected by nodes, and the unit node quantity equation is established from the energy relationship and the balance relationship, then form each unit equation group into an overall algebraic equation group, and select an appropriate calculation method to solve according to the specific conditions of the equation group[11].Gao Xiaoyu obtained the involute equation, the root transition curve equation and the spiral equation according to the involute gear meshing principle and three-dimensional modeling of the helical gear was carried out and the contact finite element analysis was carried out [12]. Combined with the tooth profile modification theory, the amount of tooth tip modification of the helical gear was determined according to the contact simulation result. The two types of shaping curves were used to modify the helical gears. The comparison of the shaping effects determined the gear shaping curve of the wind turbine gearbox under certain working conditions. Oguz Kayabasi first analyzed and calculated the load distribution on the gear tooth profile by experimental methods. Then the mathematical model of gear tooth profile was established, the tooth profile was optimized by finite element method, and various stress distribution clouds are drawn [13]. Wang Yi et al. used the finite element method to analyze the gear transmission process, determine the shape modification parameters and perform tooth surface modification to reduce gear vibration and noise [14].

As for the tooth orientation modification, Litvin [15] gave the design method of tooth orientation and tooth width direction to optimize the gear transmission error. Xiang Ya calculated the effective contact width and drum shape of gears based on the meshing tooth error and gear tooth deformation, and gave the tooth-shaped drum shape modification curve equation. By creating three-dimensional models of gear pairs with different drum shapes, the three-dimensional contact finite element method was used to calculate the longitudinal load distribution of tooth [16]. Xing Bin et al. considered the influence of installation error and used the finite element quasi-static analysis method to quantitatively calculate the meshing stiffness and load distribution coefficient of different modified gears. The influence of installation error, gear modification amount on gear meshing stiffness and load distribution coefficient is obtained [17]. Xiong Hegen et al. studied a two-stage helical gear reducer by static strength analysis and transmission error analysis, it was found that the contact strength safety factor of high-speed gears was low and there existed danger of pitting corrosion; The low-speed transmission error was relatively large. In order to improve the transmission performance, it was considered to adopt a comprehensive modification method combining the tooth direction drum shape modification and the tooth inclination degree modification. Based on the Romax Designer software, the genetic algorithm was selected to optimize the comprehensive modification of the tooth direction, and the corresponding optimization and modification scheme was obtained [18].

Although a lot of research on tooth shape modification has been done, the tooth shape modification in the dynamic analysis, especially the nonlinear dynamic analysis, is often neglected, and the final result can not truly reflect the influence of the shape modification. In view of the above deficiencies, this paper takes a three-speed gearbox of an electric vehicle as the research object, based on the nonlinear dynamic model theory of gear transmission system, the gear profile modification and tooth orientation are comprehensively used to select a reasonable shape modification parameter. The gear pair is optimized to provide a theoretical basis for improving the NVH performance of the transmission.

2. Modeling of transmission drive system

2.1. Dynamic modeling of the gearbox without body

The mechanical model of the gearbox is built based on MASTA. The model is shown in Fig. 1. The vehicle transmission studied in this paper is a three-speed automatic transmission. The gears are helical ones, which are mainly used in the powertrain of new energy vehicles to improve the dynamics of the vehicle and the operating conditions of motors. At the same design power, the use of the gearbox increases the rated motor speed and reduces motor torque, thereby reducing the size, weight and manufacturing cost of the motor.

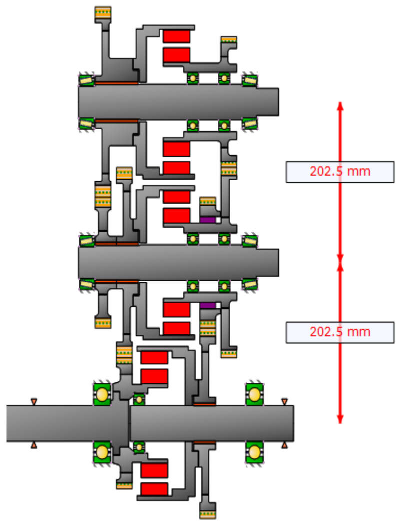


Figure 1. Gearbox two-dimensional model diagram

2.2. Importing of the gearbox housing

When the traditional transmission system is modeled, the gears and bearings are regarded as rigid bodies, and the boundary conditions are set to infinite stiffness. The elastic deformation of the shell, bearing, gear hub and other parts during working is not taken into account. Apparently, there is a certain difference from the actual situation, which can not accurately reflect the actual working state of gearboxes. Therefore, it is necessary to consider the effect of actual stiffness of shell and other parts to get more accurate calculation results.

In this paper, the MASTA software is used to establish the node finite element model. Combined with the MASTA structural flexible module, the stiffness matrix, mass matrix and position information of the shell extracted from the finite element software are imported into the MASTA. The partial data of the stiffness matrix and the mass matrix are shown in Tables 1 and 2[19].

The assembly of the shell is completed by matching the condensed joints with the corresponding bearings, thus introducing the stiffness effect of the whole system, which is more in line with the actual situation, as shown in Fig. 2 and Fig. 3. Considering the actual stiffness effect, the system deformation analysis and gear micro-modification simulation analysis are carried out.

Table 1. Stiffness matrix of the shell

	Node 1 Dx	Node 1 Dy	Node 1 Dz	Node 1 Rx	Node 1 Ry	Node 1 Rz
Node 1 Fx	5004146300	2489301.5	1310509.4	203011.71	91125243	935473.21
Node 1 Fy	2489301.5	248991450	15394431	-1634935.2	232809.22	289414.83
Node 1 Fz	1310509.4	15394431	7339887000	-915190.66	-187594.06	138778.59
Node 1 Mx	203011.71	-1634935.2	-915190.66	2629860.7	-83975.229	17659.09
Node 1 My	91125243	232809.22	-187594.06	-83975.229	59606291	-202385.85
Node 1 Mz	935473.21	289414.83	138778.59	17659.09	-202385.85	2276400.8

Table 2. Mass matrix of the shell

	Node 1 Dx	Node 1 Dy	Node 1 Dz	Node 1 Rx	Node 1 Ry	Node 1 Rz
Node 1 Fx	24.54375	0.02878	0.01885	0.00091	-0.47793	-0.07083
Node 1 Fy	0.02878	5.2774	2.04046	0.07637	-0.00056	0.00076
Node 1 Fz	0.01885	2.04046	15.00319	0.12783	-0.00218	0.00154
Node 1 Mx	0.00091	0.07637	0.12783	0.01037	-0.00017	0.00015
Node 1 My	-0.47793	-0.00056	-0.00218	-0.00017	0.02934	0.00284
Node 1 Mz	-0.07083	0.00076	0.00154	0.00015	0.00284	0.00922

The position information of the housing is as shown in Table 3

Table 3. Location information of the housing

Name	Position (m)		
	X	Y	Z
4766	2.1631E-19	0.0375	-3.4694E-18
4767	-0.101	0.0375	0.17494
4768	0.101	0.0375	0.17494
4769	-2.7039E-19	-0.212	-3.4694E-18
4770	-0.101	-0.212	0.17494
4771	0.101	-0.212	0.17494

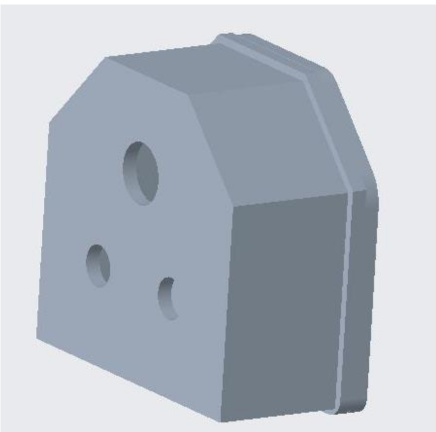


Figure 2. The model of the shell

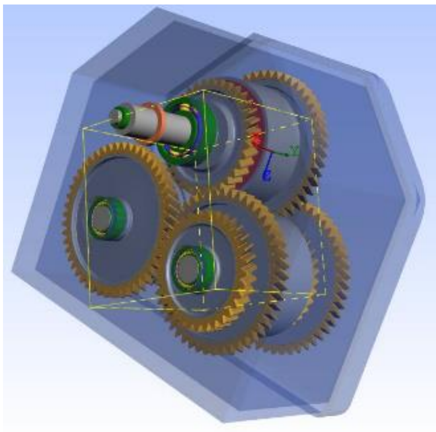


Figure 3. Overall model of the gearbox

3. Nonlinear vibration model and equation of gear system

3.1. Nonlinear vibration model

The concentrating mass method is used to establish the dynamic model of the automobile gearbox [20]. The frictional force and the influence of the oil film during the meshing process are neglected. The helical gear system can be simplified into the gear-bend-torsion-shaft coupling vibration system and the power of a couple of helical gear pairs. The model is shown in Fig. 4, where the system is a three-dimensional vibration system [21].

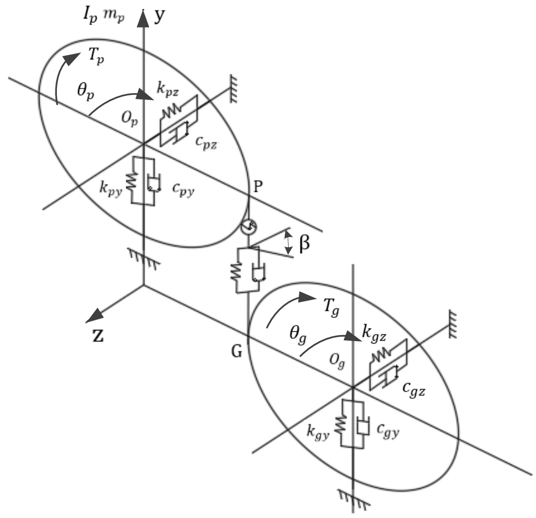


Figure 4. Dynamic model of the helical gear pair

3.2. Nonlinear vibration differential equation

From Fig. 5, assuming that the direction of rotation of the driving gear is right-handed and the helix angle is β, the relationship between the lateral direction and the axial vibration at the meshing point can be expressed as:

$$x = y\tan\beta \tag{1}$$

Regardless of the inter-tooth surface friction, the system has 6 degrees of freedom, and its generalized displacement array is:

$$\{\beta\} = \{y_p z_p \theta_p y_g z_g \theta_g\}^T \tag{2}$$

where: $y_i\, z_i\, \theta_i\ (i = p, g)$ are the translational and angular vibration displacements of the driving wheel midpoint O_p and the passive wheel midpoint O_g in the y direction and the z direction, respectively.

Therefore, the relationship between the vibration displacement at point O_p and the vibration displacement of the driving wheel and O_g point and the generalized displacement of the passive wheel is:

$$\begin{cases} \bar{y}_p = y_p + \theta_p R_p \\ \bar{y}_g = y_g - \theta_g R_g \\ \bar{z}_p = z_p - \bar{y}_p \tan\beta \\ \bar{z}_g = z_g - \bar{y}_g \tan\beta \end{cases} \tag{3}$$

Assuming that the normal stiffness of the tooth engagement is k_m, normal damping is c_m and normal meshing error is e, their components in the y and z directions are:

$$\begin{cases} k_{my} = k_m \cos\beta \\ c_{my} = c_m \cos\beta \\ e_y = e\cos\beta \end{cases} \tag{4}$$

$$\begin{cases} k_{mz} = k_m \sin\beta \\ c_{mz} = c_m \sin\beta \\ e_z = e\cos\beta \end{cases} \tag{5}$$

Therefore, the tangential dynamic meshing force F_y of the system is:

$$F_y = k_{my}\left(\bar{y}_p - \bar{y}_g - e_y\right) + c_{my}\left(\dot{\bar{y}}_p - \dot{\bar{y}}_g - \dot{\bar{e}}_y\right) \tag{6}$$

The axial dynamic meshing force F_z of the system is:

$$F_z = k_{mz}\left(\bar{z}_p - \bar{z}_g - e_z\right) + c_{mz}\left(\dot{\bar{z}}_p - \dot{\bar{z}}_g - \dot{\bar{e}}_z\right) \tag{7}$$

Therefore, the dynamic equation of the model can be derived as follows:

$$\begin{cases} m_p\ddot{y}_p + c_{py}\dot{y}_p + k_{py}y_p = -F_y \\ m_p\ddot{z}_p + c_{pz}\dot{z}_p + k_{pz}z_p = F_z \\ I_p\ddot{\theta}_p = -T_p - F_y R_p \\ m_g\ddot{y}_g + c_{gy}\dot{y}_g + k_{gy}y_g = F_y \\ m_g\ddot{z}_g + c_{gz}\dot{z}_g + k_{gz}z_g = -F_z \\ I_g\ddot{\theta}_g = -T_g - F_y R_g \end{cases} \tag{8}$$

4. Gear microscopic modification

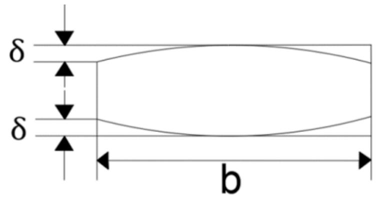
4.1. Gear tooth direction microscopic modification

The tooth orientation modification is to trim the tooth surface in a slight amount along the tooth direction, so as to deviate from the theoretical tooth surface, thereby improving the uneven distribution of the load along the tooth contact line and improving the bearing capacity of gears. The tooth direction modification mainly includes three methods of drum shaping, tooth end thinning and spiral angle modification.

The drumming is to make the teeth bulge in the middle of the tooth width, and the two sides are symmetrically distributed. Drumming is a well-adapted shaping method that compensates for the elastic deformation of gears, as shown in Fig. 5. The parameters for the drum shaping are generally used as the initial value of the elastic deformation of the tooth contact, and the formula for calculating the elastic deformation δ is:

$$\delta = 1.16\frac{F_n}{LE} \tag{9}$$

where: F_n is the normal load (N) of the gear meshing; L is the length of the contact line (mm); E is the elastic modulus.



Fgi.5 Drumming modification

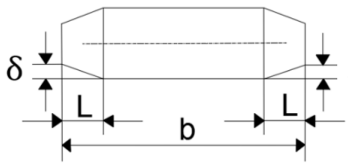


Figure 6. Tooth end thinning

Tooth end thinning means that one end or both ends of the gear teeth are gradually thinned toward the end at a small length of the tooth width, as shown in Fig. 6. The amount of modification is determined only by the amount of elastic deformation. For helical gears, the range of modification is $0.013mm \leq \delta \leq 0.035mm, L = 0.25b$, b is the effective width (mm) of the gear. The helix angle modification refers to a slight change in the helix angle of the gear so that the actual tooth surface deviates from the theoretical tooth surface position, as shown in Fig. 7.

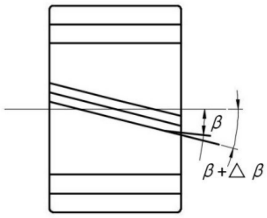


Figure 7. Spiral angle modification

4.2. Gear profile micro-shaping

During the gear meshing transmission, since gears are alternately engaged by single and double teeth, there is a significant abrupt change in the load distribution. As the load changes, the gear teeth also undergo corresponding deformation, resulting in an increase in the transmission error of gears, which in turn generates vibration, shock, noise and other issues. The tooth profile modification is to cut off the interference part of the tooth surface during the gear meshing process to reduce the transmission error and increase the meshing transmission stability, thereby avoiding the phenomenon of sudden load change. The profile modification mainly includes the top edge trimming and the root modification. The tipping edge is the position at which the teeth contact the line when the tooth begins to enter the engagement at the addendum of the driven wheel and the root of the drive wheel. To avoid early contact, the driven wheel is generally trimmed near a portion of the crest. The roots are shaped to not only be in the vicinity of the crests, but also to correct the roots of the mating gear teeth that are intended to mesh.

The profile modification parameters are mainly the amount, the length and the curve of modification. The initial shape modification is generally acquired by the calculation of the tooth profile elastic deformation amount δ_a:

$$\delta_\alpha = \frac{\omega_t}{C_r} \tag{10}$$

where: ω_t is the unit tooth width load (N/mm); C_r is the meshing stiffness of the gear $(N/mm \bullet \mu m)$.

The calculation ω_t of is as follows:

$$\omega_t = \frac{F_t}{b} \tag{11}$$

where: F_t is the tangential force (N) on the gear index circle.

The C_r is acquired as follows:

$$C_r = \sum_{i=1}^{n} \frac{F_i}{(\delta_{1i} + \delta_{2i})} \tag{12}$$

where δ_{1i} and δ_{2i} are the comprehensive deformation of the driving wheel and the driven wheel respectively; F_i is the contact force of gears (N); n is the number of contacting teeth.

The modification length can be divided into long modification and short modification. The long modification is from the starting point (or termination point) of meshing to the alternation of single and double pairs of teeth. Short modification is one-half of the starting point (or termination point) of meshing to long modification. The long modification will cause the meshing part of gears to be less than one base pitch, which easily causes the meshing discontinuity of gears to produce impact. Therefore, the short modification is often used in the profile modification. The long modification is calculated according to formula (13) and the short modification is calculated according to formula (14).

$$\begin{cases} l_t = (E_\alpha - 1)\, p_{bt} \\ l_r = p_{bt} \end{cases} \tag{13}$$

$$\begin{cases} l_t = \dfrac{E_\alpha - 1}{2} p_{bt} \\ l_r = \dfrac{E_\alpha + 1}{2} p_{bt} \end{cases} \tag{14}$$

where: E_{α} is the contact ratio of gears; p_{bt} is the pitch of gears.

The shape of gear modification varies with the parameters. The general formulas are as follows:

$$\Delta = \Delta_{max} \left(\frac{X}{L} \right)^{\beta} \tag{15}$$

Where X is the length from any point in the meshing area to the point meshing in or meshing out; L is the length of the meshing area of two teeth; Δ_{max} is the maximum modification amount of gears; β is the index, and $1 < \beta < 2$ can be taken.

5. Gear modification design and result analysis

5.1. Scheme design

The gearbox is a three-speed gearbox with four pairs of gears engaged in meshing. The gear parameters are shown in Table 4.

Table 4 Gear pair parameters

Gear pair		Modulus	Number of teeth	Helical angle (°)	Pressure angle (°)	Rotation direction
The First pair	Input	5	33	6	20	Left
	Output	5	47	6	20	Right
The second pair	Input	5	33	6	20	Right
	Output	5	47	6	20	Left
The third pair	Input	5	33	10	20	Left
	Output	5	47	10	20	Right
The fourth pair	Input	5	31	13	20	Right
	Output	5	48	13	20	Left

Before defining the load spectrum, input and output power flows should be added to the input and output axes of the gearbox model respectively. The loads on the transmission system are constantly changing and the frequency with which different gears are used at work, that is, the time taken for each gear is also different, so the relation forms a load spectrum. The load spectrum of the actual working of the gearbox provides accurate input conditions for the operation of the transmission system, so that accurate calculation results of the force of each part can be obtained.

The working conditions of the third gearbox are shown in Table 5.

Table 5. First gear condition of gearbox

Load Case (%)	Duration (hr)	Ambient Air (°C)	Speed (r/min)	Torque (N·m)
25	60	20	1000	400
50	30	20	1000	600
75	60	20	2000	800
100	30	20	2000	1000

The input of the gearbox model is defined mainly according to the load spectrum, which is the combination of different working conditions of the gearbox, and the working conditions refer to the speed, torque and acting time on the gearbox under a certain power flow. For the same vehicle, the actual load spectrum will be affected by road conditions and driver operation. Therefore, a number of tests are carried out on different roads and driving habits by special testing instruments. Statistical data are extracted as the reference load spectrum for the design of transmission system. The load spectrum is defined according to Table 5, as shown in Table 6.

Because all gears engage in meshing when the gearbox works in first gear, and the gearbox loads are the largest at this time, this paper mainly checks the influence of loads on each pair of gears in first gear.

Table 6. Load Spectrum of First Gear of Gearbox

Design state	Load Case	Duration (hr)	Power Load	Speed (r/min)	Torque (N·m)
1st Design State	25% Load	60	Input Power Load	1000	400
			Output Power Load	223.5469	-1789.3336
	50% Load	30	Input Power Load	1000	600
			Output Power Load	223.5469	-2684.0004
	75% Load	60	Input Power Load	2000	800
			Output Power Load	447.0939	-3578.6672
	100% Load	30	Input Power Load	2000	1000
			Output Power Load	447.0939	-4473.3339

Without any optimization of gears, the damage rate and safety factor of each pair of gears can be obtained by implementing the advanced system deformation, as shown in Table 7. The safety factor is that in engineering design, in order to prevent the consequences caused by material defects, manufacturing and assembly errors, and external force surges, the theoretically capable force of the force part must be greater than the actual force，that is, the ratio of allowable stress and actual stress, the theoretical safety factor of the gear should be greater than 1. If equal to 1, that is, the allowable stress and the actual stress are equal, the safety factor has no meaning; If it is less than 1, the gear system will be close to the critical value, which will cause unsafe accidents in case of external load surge. Different pairs of gears have different safety factors due to different manufacturing quality, reliability, external load force and other factors. From Table 7, it can be seen that the bending safety factor of the third pair of gears does not meet the theoretical requirements, which is the weakest link in the whole system.

Table 7. Gear safety factor and damage rate

Gear pair		Contact		Bending	
		Safety factor	Damage rate	Safety factor	Damage rate
First pair	Input	1.5921	0	1.3141	0
	Output	1.6439	0	1.2823	0
Second pair	Input	1.4273	0	1.6099	0
	Output	1.4745	0	1.0089	92.49
Third pair	Input	1.0684	41.12	0.9712	129.88
	Output	1.1048	26.36	0.9678	141.71
Fourth pair	Input	1.0896	31.76	1.1193	21.82
	Output	1.1247	20.87	1.2431	121.7

Therefore, in order to optimize the contact stress of the tooth surface and improve the service life of the gear pair, the micro-modification of the third pair of gears is carried out in this paper. According to the calculation results, the initial modification parameters of the pinion and big gear are determined. By comparing the simulation results, the modification parameters are adjusted and optimized. The contact spots, transmission errors and the reduction of maximum tooth contact and root bending stress are taken as evaluation criteria. Finally, the micro-modification quantity of pinion and big gear is determined as shown in Table 8.

Table 8. Results of micro-modification results of the third pair of gears

Shape modification	Tooth drum shape (μm)	Tooth helix angle (μm)	Tooth Profile Drum Quantity (μm)
Pinion	5	35	15
Big gear	30	40	22

The modification curve is shown in Fig. 8 and Fig. 9.

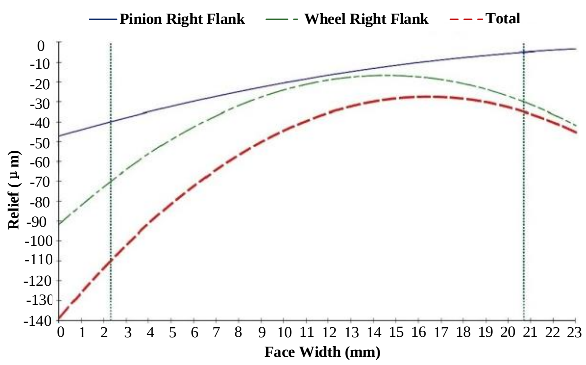


Figure 8. Tooth modification

5.2. Analysis of micro-modification results

After modification, the safety factor and damage rate of the third pair of gears are shown in Table 9. The damage rate decreases, the bending safety factor of pinion increases from 0.9712 to 1.2342, the bending safety factor of big gear increases from 0.9678 to 1.233, the bending safety factor of gears is greater than 1, and the contact safety factor also increases, which all meet the theoretical requirements of the gear safety factor. Therefore, the results show that the micro-modification of gears can improve the safety factor of the gear and prolong its service life.

Table 9. Gear safety factor and damage rate after modification

Gear pair		Contact		Bending	
		Safety factor	Damage rate	Safety factor	Damage rate
The third pair	Input	1.2825	1.63	1.2342	0.033
	output	1.3258	0.92	1.233	0.18

Fig. 10 is a comparison of the transmission error of the third pair of gears before and after shaping at different torques. Fig. 10(a) is the comparison of the transmission error before and after the gear modification at 25% torque, and Fig. 10(b) is the comparison of the transmission error at 50% torque. Fig. 10(c) is the comparison of the transmission error before and after the gear modification at 75% torque. Fig. 10(d) is the comparison of the transmission error before and after the gear modification at 100% torque The overall analysis shows that the periodic variation of the transmission error curve after gear modification is smoother.

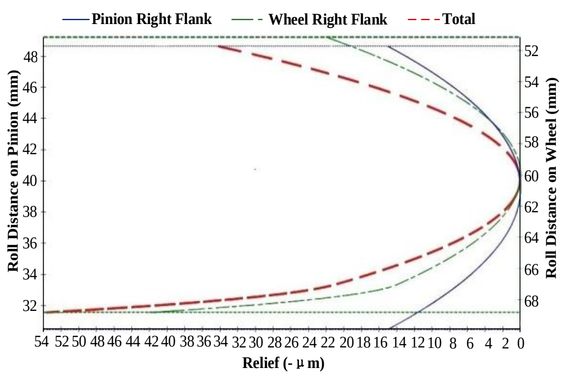


Figure 9. Tooth profile modification

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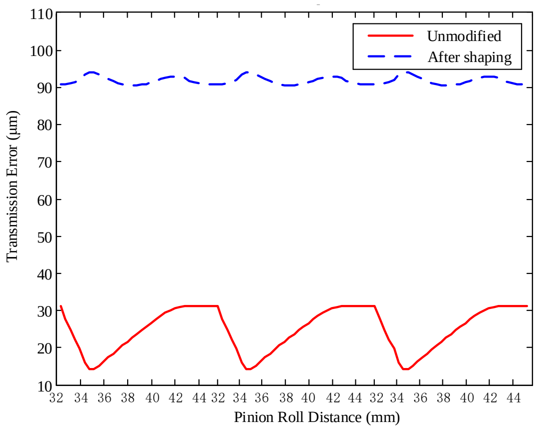
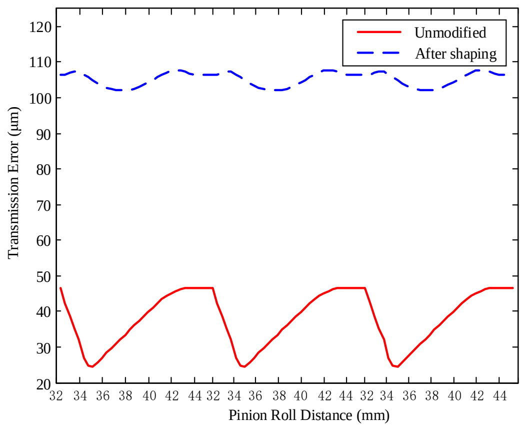
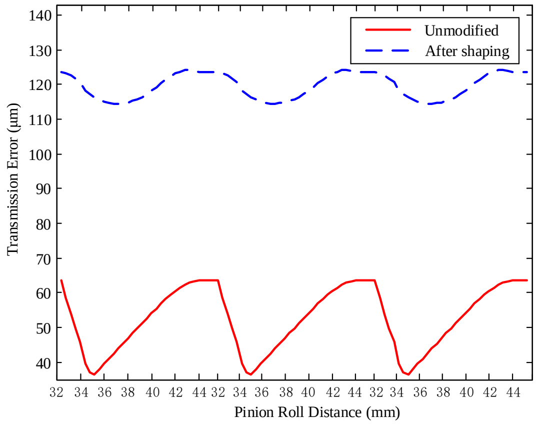


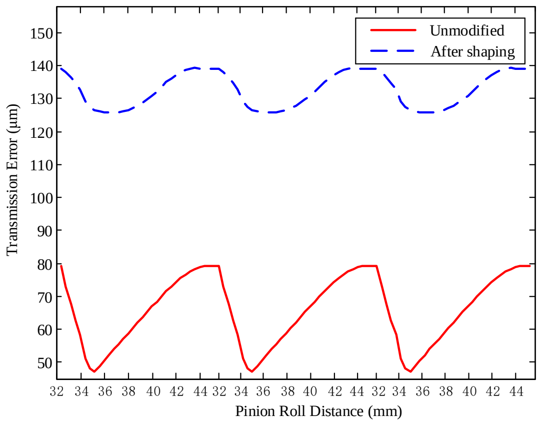
(a) Transmission error under 25% torque



(b) Transmission error under 50% torque



(c) Transmission error under 75% torque



(d) Transmission error under 100% torque

Figure 10. Comparison of transmission error before and after gear shaping

Although the transmission error value after modification is larger than that before the modification, the variation range of the transmission error under the four conditions after modification is known from Fig. 11, which are significantly smaller. The peak value is reduced by about 60% and the modification effect is remarkable. Among them, the transmission error of pinion and large gear is reduced from 22.2μm to 5.7μm under 50% torque, and the transmission error of pinion and large gear is reduced from 27.2μm to 9.6μm at 75% torque. Therefore, it can be seen that the micro-modification of gears can reduce the fluctuation range of the transmission error of the gearbox of the electric vehicle, so that the gear transmission is more stable and the vibration noise is reduced.

Fig. 12 shows the change of contact spot before and after modification under different torques. It can be concluded that the stress on the front tooth surface is not uniform, the stress on the left end of the gear is large, the gear teeth are prone to fracture during meshing, and the service life of the gear is low. After modification, the eccentric load of tooth surface is eliminated, the contact spot is uniformly stressed, and the maximum contact stress is close to the center of the gear. The load per unit length of the tooth surface is reduced by about 25% under different torques and the effect is remarkable. The maximum stress at 50% torque decreased from 1679 Mpa to 1212 Mpa, and the maximum stress at 75% torque decreased from 1894 Mpa to 1363 Mpa. Therefore, considering the transmission error and contact stress of the modified gear, the micro-modification can prolong the service life of the gear teeth and improve the gear meshing performance, achieving the expected effect.

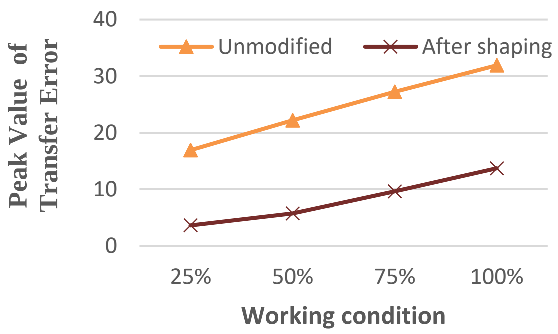
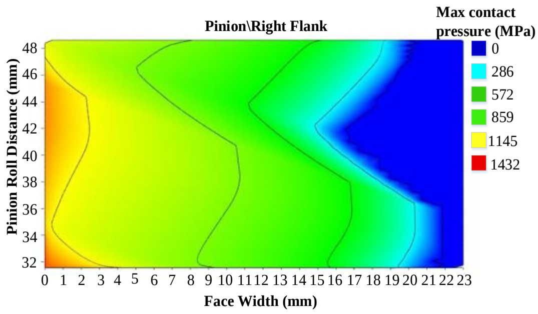
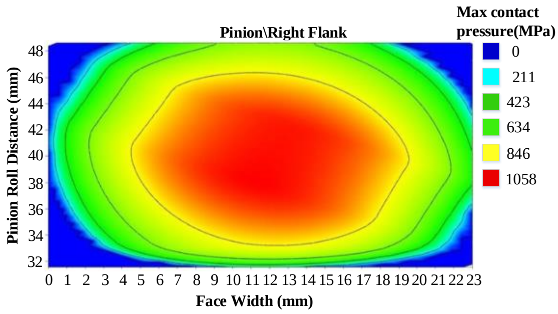


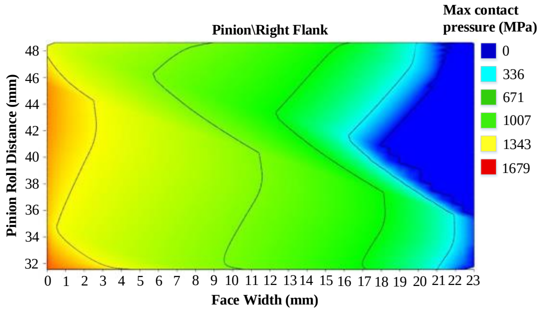
Figure 11. Comparison of peak value before and after modification



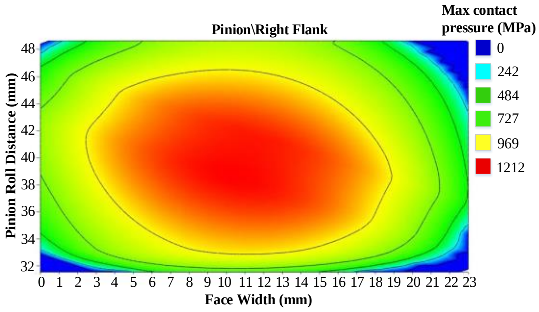
(a) Before modification under 25% torque



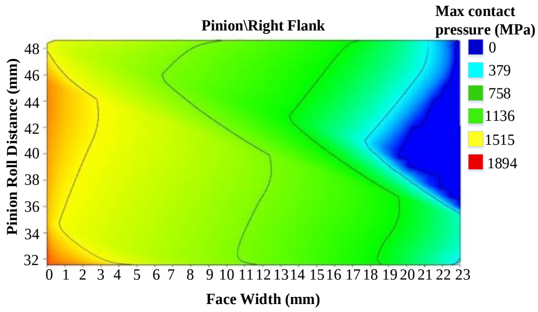
(b) After modification under 25% torque



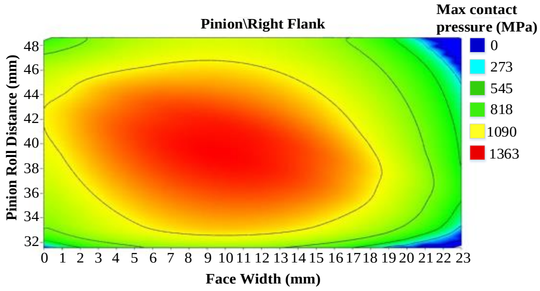
(c) Before modification under 50% torque



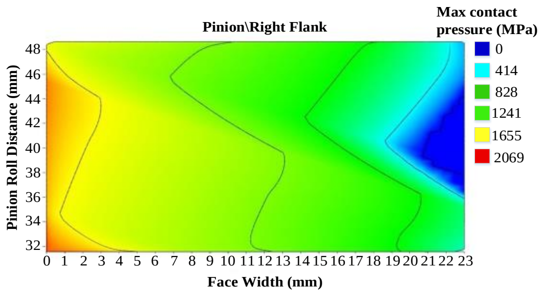
(d) After modification under 50% torque



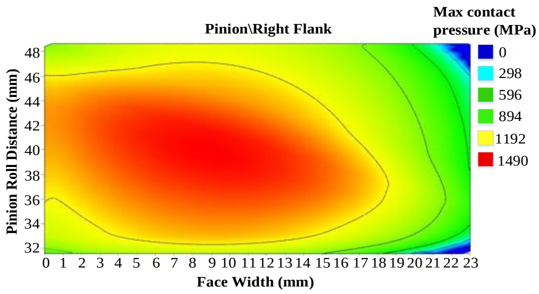
(e) Before modification under 75% torque



(f) After modification under 75% torque



(g) Before modification under 100% torque



(h) After modification under100% torque

Figure 12. Comparison of contact spots before and after gear modification

6. Conclusion

1. The finite element model of the gearbox joint is established through the software of MASTA. The safety factor and damage rate of the four pairs of gear pairs of the gearbox are obtained through system deformation analysis. It is judged that the safety factor of the third pair of gears is less than 1, which is easy to cause damage, and this is used as the modification target gear. The transmission error, maximum contact stress, and contact spot diagrams are obtained by simulation. The optimum shape is determined by comparison and analysis before modification, so as to improve the safety factor and optimize the gear meshing performance.
2. Based on the common operating conditions of electric vehicles, at 75% torque, the transmission error of the third pair of gears decreased from 27.2 m to 9.6 m after the modification, which was about 65% lower than that before the modification, and the transmission error curve changed more smoothly periodically. The maximum contact stress of the gear after modification decreased from 1894Mpa to 1363Mpa, about 28% lower than that before modification, and the contact spot becomes even, eliminating the off-load phenomenon of the tooth surface, which greatly improves the stability of the gearbox in work, and extending the service life of the gear transmission system.
3. Gear transmission error analysis and load contact analysis play a good role in optimizing the performance of transmission NVH. Through the contact analysis of the gear, the contact situation can be directly judged, and the gear shape modification can be guided to improve the transmission error of the gear and the contact between the gear teeth.

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