DYNAMIC CONTACT ANALYSIS AND TOOTH MODIFICATION DESIGN FOR EMU TRACTION GEAR

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Abstract

Taking traction helical gear of EMU CRH380A as an example, the meshing characteristics was analysed. Based on Pro/E, three-dimensional helical gears were modelled. Combined with ANSYS, the gears' contacts were analysed under multi-condition, such as start-up, continuous and high speed. Through transient dynamics, the distribution cloud of the equivalent stress and contact pressure in a meshing period are solved, which at different meshing position under different condition. And the contact state, the stress changes and distribution regularity in the meshing process were analysed. Then on the basis of the results, gear modification parameters were designed, and the modification gears finite element model were constructed. To compare gear contact stress distribution before and after being modified, it shows that the modification scheme can effectively reduce the gear meshing impact and the transmission noise.

(Received, processed and accepted by the Chinese Representative Office.)

1. INTRODUCTION

With the continuous development of China's rail transport, EMU has become an important and modern symbol of railway science and technology. Speed is the eternal pursuit of transport. Noise is the overriding factor in limiting the speed [1]. Researches showed that the person's hearing may suffer serious damage that stays in the environment above 80 dB (A) noise for a long time. Nowadays, in the European and Japanese, noise has been important indicators to evaluate the comfort of high speed train. According to the Prof. Zhou and others' researches [2], if a train will travel at the speed of 350-400 km/h, noises of sightseeing area will reach to 93-98 dB (A), and the noise will increase linearly with the increase of speed.

The noise source of EMU is mainly composed of wheel-rail noise, structure-borne noise, the noise of traction motor, the electromagnetic noise of Electric-Current Collecting System, aerodynamic noise and so on. According to the high-speed EMU noise tested by Zhang et al. [3], it shows that the middle and low frequency noise is more obvious than the high frequency noise; the frequency band is mainly around 10-1000 Hz. For the test of dynamic noise and vibration from the locomotive traction gear transmission system, Li [4] found that the peak noise frequency of the traction gear transmission is mainly concentrated in the low frequency about 450-1300 Hz, and became one of the main noise sources of the vehicle. The essence of high speed EMU traction gear transmission system is both Multi-DOF and nonlinear with time-varying parameters and the clearance, its structure-borne noise is mainly caused by the dynamic response from the internal and external excitation as the large and small gear meshing. As the speed increases, the impulsive noise will increase significantly, and it will lead some different breakdowns and damages, such as pitting, tooth root bending fatigue, break and so on, so the study of contact stress of traction gear is very meaningful.

On the basis of the energy conservation law, Bajer and Demkowicz [5] established a rigid body contact finite element model for gear system and studied dynamic contact impact problems of galaxies gear system as meshing. Using the finite element principle combined with the contact mechanics theory, Parker et al. [6] studied the gear pair nonlinear dynamic response considering the time-varying stiffness and tooth clearance. Wu et al. [7] established the finite element model of meshing gear pair and explored continuous dynamic characteristics in elastic meshing process at the low-speed and high-speed. Taking the domestic high-speed train at the speed above 300 km/h as the research object, Zhong et al. [8] established the classical dynamic model and the time-varying characteristic model on gear pair meshing stiffness, and simulated dynamically the transmission system under the driving condition. Hu et al. [9] used finite element software and display dynamic analysis method to explore dynamic meshing characteristics of helical gear pair with different displacement coefficient and helix angle parameters.

Traction drive system is the power source core of running parts of high-speed EMU. It was right under the passenger compartment. The vibration noise generated by the alternating meshing of the gear under high-speed operation not only reduces the performance of the gearing system but also reduce the passenger comfort. What is more, it even endangers the safety of train operation [10]. Therefore, it is necessary to further study the stability of traction gear drive and noise reduction technology. According to the gear noise theory, gear modification is a common measure to improve transmission performance [11], reduce vibration and noise. Reasonable tooth shape can effectively reduce the shock vibration and dynamic load of gear tooth, reduce vibration and noise caused by internal excitation, achieve uniform tooth surface load and improve transmission performance.

Park et al. [12] measured the teeth modification and meshing line tilt amount as design variables, evaluated the dynamic meshing force fluctuation between the planetary gears, assessed the stress distribution uniformity, and optimized the planetary gear modification parameters for wind turbine gearbox. Huang [13] built a modification gear model for EMU and studied the influence of three modification parts & elements on tooth contact by the finite element dynamic simulating method. He came to a conclusion that the quadratic and sine curve effect is good for contact as meshed into and out. Kognole [14] proposed a method to calculate the modification parameters based on working condition. He applied contact spot or root strain gauge method, calculated the tooth load, surface contact and root bending stress through the finite element program as the gear with machining error and assembly error. And then he modified the gear according to the calculation results. Zhu et al. [15] established a dynamic contact finite element model for megawatt wind turbine gearbox. By using ANSYS / LS-DYNA explicit dynamic method, he determined the curves and parameters by calculating the amount of modification. Yang et al. [16] determined the best modification parameters by analysing the relationship among the modification parameters, transmission error and contact stress.

Gear modification methods and parameters should be based on specific structural parameters, actual traction conditions and meshing characteristics. From the existing literature, lots of scholars' studies are limited in a specific condition [17, 18], and current high-speed EMU traction gear modification parameters are based on the continuous traction condition, which is different from the actual multi-condition, as well as ignores the impact of other traction conditions such as start-up, high-speed, braking and so on. In the design and test of planetary gear system in transmission differential of open-rotor engine, by finite element analysis and tooth contact analysis, Imai et al. [19] found that the optimal modification value, the length, and the curve of modification are different under different working conditions. In this paper, taking G301 traction helical gear of EMU CRH380A as an example, the dynamic contact under multiple operating conditions was analysed by the transient dynamic method,

the ANSYS analysis and Pro/E model. According to the results, the three parts and elements of gear modification are calculated and designed, and the modification effect is compared.

2. TRACTION CONDITIONS AND CHARACTERISTICS OF CRH380A

CRH380A continuous operation speed is 350 km/h, the maximum operating speed is 380 km/h, and it is one of the mainstream models of China's high-speed EMU. For traction helical gear G301, its helix angle is 26° and its normal pressure angle is 26°. According to the theory of gear transmission, when the helix angle is more than 16°, there will be large axial force to cause a larger temperature rise. With the pressure angle increasing, the normal pressure of the tooth surface increases. Thereby the gear pitches line impact and mesh momentum increases. Especially when gear works at high-speed, the non-linear shock and vibration noise will be amplified, due to the gear transmission error, wheel and wheel body elastic deformation as well as thermal deformation. Only through the precise modification design, it can satisfy the requirements of high precision and high performance transmission for high-speed EMU.

Fig. 1 is the relationship among tractive effort at wheel rim, motor power and EMU running speed. Curves I and IV respectively are locomotive torque curve and acceleration time curve at the design state. Curves II and III are those in the work state. The *o-a* is the traction motor constant torque start-up phase, i.e. from rest to normal operation state. It is also called start-up conditions. Seeing from the curves I and II in start-up conditions, the locomotive torque is large and maintain its value, but decreases slightly with the increase of the speed due to the limit from the starting current; the *a-b* is the constant power continuous running phase, which is the maximum continuous power, also the longest running stage, and called as the continuous conditions. Under this condition, due to the limited traction motor power, with traction force decreases, the acceleration is gradually reduced, and the running speed is gradually increased. The *b-c* is the high-speed traction stage, also called as high-speed conditions. In curve III, the speed of point C is close to 380 km/h, the highest speed during locomotive actual running. In high-speed conditions, with the higher speed of the traction gear, the gear body will expand along radial direction, and generate the great meshing shock.
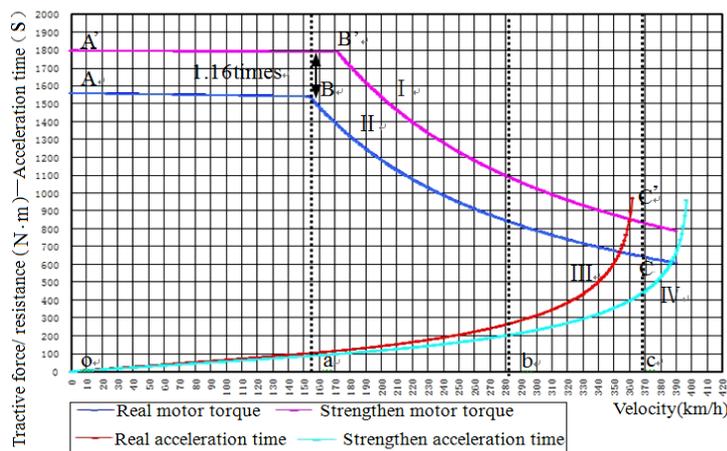

Figure 1: The traction motor torque and acceleration time curve of CRH380A in second stage.

3. THE ESTABLISHMENT OF 3D MODEL OF GEAR PAIR

3.1 Gear parameters

The materials of G301 small gear is 20CrNi2Mo, the surface adopt hardening and carburizing technology, gear head hardness is 75-85 HS; the big gear of G301 is S40C-H, the surface

adopt high frequency quenching technology, gear head hardness is 65-75 HS, the root hardness is over 60 HS. The modulus of the gear pair is smaller than that of ordinary high speed and heavy duty locomotive gears, and the helix angle and the normal pressure angle are relatively big compared to that of ordinary. The main parameters are shown in Table I.

Table I: Gear parameter table.

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

3.2 The geometric parameters of gear tooth

In Pro/E, using the default reference coordinate system draft four datum curves of gear pitch circle, base circle, addendum circle and root circle (their diameters are respectively expressed as d, d_b, d_a, d_f). In order to realize parameterized modelling, the gear geometry parameter constraint equations are added in the editor dialog box, as shown in Table II.

Table II: Gear reference circles relationship.

The geometry constraint equations of four reference circles	
$d = m_n * Z / \cos(\beta)$	$d_b = d * \cos(\alpha_t)$
$h_a = \left(h_a^* + X_n\right) * m_n$	$d_a = d + 2 * h_a$
$h_f = \left(h_a^* + c^* - X_n\right) * m_n$	$d_f = d - 2 * h_f$

3.3 Establishment of the model

In turn, to create the end-face involute, end profile, spiral and tooth entity model of the helical gear, by stretching, variable cross-section scanning, tooth array and other operations, the three-dimensional models of the big gear and small gear of gear pair are built respectively. Their assembly model is constructed by defining the assembly constraints relations between the two gears, as shown in Fig. 2.
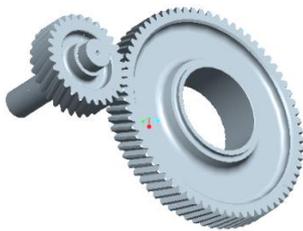

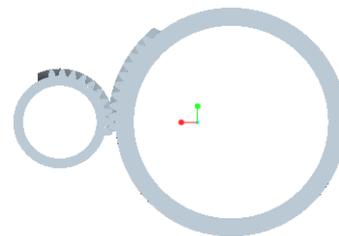Figure 2: Assembly model of traction gear pair of EMU.　　Figure 3: Simplified model.

In principle, the whole model should be chosen as the exact gear pair mesh finite element model. However, considering that the dynamic contact analysis of gears is a highly nonlinear and the calculation is quite complex, therefore, the gear model is simplified by only respectively remaining big gear and small gear 10 teeth and removing other features. The simplified gear assembly is shown in Fig. 3.

4. DYNAMIC CONTACT ANALYSIS OF GEAR PAIR

There were many achievements in the study of contact stress and contact strain of gear meshing by using finite element method. However, most of them used static contact analysis of two-dimension or three-dimension. And though the static contact analysis can get the contact stress of gear meshing by many times solving, yet it can't accurately reflect the actual gear contact state. In this paper, dynamic contact analysis of traction gear pairs is carried out by means of transient dynamic analysis.

4.1 The transient dynamic analysis

Transient dynamics is a technology to research the structure dynamic response process under the load with arbitrary changes over time. It is a kind of time domain analysis. The input data is the load on the value as a function of time, and the output data is the displacement, strain and stress, etc. The basic equation of the transient dynamic is:

$$[M]\{x''\}+[C]\{x'\}+[K]\{x\}=\{F(t)\} \tag{12}$$

In Eq. (1), $[M]$ is the mass, $[C]$ is the damping, and $[K]$ is the stiffness matrix; $\{x\}$ is the displacement, $\{F(t)\}$ is the force, $\{x'\}$ is the velocity, and $\{x''\}$ is the acceleration vector. For any fixed time (t), Eq. (1) can be regarded as a static equation considering both damping force $[C]\{x'\}$ and inertial force $[M]\{x''\}$. For the equations at different time points, the time increment Δt between two arbitrary adjacent time points t_1 and t_2 is called integral time step. In ANSYS workbench, it can be solved by using the Newmark time integral method.

4.2 Three-dimensional meshing model of gear pair

Define the gear material properties
CRH380A EMU traction helical gear material parameters are listed in Table III.

Table III: The Parameters of gear material.

	Elastic modulus E (MPa)	Poisson's ratio μ	Density ρ (kg/m^3)
The large gear	2.0×10^{11}	0.29	7800
The small gear	2.06×10^{11}	0.31	7850

In the ANSYS Workbench, to select "Analysis Systems" in the toolbox and click on "Transient Structural", to define the finite element analysis system as the transient dynamics system, double-click the sub-module "Engineering Data" in the transient dynamics analysis system, then the material properties window is pop-up. To select the "Structural Steel" option on the "Outline of Schematic A2: Engineering Date", change it to "gear1", the material properties of "gear1" can be filled in. Similarly, the material properties of "gear2" are set.

Build a finite element model
In the ANSYS Workbench, to right-click "Transient Structural", and choose "Import Model" → "Browse", from the folder, select the traction helical gear assembly model file which created by Pro/E before. To select the "Model" option in the transient dynamics analysis system, the system can be entered.

The two sides of the large and small gears are selected and respectively named as "C2", "C1". This step is mainly to facilitate the choice of contact surface and target surface after the contact pair is established between the two gears. The contact surface is generally to choose the surface being a relatively soft tooth. In this paper, it is manually modified that the target surface of contact pair as the tooth surface of the driving gear (small gear), and the contact

surface as the large gear tooth surface. To select the contact type for friction, the friction coefficient is filled in 0.1.

Finite element mesh division

Meshing has obvious influence on the calculation results; there are a few points to note. First of all, it is the number of meshes. The number of meshes can be determined by comparing the results of the meshing calculation. When the number of meshes does not significantly affect the analysis results, it is not necessary to continue to increase the number of meshes. Secondly, it is the mesh density. Depending on the influence of the calculation results, different sizes of meshes can be divided into different positions. In this article, the gear contact stress is mainly analyses. The meshes of the gear teeth should be denser with smaller meshes, and larger meshes in other areas. Thirdly, it's the quality of the mesh. In ANSYS Workbench, the automatic mesh can make analysis model to generate a suitable meshes, considering the number of meshes and meshes density impact on the calculation results, it is necessary to refine the meshes in the place of teeth meshing where stress is concentrated (Fig. 4).
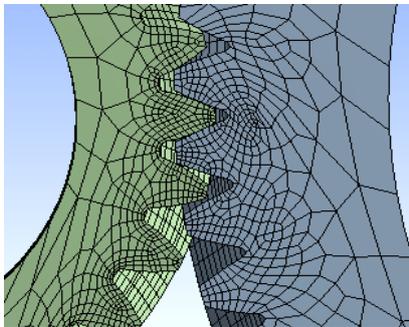

Figure 4: Mesh refinement results.

4.3 Finite element solution

Due to the basic consistency of the transient dynamic solution in the start-up, continuous and high speed working conditions. In this paper, the dynamic contact analysis process is introduced under the start-up condition as an example.

Impose constraints

As the analysis of dynamic contact stress of the helical gear pair, both the big gear and small gear must have degree of freedom of rotation around self-axis. In order to simplify the model, the redundant degrees of freedom can be constrained.

Set the time steps

The choice of time step should be appropriate, too long or too short will affect the simulation time and the accuracy of the results. To click "Analysis Settings", then in the detail window "Step End Time", "Initial Time Step", "Minimum Time Step" and "Maximum Time Step" are respectively set as "0.004", "0.0002", "0.0002" and "0.0003", and other remain the default.

Define the load

A rotational speed is applied to the driving gear (pinion gear), and a torque is applied to the driven gear (large gear). The rotational speed and torque applied under the three working conditions are shown in Table IV.

Table IV: Gear speed and torque in three working conditions.

	Start-up	**Continuous**	**High speed**
Torque (N·m)	1900	841.5	569.5
Rotate (rad/s)	31	434	641

Define the speed

The following operations are done: right-click on the "Transient", add "To be Load", select the previously created revolute pair of pinion gear, namely "Revolute - Ground To Gear1", "Type" to "rotation", set speed value in the table data. In order to guarantee the convergence, the rotational speed is changed from 0 to 31 rad/s at a time step.

Define the resistance torque

As above, add the hinge, select the previously created pinion rotation pair, "Revolute-Ground to Gear2", then set the "Hinge type" to "Moment" and the torque value to "1900".

Define the solution content

Equivalent stress and contact stress need to be analysed. The following operations are done: in turn to right-click "Solution", "Solution", "Contact Tool", and "Solution", and select "Equivalent", "Contact Tool", "Pressure" and "Solve". Then the Solution can be waiting for.

4.4 Solution results analysis

The Mises stress nephogram (Fig. 5) shows that the equivalent stress pattern of the gear pair at one moment under the start-up condition. It can be seen that in the moment of contact gear, the equivalent stress is the largest. The stress gradually spreads to gear wheel body, and the stress at roots of the tooth is larger. And the conclusion is similar in the other two conditions.
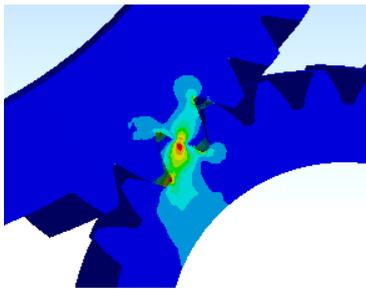

Figure 5: Mises stress nephogram of gear contact in start-up condition.

Fig. 6 is also a cloud diagram, which shows the contact stress of the helical gear pair at different meshing positions in same engagement cycle under the start-up conditions. Among them, Figs. 6 b and e show respectively the contact pressure contours at the position of mating teeth coming into or out of contact. Figs. 6 a, c, e and f are the contact pressure contours at the position of normal meshing position of helical gear pair.
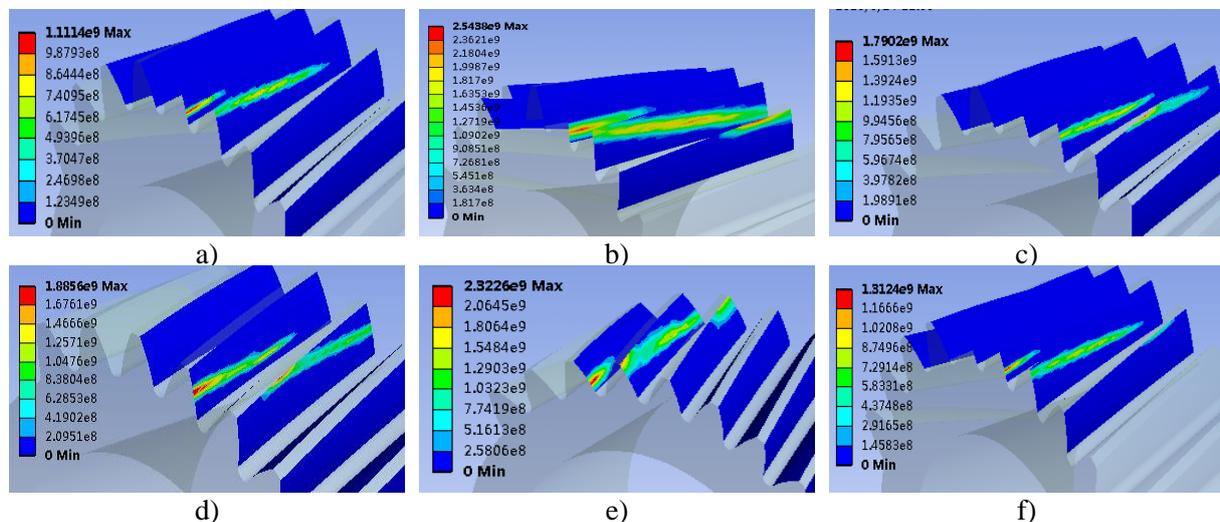

Figure 6: Contact pressure pattern of gear pair at different meshing positions under start-up conditions.

Similarly, contact pressure cloud diagram can be solved for other conditions, the contact stress data of the meshing position of gear pair in the three working conditions are listed in Table V.

Table V: The maximum contact stress at different meshing positions and working conditions.

	Fig. 6 a (MPa)	Fig. 6 b (MPa)	Fig. 6 c (MPa)	Fig. 6 d (MPa)	Fig. 6 e (MPa)	Fig. 6 f (MPa))
Meshing positions	meshing	mesh into	meshing	meshing	mesh out	meshing
Start-up	1111.4	2543.8	1790.2	1885.6	2322.6	1312.4
Continuous	1180.2	3616.4	1918.5	1320.9	3469.9	1620.1
High-speed	2147.9	4771.7	3561.2	2060.5	4559.4	2128.1

Through analysing the solution, we can get the following conclusions:

(1) The contact path of the helical gear pair during meshing is an approximate oblique line along the tooth surface, and the contact stress is non-uniformly distributed along the contact line. The number of meshing teeth is more than or equal to 2 under the three working conditions. It shows that the meshing coincidence degree of the traction helical gear in EMU CRH380A is more than 2, which is the same as the theory of helical gear transmission characteristics, and verifies the reliability of the simulation results.

(2) Comparing the maximum contact stress of gear pair in the three working conditions, with the increase of rotational speed, the maximum contact stress of the gear pair gradually increases, and the contact stress of the gear pair under high speed condition is the highest. The reason is that with the increase of speed, the gear pair will not only produce greater contact pressure, gear teeth also produce greater heat friction, the gears have a certain deformation, so as to the gear contact pressure increases.

(3) Under the three conditions, the maximum contact stress is higher in the positions of meshing into and out than that of the other meshing position under the same condition, and the contact stress of meshing into is the maximum in all positions. As meshing into and out, the tooth is subjected to a large elastic deformation, and the gears can't mesh properly. Therefore it produces too much meshing impact, makes a sharp rise in the contact stress, and meshes into impact effect on the contact stress is larger than the effect of meshes out.

5. DESIGN OF GEAR MODIFICATION PARAMETERS

In view of the above dynamic contact analysis, considering that tooth profile modification can reduce the impact vibration and dynamic load of tooth, and the tooth root modification is easy to reduce the bending strength of the gear which threaten EMU traffic safety, so only large and small gear modify their teeth top. The modification parameters was precisely designed for them by combining with the three parts and elements of gear modification, such as the amount of tooth profile modification, modification curve and the length of modification.

5.1 Gear profile modification

Gear meshing impact is result of the gear deformation making the actual base section is not equal to the actual base section for the driving gear. Therefore, the theoretical modification should be equal to the maximum deformation of the gear at the meshing moment. From the gear contact stress analysis, in the high-speed condition, the contact stress of the gear meshing is the largest. Therefore, the maximum deformation amount of the gear in the high-speed working condition is used as the modification amount. The work of the driving gear is divided into two parts: one part is the drive driven gear movement, the other part is the power consumption in the gear clearance impact. In order to avoid inaccurate result of the clearance

error, when solving the pinion addendum deformation, the pinion is fully constrained and torque is applied to the large gear. On the contrary, for large gear modification amount, the big gear is fixed, and torque is applied to the pinion (Fig. 7). According to the meshing principle, in the Pro/E software, the gear pair is adjusted to the pinion just in the engaged position. The model was imported into the Workbench for calculation (Fig. 8). The maximum deformation of the pinion gear surface was 0.02 mm. So the modification of the pinion was 0.02 mm. By the same logic, the amount of tooth top modification for big gear was 0.023 mm.
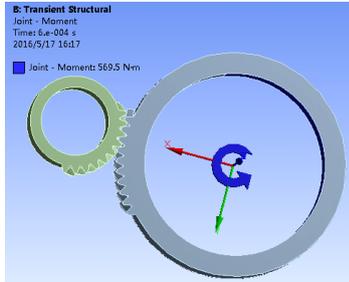
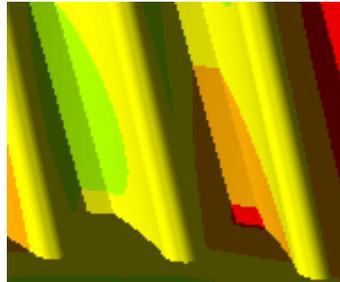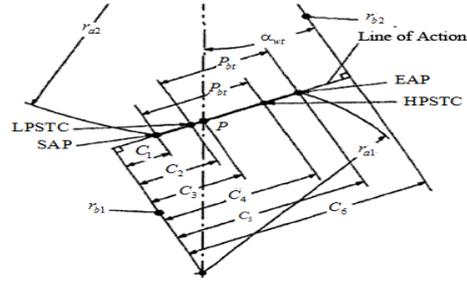| Figure 7: Set the gear load and constraints. | Figure 8: Tooth surface deformation of pinion. | Figure 9: Meshing line diagram. |

5.2 The length of the modification

Gear modification has two types: long and short, due to the relatively large degree of overlap, helical gears usually choose long-shaped modification. Fig. 9 is a graph of the meshing of the two gears on the end face. When the gear is driving, contact point is for the continuous movement in the meshing line. The gear meshing starts at point SAP and stops at the EAP point. The single pair teeth mesh interval is from HPSTC to LPSTC, double pairs teeth mesh intervals are from EAP to HPSTC and from LPSTC to SAP.

From the gear parameters and Fig. 9, the following equation is obtained:

$$C_1 = C_6 - \sqrt{r_{a2}^2 - r_{b2}^2} \tag{2}$$

$$C_2 = C_5 - P_{bt} \tag{3}$$

$$C_3 = r_{b1} \tan \alpha_{wt} \tag{4}$$

$$C_4 = C_1 + p_{bt} \tag{5}$$

$$C_5 = \sqrt{r_{a1}^2 - r_{b1}^2} \tag{6}$$

$$C_6 = (r_{b1} + r_{b2}) \tan \alpha_{wt} \tag{7}$$

$$C_5 - C_1 = \varepsilon_r p_{bt} \tag{8}$$

The specific meanings of the parameters are as follow. P_{bt}: the pitch of the base circle end face; ε_r: the contact ratio of helical gears; α_{wt}: the meshing angle of the end surface; r_{b1} and r_{b2}: radius of base circle; C_6: theoretical meshing length; C_5: the exit meshing point length; C_1: the length from entering the meshing point; $C_5 - C_1$: the actual length of meshing line; r_{a1} and r_{a2}: radius of addendum circle.

5.3 The modification curve design

The tooth profile modification types include straight line modification and curve modification. When the curve is modified, the modification curve is tangent to tooth profile involute, and the modification force of the modified tooth changes a little, and the modification effect is good. Power function modification curve can be expressed as:

$$\Delta = \Delta_{max} (\frac{x}{l})^b \tag{9}$$

In the equation, l is the length of the modification, x is the tooth meshing position, Δ is the amount of modification corresponding the mesh position, b value is usually in the range from 1.0 to 2.0. When b equals 2, the modification curve is parabola which relatively smooth with the profile transition, and the meshing impact is small. In this paper, parabola shape modification is used.

6. THE ESTABLISHMENT OF MODIFIED HELICAL GEAR MODEL

6.1 Create the modification curve

$P_{bt} = 20.771$ mm. For Eqs. (2) to (8), to substitute the relevant parameters of the gear into them, the equations for the radius of addendum modification initial circle can be obtained for the driving gear and the driven gear, as shown in Eqs. (10) and (11).

$$r_{t1} = \sqrt{C_4{}^2 + r_{b1}{}^2} \tag{10}$$

$$r_{t2} = \sqrt{(C_6{}^2 - C_2)^2 + r_{b1}{}^2} \tag{11}$$

It is calculated that the radius of the initial circle for the driving gear is 110.77 mm and the radius of the initial circle for the driven gear is 259.68 mm. Taking the following pinion as an example, in the Pro/E curve fitting modification, the specific process is as follows:

Create a modification terminal point

As the maximum tooth tip modification is the distance from modification terminal point to the profile end point on gear end face. Taking the profile end point on gear end face as a reference, the maximum modification amount is offset to create datum point.

Create a modification starting point

The starting point of the modification is the intersection point between the initial circle of the gear modification and the involute of the tooth profile. So the initial circle should be firstly sketched. The datum point is established at the intersection point between the involute of the gear tooth profile and the initial circle of the modification.

Create a modified coordinate system

In order to ensure that the modification curve and the tooth profile transition are relatively smooth, the modification curve should be tangent to the tooth profile involute line at the beginning of the modification. Therefore, the coordinate system created at the beginning of the modification should also be tangent to the involute profile. The process is as follows. To create a datum plane BTM14 which is via the starting point and perpendicular to the involute; to create a datum plane BTM on the end face of the tooth profile, and create a datum BTM16 which is perpendicular to the datum planes BTM15 and BTM14 as well as passes the datum point PNT5, so the coordinate system CS5 is created with BTM14, BTM15, BTM16.

Create a modification curve

To create curve by clicking on the "Curve" button and select "From equation", then select the coordinate system CS5 and "Cartesian" type, input the following Eqs. (12) to (14) for the pinion or Eqs. (15) to (17) for large gear. The build curves are shown in Figs. 10 and 11.
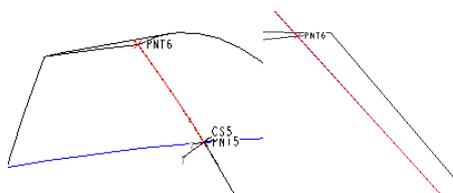

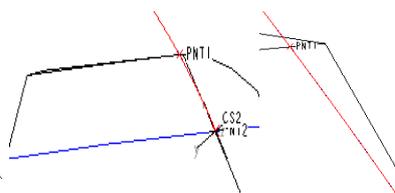Figure 10: The pinion modification curve.
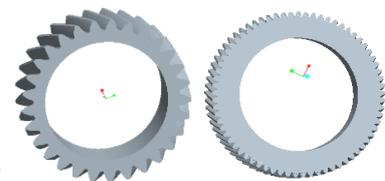
Figure 11: Large gear modification curve.

Figure 12: The two modified gear model.

The pinion modification curve equation:

$$x = 5.2 * t \tag{12}$$

$$y = 0.118 * (5.2 * t / 3.5)^2 \tag{13}$$

$$z = 0 \tag{14}$$

Large gear modification curve equation:

$$x = 5.2 * t \tag{15}$$

$$y = 0.213 * (5.2 * t / 3)^2 \tag{16}$$

$$z = 0 \tag{17}$$

6.2 Create a modified gear model

To take those modified curves as reference sketches, create modification surface and tooth. By rotating and array the tooth, modify all gear teeth of helical gear, the modified helical gear model can be gotten, shown as in Fig. 12.

7. ANALYSIS AND COMPARISON OF MODIFICATION EFFECT

The specific operation steps of the modified gear finite element analysis are the similar as before modification, only need to replace the unmodified gear model with modified gear model, so the details don't display. The equivalent contact stress distribution of the gear tooth under each working condition is shown in Table VI.

Table VI: The maximum stress in the different meshing positions and working conditions.

	Fig. 6 a (MPa)	Fig. 6 b (MPa)	Fig. 6 c (MPa)	Fig. 6 d (MPa)	Fig. 6 e (MPa)	Fig. 6 f (MPa)
Meshing positions	meshing	mesh into	meshing	meshing	mesh out	meshing
Start-up	990.86	2226.2	1987.7	1693.9	2062.9	1197.4
Continuous	1722	3001.9	2135.4	1061.5	2958.8	1503.9
High-speed	1813.8	3878.3	2920.6	2352.4	3743.5	1625.7

Comparing Table V and Table VI, the following conclusions can be drawn:

The maximum contact stress of the gear mesh into and mesh out is reduced. The maximum contact stresses as the gear mesh into are respectively decreased by 12.49 %, 16.99 % and 18.72 % under start-up, continuous and high-speed working conditions. The maximum contact stress as the gear mesh out are respectively are decreased by 12.49 %, 16.99 % and 18.72 % under the three working conditions. By optimizing the three parts and elements of gear modification, the maximum contact stress of the gears is reduced and the meshing impact is decreased and the gear transmission performance is improved.

In this paper, the traction helical gear pair of high-speed EMU CRH380A is taken as the research object, and the dynamic contact analysis of the traction helical gear pair under the three conditions of start-up, continuous and high-speed is carried out by ANSYS Workbench. According to the results of the solution, the gears were modified by the three parts and elements of tooth profile design, the dynamic contact analysis of the modified gear pair was done again, compared the maximum contact stress of the gear pair before and after modification, the gear contact state is effectively improved, which reduces the impact of the gear in running process and provides the theoretical basis for the practical modification design of gears.

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