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Tooth Surface Contact Analysis for Logarithmic Spiral Bevel Gear based on ANSYS

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Abstract: The nonlinear static contact analysis is carried out at different positions in the spiral bevel gear transmission process, and the stress variation law of the contact area is obtained; the maximum contact stress is obtained in ANSYS, and the results are compared with the theoretical values. The accuracy of the contact problem is verified by the finite element model method. At the same time, the dynamic contact conditions of this kind of gears at different meshing moments are analyzed, and the speed influence on the contact characteristics is compared. Determine whether the contact strength of the spiral bevel gear meets the requirements by comparing the change law of contact stress at different static and dynamic angles.

Keywords: Spiral bevel gear; Kinetics analysis; Static contact analysis; Dynamic contact analysis

1. Introduction

The spiral bevel gear pair is the basic component for realizing the transmission of the intersecting axis. Due to its advantages of large coincidence degree, high transmission capacity and high bearing capacity, it is widely used in various machinery and equipment such as automobile, construction machinery, helicopter, machine tool and so on. The manufacturing precision and quality of spiral bevel gears directly affect the efficiency, noise, motion accuracy and life of these devices. However, the geometrical characteristics and meshing process of spiral bevel gears are very complicated. At the same time, various factors such as machining tool, machine parameter setting, loading deformation and assembly error will cause the bearing load and vibration performance changes. Therefore, the contact fringe, the transmission error, the bending stress on the tooth surface and the distribution of the contact stress on the tooth surface have been the research and attention of scholars from all over the world, and become the key technology and commanding point in gear production [1].

Since the previous contact simulations are mostly based on fuzzy or approximate finite element models, the analysis results have inevitable errors and are not useful for practical application. Therefore, based on the three-dimensional model of the precise spiral bevel gear, the ANSYS software is used to analyze the tooth surface of the bevel gear statically and dynamically, and the accurate contact stress and stress variation law are obtained.

2. Static Contact Analysis for Spiral Bevel Gear

Based on the finite element software ANSYS Workbench platform, the static contact analysis of the spiral bevel gear, the main steps include five parts: Model simplification, meshing, contact definition, application of boundary conditions and loads, solution and post-processing. In the static contact analysis, only a few pairs of gear teeth are in the meshing contact state, and the gears that are not in contact have no contact stress, so in the case of neglecting the influence of other gear teeth of the large gear on the comprehensive stiffness matrix, the simplified one is selected. The six-tooth model is the research object, and static contact analysis is performed on a pair of meshed spiral bevel gears. At the same time, four different meshing positions in the meshing period are selected, and the angular variables with respect to the initial position are 0°, 0.72, 1.44° and 2.160, respectively.

2.1. Establishment of finite element model

In this example, the material of the meshing gear is 20CrMnTi, the elastic modulus $E = 2.0675\times10^5$ MPa, the Poisson's ratio μ=0.3, and the density is $7.85\times10^3 kg/m^3$.

For 3D models, ANSYS Mesh has several different meshing methods. The degree of density of the mesh directly affects the accuracy of the solution results, but the dense mesh makes the calculation cost increase. Because the hexahedral dominant mesh is more accurate than the tetrahedral dominant mesh, it takes up more computer resources. In order to obtain more accurate calculation results, taking into account the complex structural characteristics of the spiral bevel gear and its boundary conditions and loads, the mixed tetrahedral and hexahedral meshes are used in this example, and the gear contact surface of the meshing gear is partially refined. Similarly, in order to reduce the amount of calculation, only the six-tooth model of the driven wheel is used to solve the problem to ensure the accuracy of the calculation. The meshing model of the spiral bevel gear contact is shown in Figure 1.

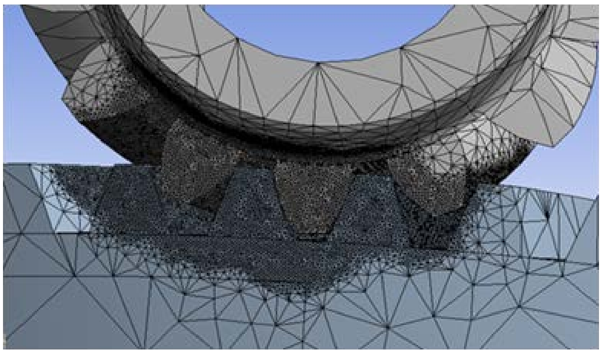


Figure 1. Contact area mesh refinement

When using ANSYS to make contact analysis of the meshing gear, it is necessary to define the contact area of the gear. Due to the complexity of the helical bevel gear meshing tooth surface, this paper adopts the automatic contact method in the surface-surface contact to define the contact pair and judge and solve the contact surface with defined contact relationship. The tooth surface of the contact is not defined, and even if the tooth surface contact is mutually infiltrated, the computer does not make a contact calculation. In this example, the driving wheel is defined as the contact surface, the driven wheel is defined as the target surface, and the defined contact pair is shown in the blue and red display areas of Figure 2. For the nonlinear calculation of this part of the contact, a penalty function is used in the Workbench platform to calculate the nonlinear contact. In the definition of contact pairs, some main options need to be set.

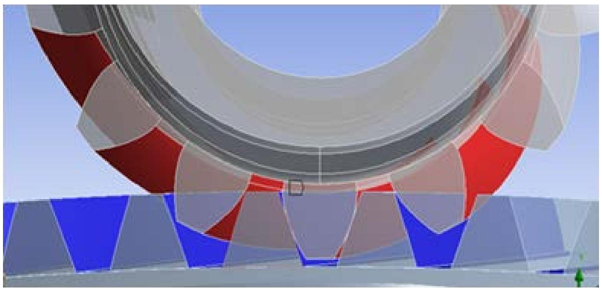


Figure 2. Contact pair definition

All contact problems must define contact stiffness, including normal and tangential stiffness, and normal stiffness K_N is a key factor for excessive penetration tolerance and affecting ill conditions. Considering the problem of computational convergence, etc., in this example, the normal stiffness factor FKN=1 is taken.
FTOLN is the maximum penetration tolerance, and penetration above this value will do a new iterative calculation. The penetration tolerance value depends on the contact stiffness, which is too small to make calculations difficult to converge. Therefore, the maximum penetration tolerance selected in this paper is FTOLN=0.1.
Considering the influence of the friction factor, the friction coefficient between the contact faces of 20CrMnTi carburized steel gears is 0.06 at the normal temperature of the penalty coefficient [2].

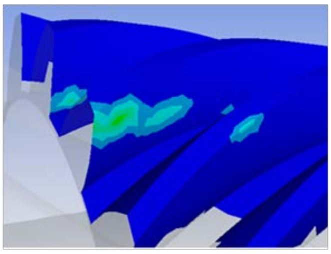
2.2. Boundary conditions and loads

Fix all the nodes on the inner ring surface of the large spiral bevel gear, and fix the radial and axial degrees of freedom of the control nodes of the small gear inner ring, so that the circumferential freedom of rotation around the axis is free. Assume that the gear is driven by a motor with a power of 500 kW. For the high-speed transmission gear in this example, the pinion spindle speed is 10000 r/min, and the calculated driving torque is T=300 N•m. Turn on the large deformation switch that solves the settings to prevent computational convergence difficulties.

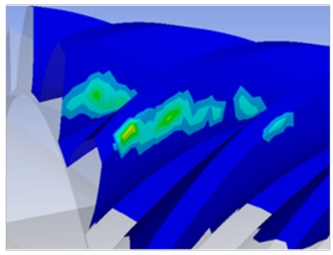
2.3. Solution and post processing

In order to obtain the distribution of the tooth surface contact stress during the meshing process of the spiral bevel gear, the meshing models under different corners of the gear under static state are analyzed separately. In the meshing process of the spiral bevel gear, the angle of the rotation angle ranges from 0 to 2.16°, and the meshing model of a spiral bevel gear transmission is taken every 0.72°. The contact stress at different positions obtained by the spiral bevel gear meshing model is shown in (a), (b), (c), (d) of Figure 3. Through analysis and solution, the stress and strain distribution of the gear contact surface at different corner positions and the stress and strain values of each node in different directions can be obtained.
The above method of taking finite element contact analysis calculations from several locations has the following two characteristics:
The analysis model of each meshing position can be realized by the node rotation constraint of the driving wheel, so this method can realize the contact analysis at any instantaneous position in the gear meshing period.

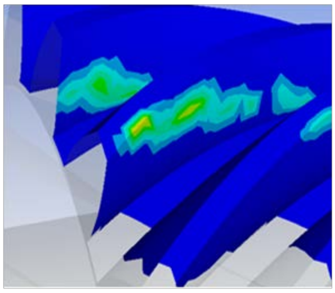
It is not necessary to preset the distribution of the contact load, because in the meshing contact state, ANSYS can automatically capture and calculate the contact area and the contact load distribution.



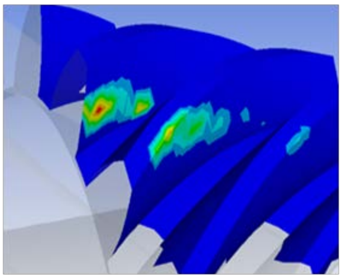
(a) Gear contact stress cloud diagram with a corner angle of 0°



(b) Gear contact stress cloud diagram with a corner angle of 0.72°



(c) Gear contact stress cloud diagram



(d) Gear contact stress cloud diagramwith a corner angle of 1.44° with a corner angle of 2.16

Figure 3. Tooth surface contact stress cloud diagram during meshing period

It can be seen from the figure that during the meshing process, the gear begins to gradually enter the meshing, the contact area gradually increases during the meshing process, and the contact stress on the tooth surface is gradually increased. The gear teeth are always in the meshing state, and the contact stress region is gradually increased, and the contact stress value is gradually increased. Observing the contact spot area, it can be found that each contact spot area is linear in distribution. At the various stages of gear meshing, edge contact occurs near the tooth tip, which adversely affects the spiral bevel gear transmission under high speed and heavy load conditions.

The solution results of the static contact analysis in the gear meshing transmission period indicate that the trajectory of the spiral bevel gear in the contact spot region is substantially linear in the direction. In the gear meshing process, the spiral bevel gear transmission is a contact that extends from the point contact to the area, and is obliquely linearly distributed from the small end to the big end and toward the tooth tip.

During the meshing process of the spiral bevel gear, almost two pairs of gear teeth are in contact state, so the spiral bevel gear has high bearing capacity, stable transmission and high transmission efficiency. During the biting process, the contact spot area gradually increases and the maximum contact stress increases. Edge contact of the spiral bevel gear tooth surface affects the stability of the gear transmission During the gear engagement process, the area of the meshing contact spots is gradually reduced, and the maximum contact stress is also gradually reduced. In summary, during a meshing cycle, the contact stress on the tooth surface of the spiral bevel increases first and then decreases.

Through the processor solution, the stress, strain, contact state and contact stress of the assembly can be visually observed. Figure 4 shows the gear contact equivalent stress cloud at the initial position. It can be seen that the maximum equivalent contact stress value is 871.1 Mpa.

Figure 4. Initial position contact stress cloud

The position of the equivalent stress cloud of the driving wheel and the position of the maximum equivalent stress can be seen from Figure 5 and Figure 6. The maximum equivalent stress of the slave and the driving wheel are 274.7Mpa and 366.3Mpa, respectively. The maximum equivalent stress is at the edge of the contact line and close to the root of the tooth. At the same time, attention is paid to the stress concentration at the edge of the tooth. The equivalent stress of the driving wheel of the intermeshing transmission is larger than that of the driven wheel. This is because in the static contact analysis, the driven wheel is fully constrained, and the moment is loaded on the driving wheel during the analysis. At the moment of contact with the initial state, the driving wheel contact impact is greater than the driven wheel.

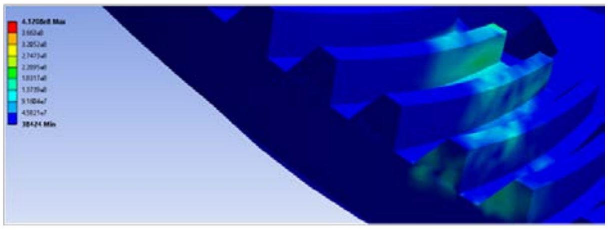


Figure 5. Driven wheel equivalent stress cloud

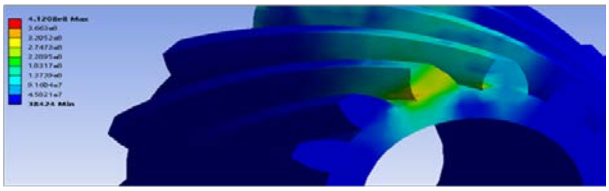


Figure 6. Driving wheel equivalent stress cloud

3. Dynamic Contact Analysis for Spiral Bevel Gear

Since the load is transmitted from tooth to tooth during the gear transmission process, the single and double teeth alternately mesh during the transmission process, causing the vibration and impact of the gear meshing force and the fluctuation of the load and the rotational speed, resulting in a cyclical change of the meshing stiffness. At the same time, nonlinear factors such as friction between the meshing surfaces are present, It is only after a number of static contacts at different meshing positions to calculate the contact stress, and only considers a transient state of the meshing process, and can not accurately and comprehensively reflect the mechanical characteristics of the gear during the transmission process. Since the working environment of spiral bevel gears is often in high-speed heavy-duty applications, in order to simulate the continuous transmission process of the gears, the contact stress, strain and other changes in the whole cycle process with time, it is necessary to carry out dynamic contact analysis of the spiral bevel gears [3].
Considering the influence of speed and other factors, the finite element model of spiral bevel gear contact analysis was established in ANSYS Workbench, and the transient dynamics analysis was carried out to obtain the real-time variation law of dynamic contact stress of the transmission process. At the same time, high and medium were also analyzed. And the stress variation law of dynamic meshing contact under three different working conditions, which provides reference for gear design [4].

3.1. Dynamic contact analysis steps and processes

In ANSYS Workbench, the steps and processes of dynamic contact analysis are similar to static analysis, with only the analysis type, related parameter settings, and different load boundary conditions. Among them, the analysis type is set to Transient. Transient dynamics analysis includes both linear transient dynamics analysis and nonlinear transient dynamics analysis.
The definition of contact with static analysis is basically the same. In this section, the large gear tooth surface is defined as the target surface, and the pinion tooth surface is defined as the contact surface, and the contact pair is identified by an automatic method. Other settings refer to static contact analysis.
During the dynamic meshing of the gear, the motor applies an angular velocity about the axis rotation to the driving wheel through the coupling and the shaft. At the same time, the driven wheel rotates around the central axis through the contact of the main and driven wheels, and the driven wheel reaches the force balance under the action of input torque and load resistance torque. Among them, insert the "Joint" rotation pair for the main and driven wheels, set "Connection" to "Body-Ground", and select "Revolute" for the hinge type. The angular velocity of the driving wheel about the central axis is 1000 rad/s, and the resisting torque of the driven wheel is 120 N. m [5].
In the transient analysis setting, open the large deformation switch, set the end time of the load step to 0.0013s, turn off the automatic time step switch, set the load step to 30, and use the sparse matrix solver to solve the transient contact. Figure 7 shows the force convergence curve during the calculation of the solution. It can be seen from the periodic oscillation of the curve around the curve that the nonlinear contact analysis solution is convergent. In order to analyze the contact situation at different times of the meshing process, the contact state diagram of the gear assembly at six different times in the high speed state is selected. The equivalent stress cloud diagram is shown in Figure 8.
The maximum contact stress at each position at different times in Figure 8 is selected, and the stress results are shown in Table 1.
Analysis of Figure 8 and Table 1 shows that when the gear is in a high-speed rotation state, the maximum contact stress of the gear meshing and meshing is greater than the contact stress of the single tooth meshing. This is because the biting and engaging points deviate from the theoretical gear to bite and bite the position, causing a bite and a biting impact. The maximum contact stress of the double tooth mesh is significantly smaller than that of the single tooth mesh, this is because the load is

shared by a plurality of gear teeth, and the contact area is increased, the comprehensive meshing rigidity is also increased, the meshing contact becomes elastic, and the solid contact stress is reduced.

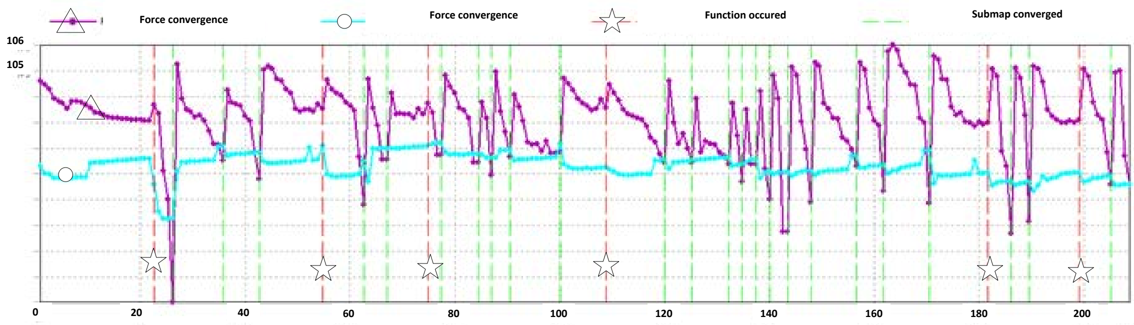


Figure 7. Force convergence curve

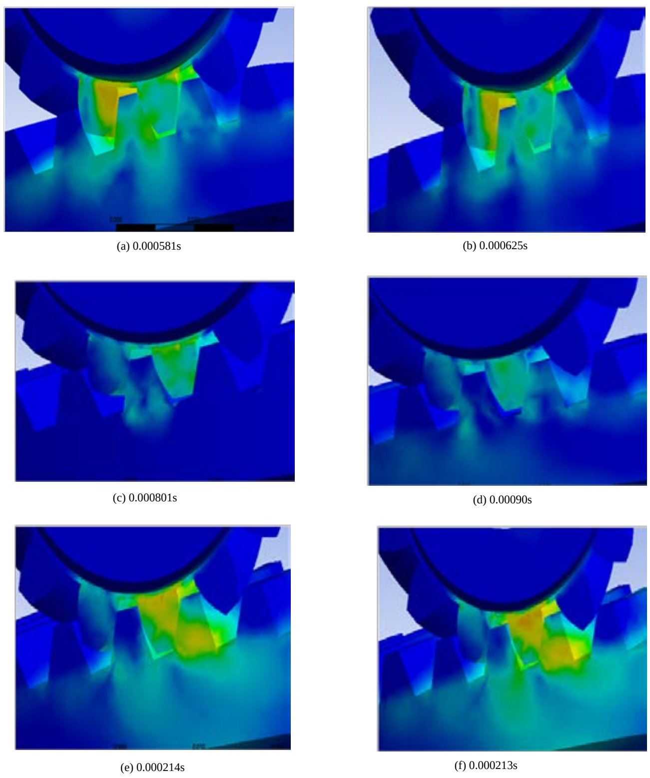


Figure 8. Equivalent stress cloud at different times

Table 1. Maximum contact stress at different meshing positions of gears

Drawing number	**A**	**B**	**C**	**D**	**E**	**F**
Engagement position	Single tooth meshing	Hitting	Double tooth engagement	Double tooth engagement	Knock out	Single tooth meshing
Maximum contact stress(Mpa）	1132	1240	859	642	1093	1045

3.2. Effect of speed on dynamic contact

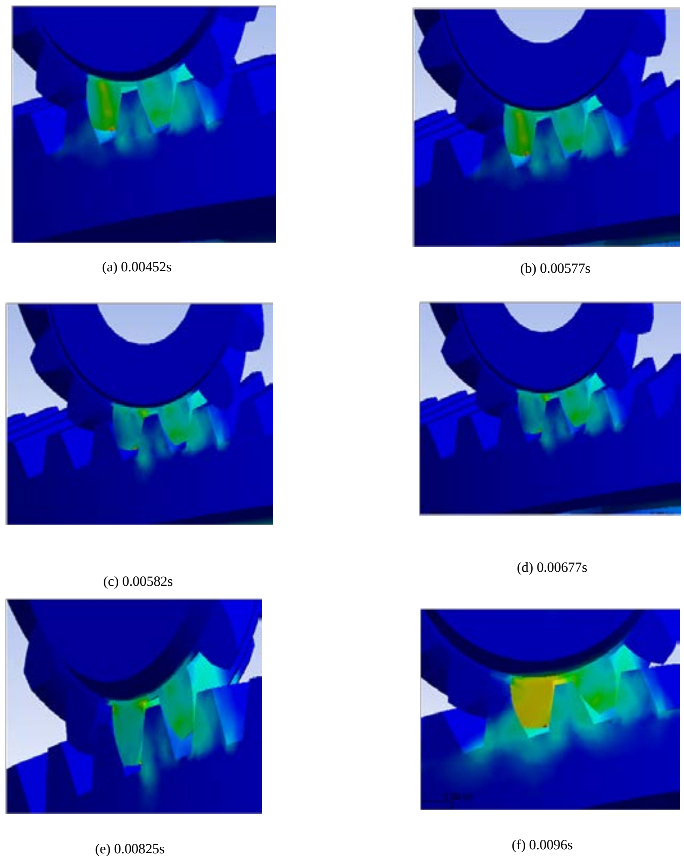
In order to study the effects of different speeds on the dynamic contact stress of gears under different working conditions, this section selects three representative speeds of high, medium and low. The ANSYS Workbench solves the transient dynamics of the meshing gears. The load boundary conditions are shown in Table 2. The working condition 1 is a high speed situation, which has been solved in the front.

Table 2. Load condition

Working condition	Angular velocity(rad/s)	Load torque(N.M)
1	1000	120
2	100	120
3	1	120

In the working condition 2, according to the engagement period corresponding to the rotation speed, the end time of the load step is set to 0.01 s, the automatic time step switch is turned off, the load step is set to 40, and the meshing positions of the gears at different times in the middle speed are extracted. The equivalent stress cloud diagram is shown in Figure 9.

The maximum contact stress at each meshing position at different times in Figure 9 is selected, and the stress results are shown in Table 3.



(a) 0.00452s (b) 0.00577s (c) 0.00582s (d) 0.00677s (e) 0.00825s (f) 0.0096s

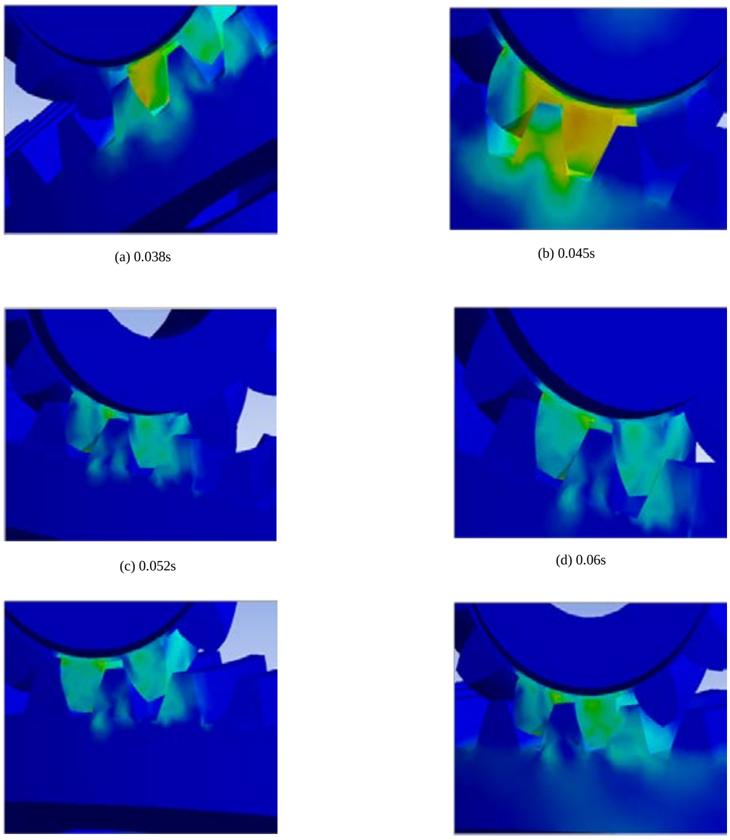
Figure 9. Working condition 2 equivalent stress cloud diagram at different times

Table 3. Working condition 2 maximum contact stress at different meshing positions

Drawing number	A	B	C	D	E	F
Engagement position	Single tooth meshing	Hitting	Double tooth engagement	Double tooth engagement	Knock out	Single tooth meshing
Maximum contact stress (Mpa)	481	426	319	332	411	452

It can be seen from analysis of Figure 9 and Table 3 that the maximum contact stress of the gear meshing and the engagement is slightly larger than the contact stress when the single tooth is meshed, and is also close to the single tooth engagement. It shows that with the decrease of the rotational speed, the gear slowly enters the stationary period of the transmission, and its load deformation also changes little, and the gear is less affected by the meshing impact.

Similarly, for the case of working condition 3, set the end step of the load step to 0.08 s, turn off the automatic time step switch, and set the load step to 40 steps. The equivalent stress cloud diagram of the meshing positions of the gears at different times at low speed is shown in Figure 10.



(a) 0.038s (b) 0.045s

(c) 0.052s (d) 0.06s

(f) 0.08s

Figure 10. Working condition 3 equivalent stress cloud diagram at different times

Table 4. Working condition 3 maximum contact stress at different meshing positions

Drawing number	A	B	C	D	E	F
Engagement position	Single tooth meshing	Hitting	Double tooth engagement	Double tooth engagement	Knock out	Single tooth meshing
Maximum contact stress(Mpa）	265	173	206	213	168	254

4. Comparison between Static and Dynamic Contact Analysis

Selecting the same load torque for static and dynamic contact, can be observed from the equivalent stress cloud of the two, The contact lines of static and dynamic contact are straight lines along the small end of the spiral bevel gear to the big end and inclined toward the tooth tip, and the maximum equivalent stress is at the edge of the contact line and close to the root of the tooth. Dynamic contact stress is greater than static under the same load conditions. This is due to the load deformation of the gear during the meshing process and the large meshing impact under the internal excitation. The biting and the pinning point deviate from the theoretical position, it makes the meshing gears mesh out, causing periodic vibration and shock. Compared with static contact analysis, dynamic analysis can easily solve the change of contact stress at the meshing position at all times of the whole process of meshing in one algorithm, and is closer to the actual working condition [6].

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