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Study of Orthogonal Face Gear's Contact Spots and Transmission Errors

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Abstract: By single-gear pair contact analysis, this paper, based on the Hertz theory, gets the mesh density of finite element solver in convergence case and then studies contact spots with multi-gear pairs. Through simulation analysis, this paper studies the face gear transmission error in different conditions, and this paper studies the face gear transmission error in different conditions. Then come to a conclusion that: In a set of three different load cases, the greater the torque load is applied to the face gear, the greater the transmission error there is, but the speed of face gear almost has no effect on the gear transmission error. When the face gear is installed with shaft angle of 0.02°, its transmission error magnitude will be 2.0~2.3 times bigger compared with installed without any error. The study can provide guidance for the design, production and practical application of the face gear. *Copyright © 2013 IFSA.*

Keywords: Face gear, Simulation analysis, Contact spot, Transmission error.

1. Introduction

The tooth contact and transmission errors of face gear are related to the vibration, noise and transmission precision of face gear's transmission process. Because face gear is becoming more and more important in the use of aviation industry and precision mechanical transmission in recent years, face gear with quality of low noise, small vibration, stable transmission and high transmission precision is widely required. So the research of the tooth contact and transmission error has very important significance.

Many scholars have done a lot of study about face gear. Claudio Zanzi [1] modified the profile of the pinion of face gear pair and did the analysis of tooth contact and tooth root bending stress. Litvin [2] studied the face gear contact stress and bending stress by finite element method. The experts of face gear are mainly focused on face gear processing, modification, surface contact stress and face gear strength. But they haven't studied tooth contact and transmission errors of face gear deeply. This paper studies contact sport's coordinates of face gear and its realization of visualization through comparing the theory and finite element simulation, gear transmission error under standard installation in different working conditions, but also studies the transmission error of face gear in different coaxial angle installation error. This paper provides guidance for face gear design, noise reduction etc. So the study of this paper has practical significance

2. Mathematical Modeling and Working Load of Orthogonal Face Gear

The tooth working area of face gear is generated by involute tooth profile of the cylindrical gear slotting tool. Transition curved surface is generated by the top edge of Slotting tool. Tooth fillet of face gear is decided by the tip structure of slotting knife. Tooth surface equation of face gear [5] is as (1) shows:

$$\begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{bmatrix} r_{bs}\cos\phi_2\cos\gamma(\cos\phi_0+\theta_s\sin\phi_0)-r_{bs}\sin\phi_2(\sin\phi_0-\theta_s\cos\phi_0) \\ -\dfrac{r_{bs}(i_{s2}+\cos\gamma)\cos\phi_2}{\cos\phi_0} \\ r_{bs}\sin\phi_2\cos\gamma(\cos\phi_0+\theta_s\sin\phi_0)+r_{bs}\cos\phi_2(\sin\phi_0-\theta_s\cos\phi_0) \\ -\dfrac{r_{bs}(i_{s2}+\cos\gamma)\sin\phi_2}{\cos\phi_0} \\ r_{bs}\sin\gamma(\cos\phi_0+\theta_s\sin\phi_0)+\dfrac{r_{bs}(i_{s2}+\cos\gamma)\cos\gamma}{\sin\gamma\cos\phi_0} \end{bmatrix} \quad (1)$$

Because two tooth surface of a face gear tooth are symmetrical about surface yoz, so when change the sign of θ_s, the mathematical model of the other side of the face gear tooth can be got. The equation is as (2) shows:

$$\begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{bmatrix} r_{bs}\cos\phi_2\cos\gamma(\cos\phi_0-\theta_s\sin\phi_0)-r_{bs}\sin\phi_2(\sin\phi_0+\theta_s\cos\phi_0) \\ -\dfrac{r_{bs}(i_{s2}+\cos\gamma)\cos\phi_2}{\cos\phi_0} \\ r_{bs}\sin\phi_2\cos\gamma(\cos\phi_0-\theta_s\sin\phi_0)+r_{bs}\cos\phi_2(\sin\phi_0+\theta_s\cos\phi_0) \\ -\dfrac{r_{bs}(i_{s2}+\cos\gamma)\sin\phi_2}{\cos\phi_0} \\ r_{bs}\sin\gamma(\cos\phi_0-\theta_s\sin\phi_0)+\dfrac{r_{bs}(i_{s2}+\cos\gamma)\cos\gamma}{\sin\gamma\cos\phi_0} \end{bmatrix} \quad (2)$$

In the above formulas (1), (2), x, y, z are coordinate points of the tooth surface, r_{bs} is the tools' radius of base circle, φ_2 is turned angle of the face gear, r = 90°, i_{s2} is the transmission ratio of the face gear and the tool, φ_s is turned angle of the tool, φ_{s0} is the angle between tool tank symmetrical line and involute starting point, θ_s is the angle between involute starting point and generating line, and $\varphi_0 = \varphi_s + \theta_s + \theta_{s0}$

Using the software of MATLAB calculates tooth surface discrete coordinate data of face gear and put the data into Pro/E to get the figure of tooth surface distribution points. It is shown in the following Fig. 1.

Because the data of points is obtained from the tooth surface equation, so the graphics can well reflect the tooth surface profile of face gear. By using Pro/E, connect points into lines, process lines into surfaces, and finally get single tooth model of face gear through substantiated operation. Use Pro/E software to model the involute spur pinion, and assemble face gear with Involute cylindrical pinion through Pro/E software, Specific assembly model are as shown in Fig. 2.

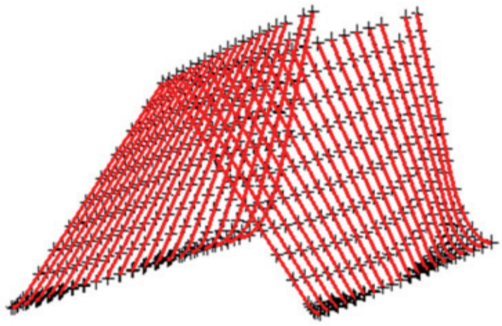


Fig. 1. Tooth surface discrete points of face gear.

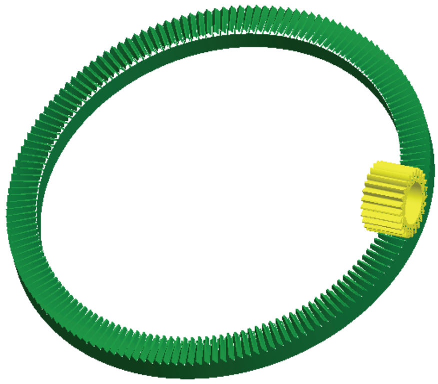


Fig. 2. Assembly drawing of face gear.

Using simulation technology to study face gear transmission error, we apply the speed of: 100 r/min, 200 r/min, 400 r/min to the pinion, and apply the resistance moment of: light load: 5E5 $n \cdot mm$

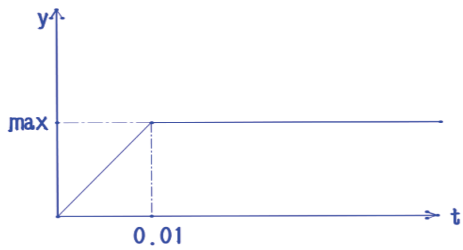


Fig. 3. Loading curve.

In Fig. 3, abscissa refers to the loading time which unit is seconds, ordinate refers to the speed of active pinion or the torque applied to large gear. As the graph shows that, from 0 s to 0.01 s [6], loadings linearly increase, after 0.01 s loadings reach maximum and remain unchanged. Considering the solving characteristics of the software ABAQUS, we choose such a method to apply loads to avoid the non-convergence problems caused by applying loads too large at first. Through changing the speed of pinion and the torque applied to large gear, this paper studies the transmission error when the speed of the pinion is 200 r/min, the resistance moment of large gear is different, and when pinion is in the above three different speeds, the resistance moment of large gear is the same. The basic parameters of orthogonal face gear are shown in Table 1. Because the contact spots can reflect the face gear's loading capacity and transmission accuracy, it is necessary to study the contact spots of face gear before the study of its transmission error.

Table 1. Transmission parameters of orthogonal face gear.

Name	Number of teeth	Width (mm)	Pressure Angle (°)	Modulus (mm)	Inner diameter (mm)	External diameter (mm)	Shaft Angle (°)	Gearblank thickness (mm)
Face gear	160	89	25	6.35	471	560	90	40
Pinion	25	90	25	6.35	——	——	90	——

3. Study of Face Gear Contact Spots

According to the principle of tooth contact analysis (TCA), two gears are continuous tangential contact in gear meshing transmission. So in the fixed coordinates system of two gear tooth surfaces, the two surfaces have public contact points in random moment. Because they have common normal line at contacting point, solving simultaneous equations of two gear tooth surface can get one of the basic equations of the tooth contact analysis:

$$\begin{aligned} \vec{r}_h^{(1)}\left(\theta_{a1},\phi_{b1},\varphi_1\right) &= \vec{r}_h^{(2)}\left(\theta_{a2},\phi_{b2},\varphi_2\right) \\ \vec{n}_h^{(1)}\left(\theta_{a1},\phi_{b1},\varphi_1\right) &= \vec{n}_h^{(2)}\left(\theta_{a2},\phi_{b2},\varphi_2\right) \end{aligned} \tag{3}$$

In the equations, Superscript "1" and "2" separately refer to cylindrical pinion and face gear, the subscript means that each vector is in the same coordinate. S_h , $(\theta_{a1},\varphi_{b1})$ and $(\theta_{a2},\varphi_{b2})$ refer to the tooth surface coordinates equation of cylindrical pinion and face gear, φ_1 and φ_2 mean the angle turned by the drive gear (pinion) and driven gear (face gear) which both meet the requirements of the transmission ratio. Because of $\left|\vec{n}_1^{(d)}\right| = \left|\vec{n}_2^{(d)}\right| = 1$, Equations (2) can be divided into five scalar equations. A coordinates of contact point of two gear tooth surfaces can be got with fixed value of φ_1 . Then use a certain step length to change the value of φ_1 . Use these values to do an iterative solution until the coordinates of contact point exceed the efficient frontier of face gear's tooth height. These instantaneous contact points of tooth surface form the face gear's contact path. In the actual bearing case, tooth surface of face gear has a certain degree of elastic deformation. At each contact point, we can get the value of instantaneous contact ellipse's short semi-shaft and long semi-shaft. The series of contact ellipse imprints form the tooth surface contact area of face gear. By coordinate transformation of the tooth surface contact equation, we can get three coordinate component formulas of face gear's tooth surface contact trajectory. These formulas are listed below

$$\begin{aligned} \vec{n}_{xf}^{(1)}\left(\phi_1',\theta\right) &= \vec{n}_{xf}^{2}\left(\phi_2',\theta_s,\phi_s\right) \\ \vec{n}_{yf}^{(1)}\left(\phi_1',\theta\right) &= \vec{n}_{yf}^{2}\left(\phi_2',\theta_s,\phi_s\right) \\ \vec{n}_{zf}^{(1)}\left(\phi_1',\theta\right) &= \vec{n}_{zf}^{2}\left(\phi_2',\theta_s,\phi_s\right) \end{aligned} \tag{4}$$

Assume that angle φ_2' is a known parameter. So that, giving a fixed angle φ_2', a group of coordinate data can be got. In this way, we can get a contact point. Use a certain step length to change the value of φ_2'. Then we can work out a series of tooth contact points of face gear. According to the formulas and using software MATLAB, We can get the coordinate values of contact trajectory [4]. As is show in Table 2.

Table2. Contact trajectory coordinates component (mm).

x	*y*	*z*
9.7622	-497.8976	-85.649
8.8386	-497.7834	-83.6832
7.9181	-497.7032	-81.7174
7.3056	-497.6686	-80.4069
6.0825	-497.6447	-77.7858
4.8604	-497.6813	-75.1647
3.6374	-497.7784	-72.5437
2.4116	-497.9359	-69.9226
1.1811	-498.1538	-67.3015
0.254	-498.3568	-65.3357
-0.3665	-498.5109	-64.0252
-0.9893	-498.68	-62.7147
-1.6146	-498.8641	-61.4041
-2.2425	-499.0632	-60.0936

According to Table 2, use software MATLAB to display the coordinate values of tooth surface contact points in the tooth surface. Specific is shown in the following Fig. 4.

According to the L. L ITVIN's book [7], which puts forward the contact ellipse, as is shown in the following Fig. 5.

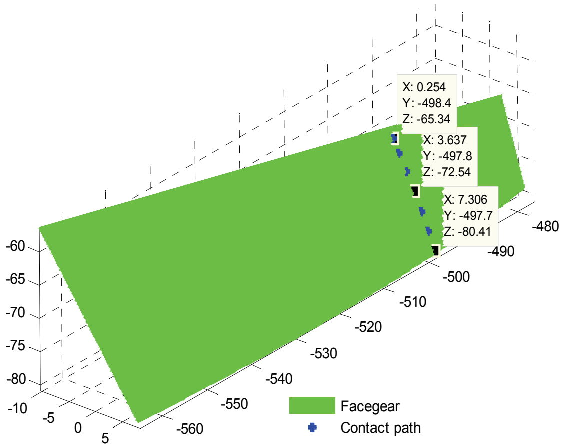


Fig. 4. Contact trajectory.

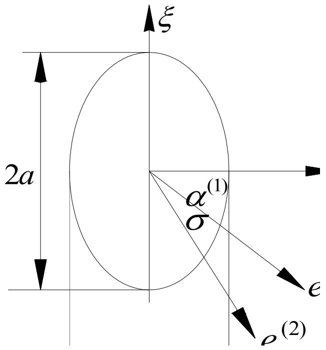


Fig. 5. Contact ellipse.

We can get the length of two half shafts a and b, which are in face gear meshing area.

$$a = \left|\frac{2\delta}{k_a}\right|^{1/2}, \quad b = \left|\frac{2\delta}{k_b}\right|^{1/2}, \tag{5}$$

where δ is the total indentation of the gear contact. Because it is single tooth contact, the load distribution factor is approximated as 1[8] in this paper. To get contact spots, the theoretical calculation of torque is calculated by the following formulas (6) and (7) in the condition that the speed of pinion is 300 r/min and the power of the pinion is 1100 kW.

$$T = \frac{P}{\omega} \tag{6}$$

$$\omega = \frac{2\pi n}{60} \tag{7}$$

In the above formulas, n is speed of the pinion, ω is the angular velocity, P is power. After solving the formulas, the value of torque is 350.14 $n \cdot m$.

k_a, k_b are induced normal curvature of long and short half shaft direction in the meshing area. Get the direction of principal curvatures in the tooth surface contact area and the length of short semi-shaft and long semi-shaft. Then the shape of contact ellipse can be got. Use software MATLAB to apply a torque of 350.14 $n \cdot m$ to the pinion. According to the coordinate trajectory of tooth surface's contact spots in face gear, we can get the coordinate points of contact ellipse's short semi-shaft and long semi-shaft specific is shown in Table 3. Using software MATLAB can draw the face gear's contact trajectory, which shown in Fig. 6.

Table 3. Coordinates of contact points and the length of long and short shaft.

x	y	z	A	b
7.3056	-497.6686	-80.4069	6.6208	0.5917
6.0825	-497.6447	-77.7858	6.6140	0.4881
4.8604	-497.6813	-75.1647	6.6059	0.4573
3.6374	-497.7784	-72.5437	6.5923	0.4150
2.4116	-497.9359	-69.9226	6.5892	0.4031
1.1811	-498.1538	-67.3015	6.5567	0.3717
0.2540	-498.3568	-65.3357	6.5446	0.3452

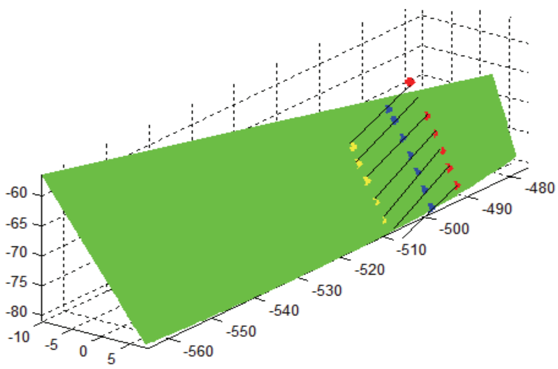


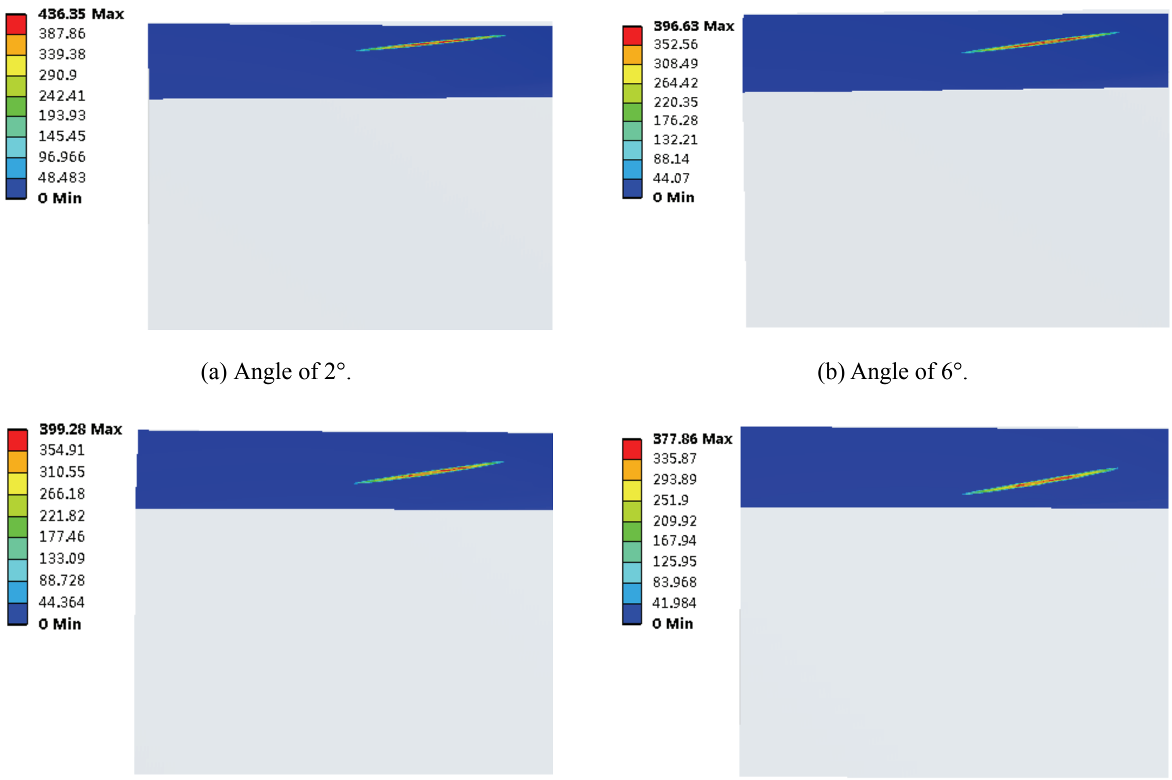
Fig. 6. Contact area of face gear.

The above conclusion is only considering the contact of a pair of tooth. Theoretically get the contact spots and the contact trajectory of face gear. Comparing the theoretical results, use finite element analysis software ANSYS WORKBENCH to analysis single tooth's static contact of face gear pair. Simulation materials for big wheel and pinion are structural steel. Simulation boundary condition is to constraint gearwheel's all degrees of freedom, and give the pinion a certain torque.

Based on the Hertz theory [3], the simulation results converge, when contact surface grid reach 0.2 mm. Under the action of torque, the pinion's each tooth turns an angle of 360/z in a circle, which means the turned angle of each tooth is 14°. Change the pinion 2° each time, so that we can simulate the whole process of dynamic meshing.

In the whole meshing process of the pinion, we study one of its teeth by the simulation. And for the face gear, the contact stress of tooth surface on one sides shown in Fig. 7.

Approximately, put the four contact trajectories shown above on one tooth surface. Then the distribution of contact lines of one tooth surface can be got, as is shown in Fig. 8.



(a) Angle of 2°.

(b) Angle of 6°.

(c) Angle of 10°.

(d) Angle of 14°.

Fig. 7. Contact stress of tooth surface under different angles.

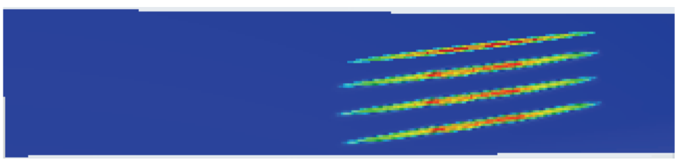
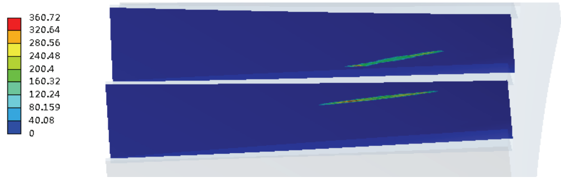


Fig. 8. Contact trajectory distribution of the same tooth surface.

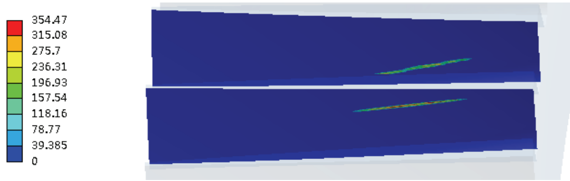
Compare Fig. 6 and Fig. 8. Without considering the influence of gear contact ratio, the face gear's contact spots are close to the small end-face and the tilt direction of contact line is from the small end-face to the large end-face when study a pair of tooth.

Under actual working conditions, contact ratio should be considered. So it is necessary to study the tooth surface contact state of multi-tooth working condition when we study the face gear's contact spots. Using the way of getting contact spots of single-tooth condition, we study the contact spots and contact path of multi-tooth condition in different angles. Specific is shown in following Fig. 9.

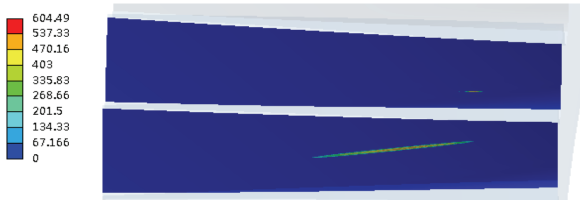
Approximately, put the contact trajectories which are in different angles on one tooth surface. Specific is shown in the follow Fig. 10.



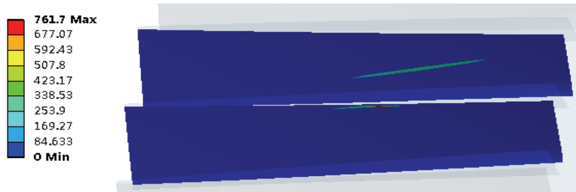
(a) Angle of the pinion: 2°.



(b) Angle of the pinion: 6°.



(c) Angle of the pinion: 10°.



(d) Angle of the pinion: 14°.

Fig. 9. Contact trajectory of tooth surface under different angles.

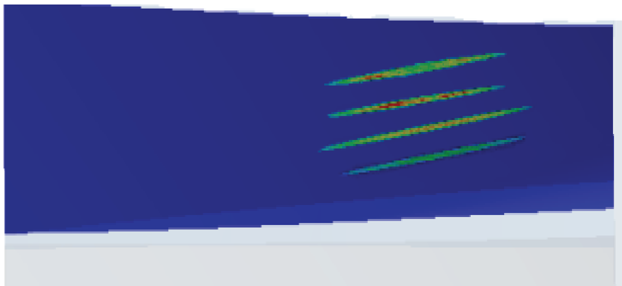


Fig. 10. Contact trajectory of the same tooth surface.

Compare Fig. 8 and Fig. 10. The contact ratio is considered in Fig. 10. So the trend of contact line trajectory becomes long gradually then becomes shorter, which is more accord with the actual meshing process of face gear.

Study on visualization of face gear's theoretical contact spots and face gear's contact areas after its elastic deformation by using software MATLAB. And use software to get the face gear's contact areas of simulation. So the theory and simulation can get verified with each other. Through the study of the contact spots of face gear, may provide certain theory reference for face gear vibration, noise and transmission error.

4. The Study of Face Gear Transmission Error

The ideal gear transmission is conjugate. When involute pinion turns an angle, the face gear (driven gear) also turns a corresponding angle. These two angles have an inverse relationship with two gear tooth number. But actually because of the elastic deformation produced on face gear, the transmission ratio is changed in the loading analysis. When the pinion turns an angle, the deviation of large gear's actual angle and theory angle can be described by the following formula (8):

$$\delta=(\varphi_2-\varphi_2^0)-z_1(\varphi_1-\varphi_1^0)/z_2 \tag{8}$$

In the above formula: φ_1, φ_2 are actual angle of the pinion and the large gear. φ_1^0, φ_2^0 are the initial position of two gears. $\frac{z_1}{z_2}$ is the theoretical transmission ratio of the gear pair.

This paper uses dynamic implicit algorithm to study the angular displacement error of face gear with software ABAQUS. To achieve higher precision, all simulations are analyzed by using hexahedron element. Enlarged view of local grid is shown in following Fig. 11.

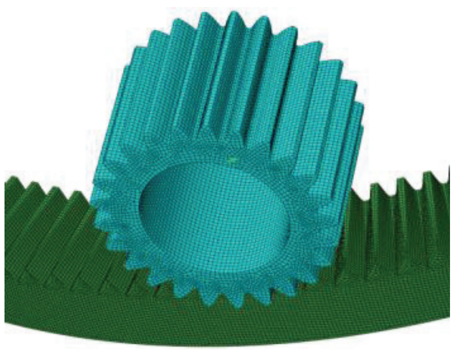


Fig. 11. Model of face gear with mesh.

Based on Hertz theory, we study the tooth surface contact stress through refining mesh size of tooth surface from 1mm to 0.2 mm. In the process, the tooth surface contact stress gradually reaches the theoretical value, but the amount of calculation is increased accordingly. Considering the calculation precision and calculation time, we choose 0.2 mm to be the mesh size. Then we set the boundary conditions of simulation analysis. Set reference point in the pinion's circle center and the large gear's circle center respectively. Use the function of coupling to couple the reference point with the gear's inner surfaces respectively. Apply the pinion's speed to its reference point and apply the torque to the large gear's reference point. Constrain the reference points' degrees of freedom, release the pinion's tangential degrees of freedom and release the large gear's tangential degrees of freedom. Apply a speed of 200r/min to the pinion and apply the resistance moment of: light load: 5E5 $n \cdot mm$, moderate load: 3E6 $n \cdot mm$ and heavy load: 6E6 $n \cdot mm$ to the large gear. The setting is shown in the following Fig. 12.

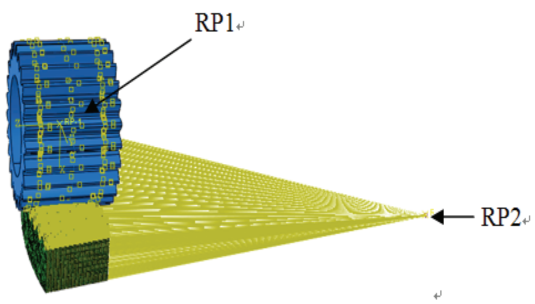


Fig. 12. Boundary conditions of face gear.

Apply the angular velocity on the reference point (RP1) of the pinion's circle center and apply the torque on the reference point (RP2) of large gear's circle center. Study the transmission error while keep the speed of the pinion unchangeable and change the resistance moment of the large gear. We can get the transmission error curves under different conditions. The curves are shown in the following Fig. 13.

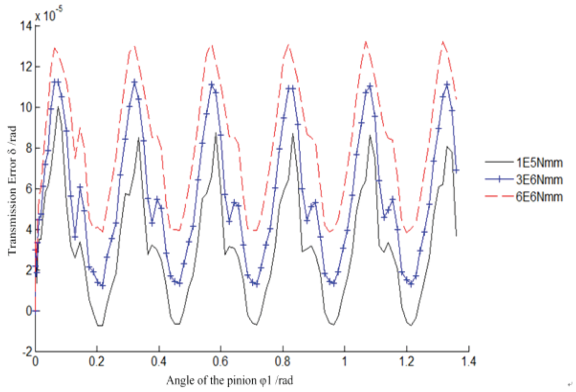


Fig. 13. Transmission error curves under different loads.

From the Fig. 13 we can see that: the transmission error of face gear is small, the shape of error curves approximates to parabola, transmission error curves under three kinds of loads tend to be uniform, and the fluctuation range of the error curves becomes larger as the load is high. In the analysis process, we use the way, which is shown in Fig. 3 to load. So when the pinion's angle turns from 0 to 0.2 radian, the load applied to the face gear increases linearly. Because the load is not stable, there producibility of the curves before 0.2 radian is worse than the reproducibility after 0.2 radian.

This paper studies the characteristic of face gear's transmission error when we apply low speed: 100 r/min, medium speed: 200 r/min, high speed: 400 r/min to the pinion and apply the constant torque of 5E5 $n \cdot m$ to the large gear. The curves are shown in following Fig. 14.

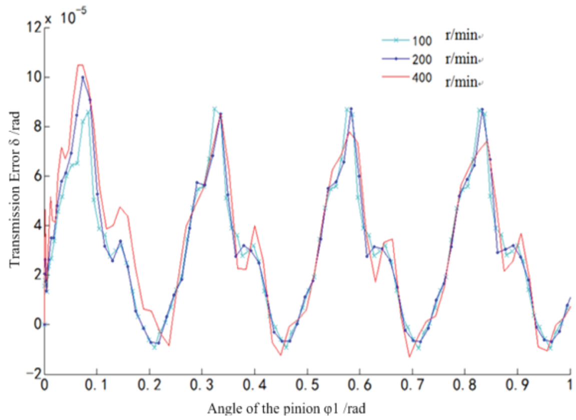


Fig. 14. Transmission error curves under different speeds.

From the Fig. 14 we can see that: when the pinion is in different speeds, the shape of error curves approximates to parabola. The difference between error curves is very small. The absolute differences of adjacent peaks in the curves are all between 9.6E-5 rad and 9.9E-5 rad. In transmission process, we can know that the change of drive gear's speed almost has no influence on face gear's transmission error. In the analysis process, we use the way which is shown in Fig. 3 to load. So when the pinion's angle turns from 0 to 0.2 radian, the load applied to the face gear increases linearly. At this time, the fluctuation range of the error curves increases as the speed becomes higher and the characteristics of gear vibration are more obvious. When the pinion's angle is greater than 0.2 radian, the load is constant and the transmission is stable. This study also explains that face gear pair is suitable for high speed transmission, its noise and vibration are small, and transmission is stable.

Through simulation technology we study the tooth surface transmission error of the face gear with staggered shaft installation error [9]. The angles are γ = 0.02 °, γ = 0 ° and γ = + 0.02 °. Use software ABAQUS to respectively extract angle curve of the large gear and pinion. Then according to formula (6), we get transmission error curves of the face gear under three different shaft angles. The curves are shown in the following Fig. 15.

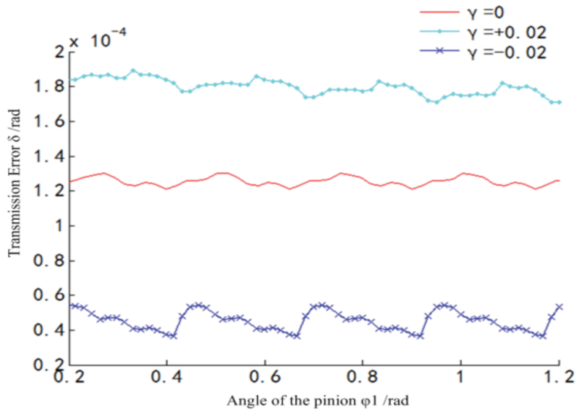


Fig. 15. Transmission errors curves under different shaft angles.

Put the three transmission error curves together. Compare the fluctuation under the three different angles. The curve is shown in the following Fig. 16.

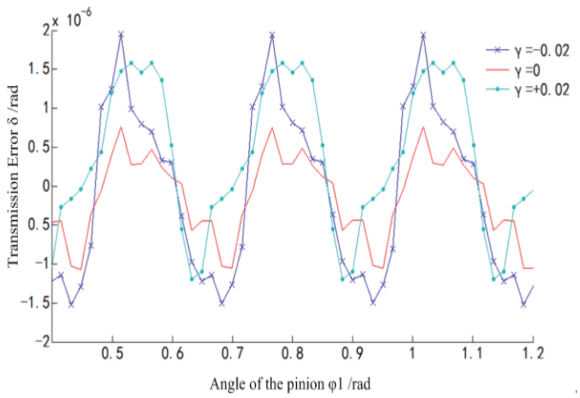


Fig. 16. Comparing of transmission errors curves under different shaft angles.

From the Fig. 16 we can see that when the face gear pair has installation error, the amplitude of transmission error increases obviously. The amplitude under the condition of having the installation error is 2.0 ~ 2.3 times bigger than the amplitude under the condition of standard installation. This accounts for that the vibration and noise will aggravate and the gear transmission precision will decline when the face gear works with installation error.

When the gear pair has a positive installation error: $\gamma = +0.02$, the instantaneous contact stress of face gear under different angles is shown in the following Fig. 17.

(a) Angle of 2°.

(b) Angle of 6°.

(c) Angle of 10°.

(d) Angle of 14°.

Fig. 17. Instantaneous contact stress of face gear with positive installation error.

Put the contact trajectories which are in different angles on one tooth surface. Specific is shown in the following Fig. 18.

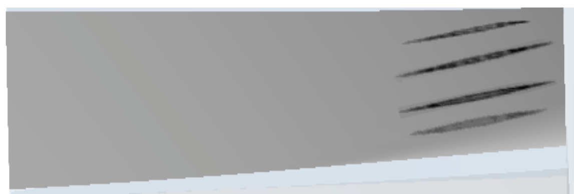


Fig. 18. Contact trajectory of the same tooth surface with positive installation error.

Compare Fig. 10 and Fig. 18, we can see that when the pair has a positive installation error, the tooth contact position is moving to face gear's inner surface and the edge contact is easily happened to the tooth. So the gear vibration and transmission error are increased.

When the gear pair has a negative installation error: $\gamma = -0.02$, the instantaneous contact stress of face gear under different angles is shown in the following Fig. 19.

(a) Angle of 2°.

(b) Angle of 6°.

(c) Angle of 10°.

(d) Angle of 14°.

Fig. 19. Instantaneous contact stress of face gear with negative installation error.

When the pair has a negative installation error, we study the contact spots in the same tooth surface through simulation analysis. The trajectories are show in following Fig. 20.

Fig. 20. Contact trajectory of the same tooth surface with negative installation error.

The face gear contact trajectory moves from its inner diameter to outer diameter when the face gear has negative installation error. The tooth thickness of face gear is not uniform, so when the gear has negative installation error, the gear clearance near the outside part increases in the ocess of gear transmission, tooth collision becomes more severe and vibration and noise are more serious.

5. Conclusion

This paper uses MATLAB to calculate the theory discrete points of the tooth surface of face gear. Through software Pro/E, we get the 3D model of face gear. Work out the coordinates of gear contact points and realize the visualization of face gear contact trajectories by the calculation of contact ellipse. Using software ABAQUS, we work out the face gear's transmission error curves and angular velocity error curves under different working conditions. And through simulation analysis, we get the transmission error curves of face gear with different shaft angle errors. This paper can provide a favorable reference for face gear design, processing and noise reduction.

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