JJMIE

Volume 14, Number 3, September. 2020
ISSN 1995-6665
Pages 271 - 277

Jordan Journal of Mechanical and Industrial Engineering

Research on the Effect of Rotation Speed on the Meshing Characteristics of Elliptical Cylindrical Gears

Changbin Dong* , Yongping Liu, Xingwen Fu, Chenchen Dou, Yuxin Yin

College of Mechanical & Electrical Engineering, Lanzhou University of Technology, Lanzhou 730050, P. R. China

Received October 19 2019 *Accepted August 4 2020*

Abstract

Aiming at the elliptical cylinder gear pair in the reversing device of the new drum pumping unit, the dynamic meshing process of tooth was simulated by LS-PREPOST software. The distribution law of effective plastic strain, effective stress, tooth surface pressure, tooth surface displacement and meshing force in the direction of tooth lines and tooth profiles under different rotational speeds were obtained. The results show that the effective plastic strain, the effective stress and the surface pressure in the tooth lines direction will decrease as the center position of the elliptical contact area of the tooth surface expands to both sides. The effective plastic strain, the effective stress and the surface pressure in the tooth profiles direction will increase with the increase of the rotational speed, and the rotational speed will affect the changing period of the tooth surface displacement and the meshing force. The research results can provide theoretical basis and certain guiding significance for the dynamic design, meshing analysis, modification and engineering application of non-cylindrical gears.

Keywords: Elliptical cylindrical gear; Tooth contact analysis; Effective plastic strain; Effective stress; Contact characteristics;

1. Introduction

As one of the simplest Non-Cylindrical gears, the elliptical cylindrical gear is distinguished from the ordinary cylindrical gear by its elliptic pitch curve, which is widely used in automatic machinery, printers, hydraulic pumps, hydraulic motors and flow meters for its compact structure and variable-ratio transmission. In recent years, tooth contact analysis (TCA) technology has developed rapidly in the field of gear, but the traditional TCA technology only considers the normal engagement of gear pair under the theoretical contact condition, and does not consider the influence of load on gear engagement. In view of this situation, the loaded tooth contact analysis technology (LTCA) has attracted extensive attention, which is a bridge connecting geometric design and mechanical analysis in the field of gear research, which is more in line with the actual working conditions of gears [1-2].

The teeth of elliptical cylindrical gears are different, but each tooth can be regarded as a tooth on the equivalent cylindrical gears. The contact analysis method of cylindrical gears can be used to analyze elliptical cylindrical gears. As far as spur gears are concerned, there are already many complete analysis techniques [3-5], and a lot of research results have been accumulated in the research of tooth surface contact. References [6-7] use ANSYS LS-DYNA analysis software to study gear meshing and contact characteristics of tooth. SANCHEZ-MARIN [8] proposed discretization and geometric adaptive refinement of the contact surface of teeth to solve the contact problem and calculate the instantaneous contact area of gear in meshing process. Wang Chen [9] proposed a method of tooth profiles modification based on tooth contact analysis technology. The modification parameters of rack cutter obtained by TCA technology can be transformed into the modification parameters of gear profile. Chen Ruibo [10], considering the contact relationship between the tooth surface, established the dynamic model of gear transmission. On this basis, the effects of meshing stage and operating conditions on the contact characteristics and dynamic characteristics of tooth are studied. Vasie Marius [1] proposed the method of generating pitch curve and tooth profiles of elliptical cylindrical gear and simulated the meshing of tooth in 2D and 3D environments respectively, and elaborated the meshing path, contact area and its change of tooth in detail. Zhang Huang [11] carried out numerical simulation of elliptical cylindrical gear transmission based on gear meshing principle and constructed a complete contact analysis method of elliptical cylindrical gear tooth profiles. Zhang Guohua [12] aimed at the high-order modified elliptical gears, and the contact stress of the tooth was obtained by simulating the meshing state of the tooth. The above research is of great significance to analyze the meshing characteristics of non-cylindrical gears, but there are few studies on the tooth surface contact in the dynamic meshing process of non-cylindrical gears. In this paper, a pair of elliptical cylindrical gears in the reversing device of

* Corresponding author e-mail: lutdcb@126.com

a new type of drum pumping unit is taken as the research object, and an accurate finite element model is established. Based on LS-PREPOST software, the dynamic meshing process of elliptical cylindrical gears is simulated to study the distribution of stress and strain in the meshing process of gears at different rotational speeds. Figure 1 is a model of elliptical cylindrical gear reversing device.

Figure 1. Reversing device of planetary gear train with elliptical cylindrical gears

2. Meshing theory of elliptical cylindrical gears

The curvature radius of elliptical cylindrical gears varies everywhere on the pitch curve, and the coincidence degree varies with it in the actual meshing process and is always greater than 1, which is mainly manifested by the meshing of one or two pairs of gears. According to the engagement of involute tooth profiles, the contact characteristics can be analyzed according to the Hertz theory of two cylinders contact models [13], so the contact stress of the tooth surface is as follows:

$$\sigma_H = \sqrt{\frac{p_{ca}}{\sum\rho}\cdot\frac{1}{\pi\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)}} \tag{1}$$

$$p_{ca} = \frac{F_n}{B} \tag{2}$$

$$\sum\rho = \frac{\rho_1\rho_2}{\rho_1+\rho_2} \tag{3}$$

Where p_{ca} is the calculated load on the unit length; B is the tooth width; F_n is the normal force on the tooth surface; E_1, E_2 elastic modulus of the two gears; μ_1, μ_2 are Poisson's ratio of the two gears in contact with each other; $\sum\rho$ is the comprehensive curvature radius at the two contact surfaces. The center distance of is a, and the extreme diameters of the pitch curve of the driving and driven wheels are r_1 and $r_2=a-r_1$. The input torque of the driving wheel is $T_1(t)$, and the output torque of the driven wheel is $T_2(t)$. The force diagram of the driving wheel in a pair of meshed non cylindrical gear pairs is shown in Figure 2.

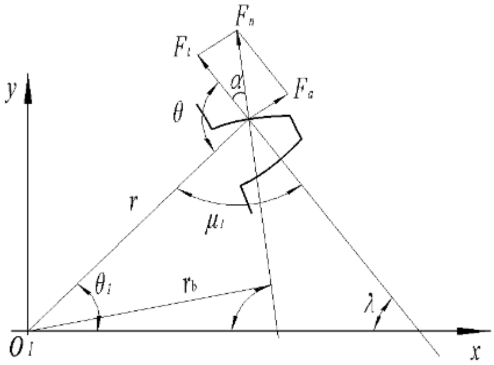


Figure 2. Stress diagram of involute profile of non-cylindrical gear

Where α is the angle between the tangential force F_t and the normal force F_n; F_a is the circumferential force; θ is the angle between the tangential force and the radius of the pitch curve; θ_1 is the angle between the radius of the pitch curve and the x-axis, and λ is the angle between the tangential force and the x-axis.

The tangential force F_t and the normal force F_n of the tooth surface of non-circular involute spur gears are respectively:

$$F_t = \frac{T_1}{r_1\sin\mu_1} \tag{4}$$

$$F_n = \frac{F_t}{\cos 20^\circ} = \frac{T_2(t)\sqrt{r_1^2+\dot{r}_1^2}}{r_1(a-r_1)\cos 20^\circ} \tag{5}$$

When a pair of tooth profiles engages at the pitch curve of non-circular involute spur gears, the meshing force is larger. When the gear material is the same, the nominal value of contact stress and the calculated value of surface contact stress are respectively

$$\sigma_{H0} = \sqrt{\frac{F_n E}{2\pi b(1-\mu^2)}\left|\left(\frac{1}{\rho_1}+\frac{1}{\rho_2}\right)\right|} \tag{6}$$

$$\sigma_H = \sigma_{H0}\sqrt{K_S K_A K_V K_{H\beta} K_{H\alpha}} \tag{7}$$

Where ρ_1, ρ_2 is the curvature radius at the two contact tooth surfaces; K_S is the meshing stiffness coefficient; K_A is the use coefficient; K_V is the dynamic load coefficient; $K_{H\beta}$ is the load distribution coefficient of the contact stiffness calculation, and $K_{H\alpha}$ is the load distribution coefficient of the teeth calculated by the contact stiffness.

The force acting on elliptical cylindrical gears in meshing process is complex and time-varying. The above expression can calculate the static force acting on the teeth, but some coefficients need to be selected according to experience, and it is difficult to calculate the instantaneous stress, strain and meshing force in meshing process. The LS-PREPOST software avoids the complex finite element programming and calculation and can simulate the actual meshing process completely. It can realize the load contact analysis of gears, which is more in line with the actual working conditions in the process of gear meshing, so it can be used to analyze the meshing characteristics of elliptical cylindrical gears. The parameters of elliptical cylindrical gears are shown in Table 1.

Table 1. Elliptical cylinder gear design parameters

Modulus m	3
Tooth number Z	47
Center distance a	150
Tooth top height coefficient h_a^*	1
Top clearance coefficient C^*	0.25
Tooth width B	30
Eccentricity e	0.3287
Pitch curve equation r	$r = \dfrac{64.667}{1 \pm 0.3287\cos\varphi}$

3. Analysis of dynamic meshing characteristics of elliptical cylindrical gears

In the process of gear meshing, load and power are transmitted in the form of tooth surface contact. The tooth profiles and tooth lines are two important characteristics of the tooth surface, and they are also the main factors affecting the tooth surface shape, meshing characteristics and contact characteristics [14]. Rotation speed will affect the impact time in the process of gear meshing, and then affect its meshing characteristics. In order to simplify the analysis process, the dynamic meshing characteristics of elliptical cylindrical tooth are analyzed from two aspects: the direction of tooth lines and the direction of tooth profiles, respectively, without considering the manufacturing and installation errors and displacement factors.

3.1. Load step analysis of gear tooth engagement

In order to simulate the actual contact of tooth in meshing process, the following boundary conditions should be set: the inner ring of rigid shaft hole drives the flexible body of gear to rotate, the gear material is Solid164 flexible body, the inner ring material of shaft hole is Shell163 rigid body. The driving and driven wheels limit the degrees of freedom of movement in three directions of X, Y and Z and the degrees of freedom of rotation in X and Y. The rotational speed of the driving wheel is 600 r/min. In the process of solving gear meshing model, excessive time steps and the proportion factor of calculation time step will lead to negative volume and interrupt simulation. The generation of negative volume is mostly caused by mesh distortion, which is related to mesh quality, material and load conditions. Therefore, appropriate time step should be taken to avoid negative volume. After debugging, the time step ratio factor TSSFAC is 0.5, and the time step DT2MS is -2×10^{-7}, which can completely simulate gear meshing.

According to the set boundary conditions, the finite element model of elliptical cylindrical gear is solved, and the dynamic meshing simulation of 0.1s is obtained. The elliptical contact area of the tooth at 0.0079s is shown in Figure 3. In the meshing simulation time of 0.1s, six time points are selected randomly, and the maximum contact stress in the meshing process of tooth are shown in Table 2.

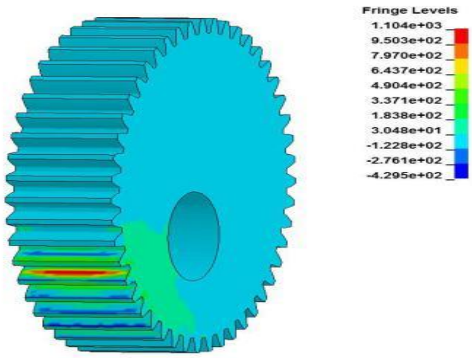


Figure 3. Elliptic contact area of elliptical cylindrical gear meshing

Table 2 Maximum contact stress of gear

Time(s)	Stress(×10^3 Mpa)	Time(s)	Stress(×10^3 Mpa)
0.0079	1.104	0.029	0.9724
0.038	0.9005	0.049	0.7403
0.078	0.6217	0.083	0.6233

The data in Table 2 show that the maximum contact stress of tooth decrease with the change of meshing time. This is due to the fact that in the initial meshing stage, point contact is dominant, and the impact is large. When the contact form is changed from point contact to line contact, the meshing is stable, and the stress of tooth tends to be stable. In Figure3, the contact area of the tooth is elliptical. During meshing, the elliptical contact area will expand symmetrically from the middle section to both sides, and the maximum stress occurs at the middle section. The stress decreases gradually as the transition from the middle section to the two ends of the teeth. When the thickness of the two meshing teeth is the same, the distance between the boundary of the elliptical contact area and the end face of the gear is about 5%~10% of the thickness of the tooth. If the width of the two meshing teeth is different, the elliptical contact area of the gear with smaller width will be larger.

3.2. Analysis of stress and strain in the direction of tooth lines at different speeds

Literature [15] points out that the tooth profiles of a gear are generally composed of three parts: the top part, the root part and the working area. The working area of a non-cylindrical gear, represented by an elliptical cylindrical gear, is generally near the pitch curve. Therefore, in order to study the stress and strain distribution in the direction of the tooth lines, it is necessary to collect isometric data from the working area near the pitch curve of the tooth lines. The location of the data acquisition point is shown in Figure 4.

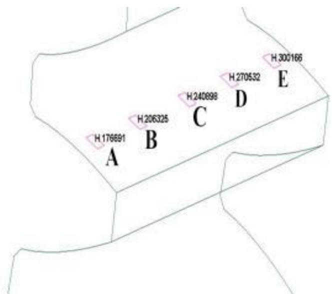
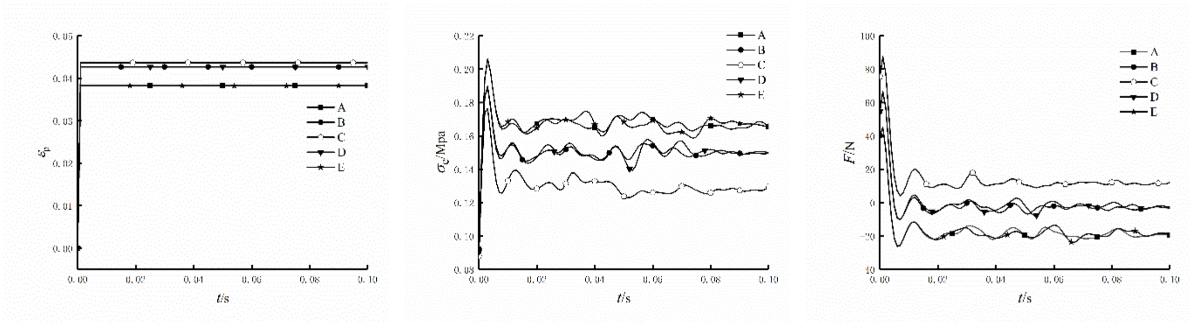


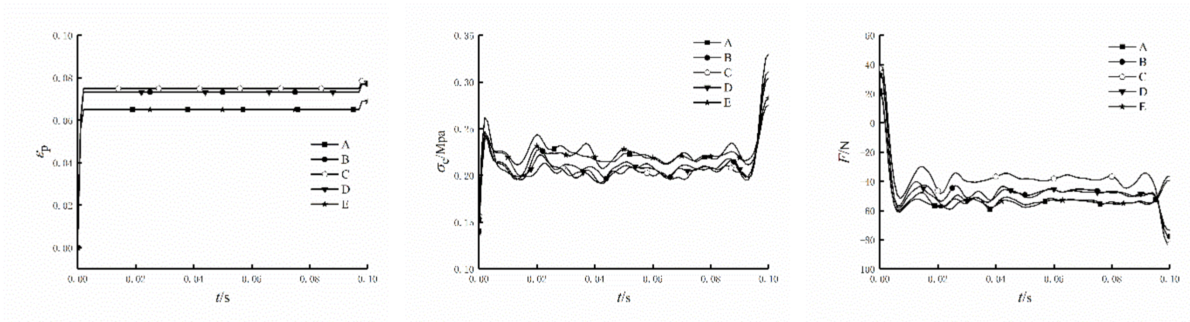
Figure 4. Data acquisition point of tooth surface

The increase of rotational speed will aggravate the wear and collision between teeth. In order to study the influence of rotational speed on the stress and strain distribution during meshing, the changing trends along the tooth lines under three conditions of rotational speed of 300r/min, 600r/min and 900r/min were obtained, as shown in Figure 5. The longitudinal coordinate ε_p represents the effective plastic strain, σ_e represents the effective stress, F represents the pressure on the tooth surface, and the abscissa represents the time.

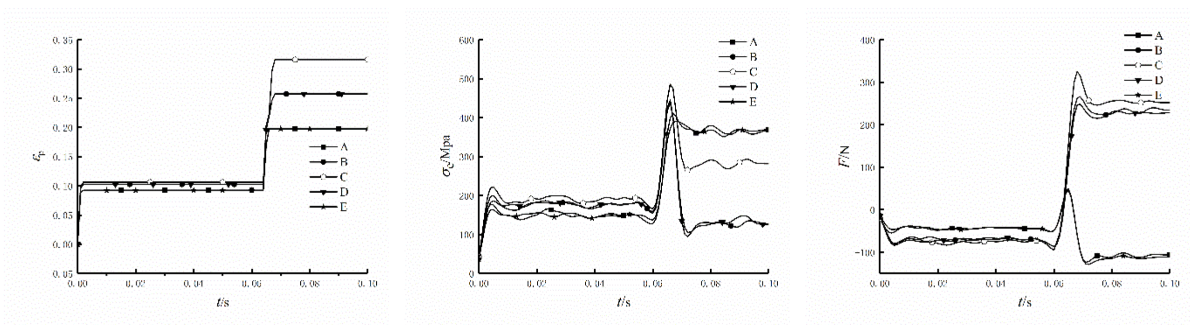
It can be concluded from the analysis that the effective plastic strain at point C is the largest, followed by points B and D, and points A and E are the smallest under three rotating speeds. With the increase of rotational speed, the effective plastic strain and surface pressure at the center of the elliptical contact area of tooth surface are the largest, and both decrease in varying degrees during the transition from the center position to both sides. With the increase of rotational speed, the increasing trend of C point is more obvious. At low speed, the two sides of the elliptical contact area in the direction of tooth lines are the largest, and the value of the elliptical contact area tends to decrease when it transits to the center area. When the rotational speed increases, the distribution law of the elliptical contact area in the center area is the largest and the transition area in the two sides decreases. The reason is that the meshing slip speed between the teeth is small at low speed, the stress and strain distribution in the micro contact area of the tooth surface is relatively concentrated and the numerical difference is small. At high speed, the slip velocity and wear between the teeth increase, and the stress-strain values differ greatly.



(a) 300r/min



(b) 600r/min



(c) 900r/min

Figure 5. Comparison of stress and strain in the direction of gear line at different speeds

3.3. Variation of Tooth lines Directional Displacement at Different Speed

The above analysis shows that the effective plastic strain, effective stress and surface pressure will be different between data acquisition points under different rotational speeds on the same tooth surface. In order to study the displacement distribution of data acquisition points during gear meshing process, the variation trend of tooth lines direction displacement under three rotational speeds is obtained, as shown in Figure 6.

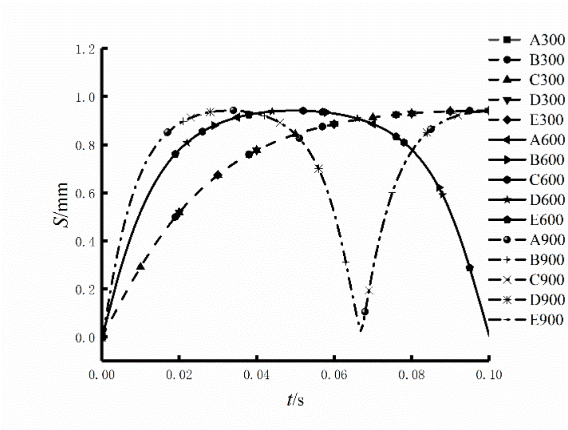
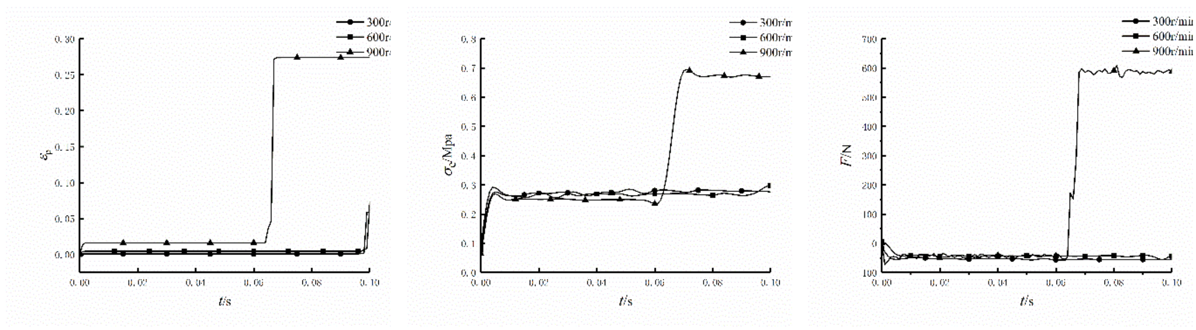


Figure 6. Variation trend of tooth lines displacement at different rotating speeds

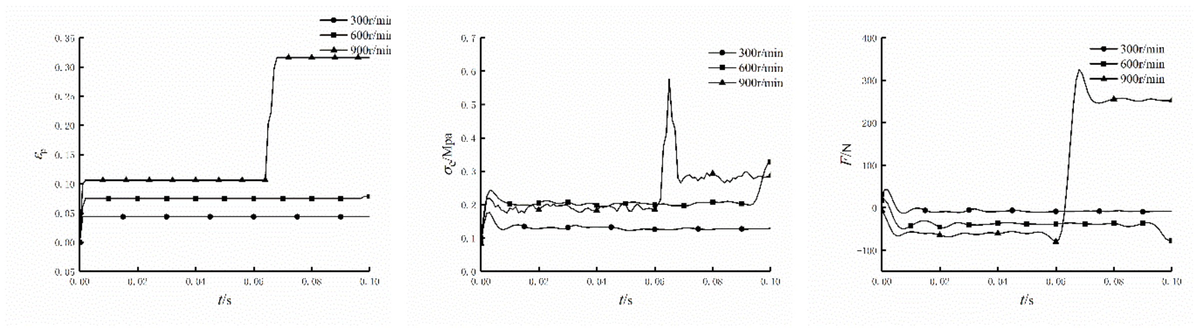
In Figure 6, the displacements of five data points on the same tooth surface show the same distribution law under different rotational speeds. The maximum displacements are basically the same, but with the increase of rotational speeds, their periods change. In the meshing time of 0.2s, the driven wheels rotate 1 cycle, 2 cycle and 3 cycle respectively under three rotating speeds, so the displacement curves in the figure 6 shows one cycle, two cycle and three cycle respectively, which indicates that the increase of rotating speed will affect the change period of tooth surface displacement, but the displacement of each point on the tooth surface will not change with the increase of rotating speed.

3.4. Stress and strain Analysis of Tooth profiles Direction at Different Rotational Speed

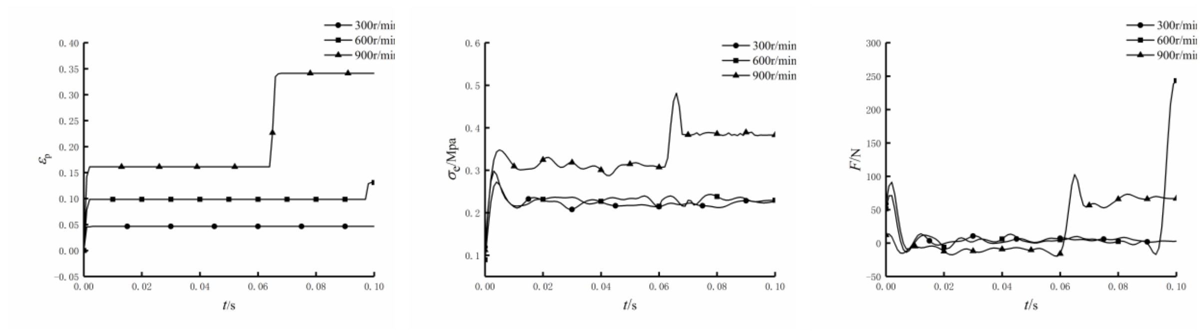
The variation trend of effective plastic strain, effective stress and surface pressure at the top of the tooth profiles, near the pitch curve and at the root of the tooth under different rotational speeds is shown in Figure 7. At 0.06s, the curves corresponding to the 900r/min rotation speed all appear sudden jump, and the effective stress and the pressure on the tooth surface near the pitch curve and the root of the tooth are impacted, which indicates that the meshing state of the tooth is not stable, and there will be noise and impact vibration. The reason is that the meshing period changes with the increase of rotational speed, and the effective plastic strain, effective stress and surface pressure of tooth increase sharply with the existence of meshing impact. But in this process, the curves are continuous and there is no interruption, which indicates that there is still a good contact characteristic between the teeth. With the increase of rotational speed, the effective plastic strain, effective stress and contact pressure at the top of the teeth, the vicinity of the pitch curve and the root of the teeth all tend to increase, which indicates that the contact condition of the gear is changing with the increase of the rotational speed. The increase of pressure and contact stress on the surface of tooth at high speed will aggravate the wear of the surface of tooth, so the plastic deformation will increase. When increases to a certain extent, the failure modes of tooth such as tooth surface gluing will occur, this is similar to that of standard cylindrical gears.



(a) The top of the tooth profiles



(b) The part near the pitch curve



(c) The root of the tooth profiles

Figure 7. Stress and strain comparison of tooth profiles at different rotational speeds

3.5. Variation of meshing force of tooth at different rotational speeds

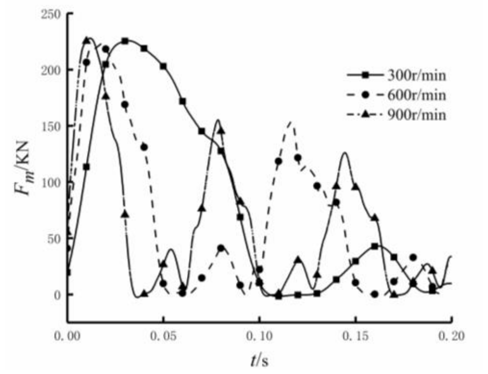


Figure 8. Trend of meshing force of tooth at different rotational speeds

Figure 8 shows that the maximum meshing force of tooth at three rotating speeds is 225.608 KN, 223.515 KN and 226.300 KN, respectively, with little difference between them. Under three rotational speeds, the meshing force increases instantaneously due to the meshing impact between the teeth in the initial meshing stage, and then changes periodically when the meshing is stable. The meshing force curve is smooth at 300r/min. When the speed increases to 900r/min, the meshing force curve will have some impact, and the gear system will have vibration and noise. For the driven wheels rotate 1r, 2r and 3r when the speed is 300r/min, 600r/min and 900r/min, respectively. In the meshing process of 0.2s, the meshing speed of tooth is slower at low speed, the impact vibration is smaller, and the curve is smoother. In high speed, the meshing time of tooth decreases, and the instantaneous impact and the vibration increase. The meshing force curves will produce non-smooth phenomena, but the meshing force curve is always continuous and there is no interruption, which indicates that there is no separation of tooth and have a good contact characteristic.

4. Conclusions

Aiming at the elliptical cylindrical gear pair, the LS-PREPOST software is used to simulate the dynamic meshing process of the tooth. The effective plastic strain, effective stress, surface pressure, displacement of the tooth lines and meshing force distribution of the tooth in the direction of tooth lines and tooth profiles at different rotational speeds are obtained.

1. The distribution of stress, strain and pressure in the direction of tooth lines will be affected by the speed of tooth. Along the direction of tooth lines, the effective plastic strain, effective stress and surface contact pressure in the center position of elliptical contact area are the largest and will decrease in varying degrees when the center position transits to both sides. The central part of elliptical contact area has the greatest wear. Therefore, the modification amount should be considered according to the meshing position, wear amount and load of the gear in the direction of tooth lines.
2. Along the tooth profiles direction of elliptical cylindrical gear, with the increase of rotational speed, the effective plastic strain, effective stress and surface contact pressure all tend to increase. The increase of rotational speed makes the meshing period of the tooth change, and the vibration and noise of the tooth will appear because of the meshing impact when the speed is large.
3. The rotation speed has a certain influence on the displacement of each data acquisition point in the tooth surface and the change period of the engagement force. The value of the engagement force will not change significantly with the increase in rotation speed. The engagement speed is slow at low speed, and the impact vibration is small. The engagement times decreases in high speed, and the wear impacts vibration between the gear surfaces increases, which will reduce the service life of gear. However, the meshing force curves of the three rotating speeds are continuous, which shows that the elliptical cylindrical gears have good contact performance under the three rotating speeds.
4. The analysis method and analysis process proposed in the article can be applied to the same type of non-cylindrical gears, which provides a new method for the subsequent study of the dynamic meshing characteristics of non-circular gears and the influence of working condition parameters on the meshing characteristics.

Conflicts of Interest

The authors declare that there are no conflicts on interests regarding the publication of this paper.

Acknowledgements

The research was supported by the National Natural Science Founding of China (No. 51765032)

Reference:

[1] Vasie Marius, Andrei Laurenţia. Analysis of noncircular gears meshing[J]. Mechanical testing and diagnosis, Vol.2,(2015).No.2,70-78.

[2] Wang Yunzhi. Load contact analysis and strength calculation of spur gear.[J]. Journal of Mechanical transmission, (2016).No.3,74-77.

[3] Chang Qinglin, Hou Li. Parallel Translating Mechanism Process-Oriented Mathematical Model and 3-D Model for Cylindrical Gears with Curvilinear Shaped Teeth.[J]. Jordan Journal of Mechanical and Industrial Engineering, Vol.10, (2016). No.3, 171-177.

[4] Tang Qian, Jin Xiaofeng, Fan Qiulei. Parametric Coordination and Simulation Study on Nonstandard Spur Gears.[J]. Jordan Journal of Mechanical and Industrial Engineering, Vol.8, (2014). No.2, 50-55.

[5] Ehsan Rezaei, Mehrdad Poursina, Mohsen Rezaei, Alireza Ariaei. A New Analytical Approach for Crack Modeling in Spur Gears.[J]. Jordan Journal of Mechanical and Industrial Engineering, Vol.13, (2019). No.2, 69-74.

[6] Tang Jinyuan, Liu Xin, Dai Jin. Study on corner contact shock of gear transmission by Ansys/Ls-Dyna Software.[J]. Journal of vibration and shock, Vol.26, (2007). No.9, 40-44.

[7] Liu Yanxue, Wang Jianjun, Zhang Tao. Dynamic meshing characteristics analysis of spur gears based on LS-DYNA.[J]. Journal of Beijing University of Aeronautics and Astronautics, Vol.42,(2016). No.10,2206-2213.

[8] Sanchez-Marin Francisco, Iserte Jose L, Roda-Casanova Vctor. Numerical tooth contact analysis of gear transmission through the discretization and adaptive refinement of the contact surfaces[J]. Mechanism & Machine Theory, Nol.101, (2016). 75-94.

[9] Wang Chen, Wang Shouren, Wang Gaoqi. A calculation method of tooth profiles modification for tooth contact analysis technology[J]. Journal of the Brazilian Society of Mechanical Sciences & Engineering, Vol. 40, (2018). No.7,340-349.

[10] Chen Ruibo, Zhou Jianxing, Sun Wenlei. Dynamic characteristics of a planetary gear system based on contact status of the tooth surface[J]. Journal of mechanical science and technology,Vol.32, (2018). No.1, 69-80.

[11] Zhang Huang. Digital tooth contact analysis of noncircular gear.[D]. Harbin Institute of Technology,2013.

[12] Zhang Guohua. Analysis of meshing characteristics and carrying capacity of high order deformed elliptic gear.[D]. Hefei University of Technology, 2017.

[13] Yao Wenxi. Non-circular Gear Design [M]. Beijing: Machinery Industry Press, 2012.

[14] Wei Yongqiao, Ma Dengqiu, Wu Yang, et al. Study on the tooth surface and curvature characteristics of cylinder gear with variable hyperbolic arc tooth trace. Advanced Engineering Sciences, Vol49 (2017). No.6, 196–203.

[15] Bair Biingwen. Tooth profiles generation and analysis of crowned elliptical gears[J]. Journal of mechanical design, Vol.7, (2009). No.131, 1-6.