International Journal of Science and Engineering Applications
Volume 15-Issue 04, 92 – 94, 2026, ISSN:- 2319 - 7560
DOI: 10.7753/IJSEA1504.1017

Dynamic Stress Analysis of Transmission System Gears

Wang Yu
College of Automotive and
Low Altitude Vehicles
Zibo Polytechnic University
Zibo 255300, PR China

Abstract: The dynamic analysis of the transmission process of the gear pair was conducted through the transient dynamics module. The changes in the maximum value of the first principal stress of the gears at different meshing positions during the transmission process were analyzed. The time-domain stress curves were transformed into frequency-domain stress curves through Fourier transformation, and the distribution of stress in the frequency domain was obtained.

Keywords:helical gear; transient dynamics; first principal stress.

1. INTRODUCTION

The main reducer is an important component of the electric vehicle transmission system. Its function is to reduce the motor speed and increase the motor torque, meeting the needs of acceleration, climbing, etc. The main reducer gears include arc tooth bevel gears, straight gears, double surface gears, helical gears, etc. The gears of the main reducer in electric vehicles, due to the smaller torque they bear, mainly adopt the forms of straight gears and helical gears. Compared with traditional bevel gears, helical gears have a simpler structure and higher transmission efficiency. This paper takes the helical gear pair of the main reducer of a certain low-speed-level electric vehicle as the research object, and conducts stress and fatigue life analysis of the gear pair. The geometric parameters are shown in Table 1.

Table 1. Main reducer helical gear parameters

Name	Module	Number of teeth	Pressure angle	Helix angle	Shift coefficient
Driving gear	2	18	20	13	0.50
Driven wheel	2	58	20	13	0.58

Kisssoft software has the advantage of parametric modeling and can directly create gear pairs without manual assembly. According to the gear geometric parameters in Table 1, the gear pair model is established in the professional gear software Kisssoft, as shown in Figure 1.

Figure. 1 Geometric model of main reducer gear pair

2. GEAR MATERIAL PARAMETERS

During operation, gears are subjected to alternating stress, which makes them prone to fatigue damage. To prevent premature fatigue damage to gears, the gear material must have sufficient contact and bending fatigue strength. The gear material studied in this paper is 20CrMnTi, which, after carburizing and quenching processing, has a surface hardness of 58-62 HRC. This material has high strength and toughness and, with its excellent comprehensive performance, accounts for approximately 50% of the market share of automotive gear materials. Its physical property parameters are shown in Table 2.

Table 2. Material parameters of 20CrMnTi

Elastic modulus	Poisson's ratio	Density	Tensile strength	Yield strength
207	0.25	7800	1080	835

3. TRANSIENT DYNAMICS ANALYSIS

3.1 Meshing Generation

Transient dynamic analysis involves large computational requirements and high memory usage. The dynamic meshing process of helical gears is a typical nonlinear contact problem, and high mesh quality is required. Reasonable meshing division is very important for transient dynamic analysis. For transient dynamic analysis, gears are in a rotating state during the analysis process, so all the teeth involved in meshing during the analysis time need to be encrypted.

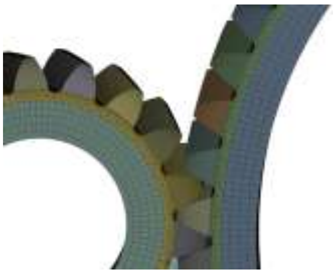


Figure. 2 Mesh model of gear pair

During the analysis time, 6 teeth of the driving gear and the driven gear, as well as the tooth roots, participate in meshing. The mesh size is 0.2mm, and the remaining parts are freely divided. A total of 727,558 elements and 2,408,617 nodes are obtained. The divided mesh model is shown in Figure 2.

3.2 Contact Settings

When the gears are in operation, there is a contact relationship between the driving gear and the driven gear. It is necessary to define the contact between the gear tooth surfaces. All the tooth surfaces of the driving gear are set as the contact surface, and all the tooth surfaces of the driven gear are set as the target surface. The contact algorithm adopts the Lagrange multiplier method. For the Lagrange multiplier method, the normal stiffness factor FKN is an important parameter that affects the accuracy and convergence of the finite element calculation results. The larger the FKN value, the more difficult the convergence is, but the penetration between the contact surfaces is small and the calculation accuracy is high; the smaller the contact stiffness is, the easier the convergence is, but the penetration is large and the calculation accuracy is low. The normal stiffness factor is selected as an empirical value of 1, which meets the requirements of the finite element analysis calculation accuracy and convergence. The contact type is frictional contact, with a friction coefficient of 0.08. The contact settings are shown in Figure 3.

Figure. 3 Gear pair tooth surface contact setting

3.3 Boundary Condition Settings

Add hinge constraints to the inner cylindrical surfaces of the two gears, restricting the translational degrees of freedom in the X, Y, and Z directions and the rotational degrees of freedom around the X and Y axes, so that the gears can only rotate freely around the Z axis. The torque and rotational speed are consistent with the test parameters in Chapter 5. Apply a rotational speed to the driving gear and a torque to the driven gear. The small gear is the driving gear, and a rotational speed of 1100 r/min is applied to it. When the driving gear withstands a torque of 99 N • m, the torque applied to the driven gear is calculated through the transmission ratio, which is 318 N • m. Apply a torque of 318 N • m to the driven gear, and set the torque of the driven gear to be in the same direction as the rotational speed of the driving gear. The analysis time is 0.015s. Set the automatic time step, with the initial sub-step number being 25 and the minimum sub-step number being 20, and the maximum sub-step number being 250. The load application situation is shown in Figure 4.

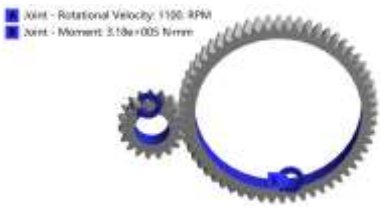
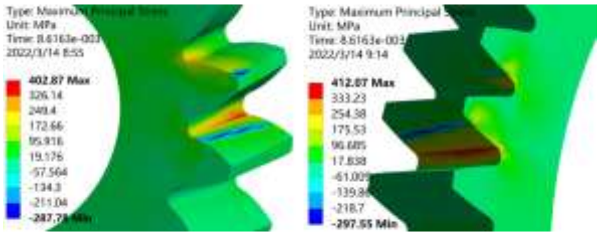


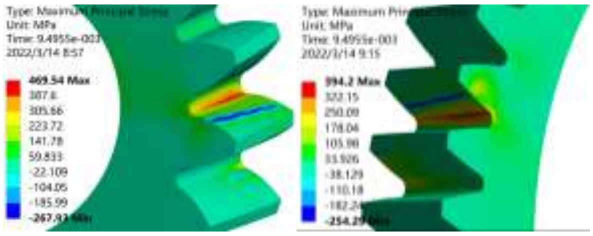
Figure. 4 Gear pair load setting

4. RESULT ANALYSIS

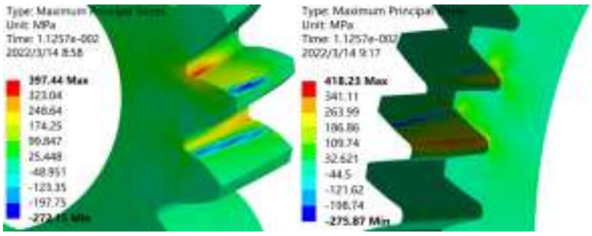
After the solution is completed, examine the changes in the distribution of the first principal stress, contact stress, and Von Mises stress at different times during one meshing cycle of the gear operation.



(a) 0.00862s after the start, engagement begins



(b) 0.00950s full meshing



(c) 0.01257s disengagement from meshing

Figure.5 The first principal stress contours of gear pair at different times

Figure.5 shows the variation of tensile and compressive stresses at the tooth root of the driving gear during one meshing cycle, from the beginning of meshing, full meshing to disengagement. The left side represents the driving gear and the right side the driven gear. When the gears mesh, the meshing tooth surface is under compressive stress, and the tooth root on the same side is under tensile stress. When the driving gear tooth is fully meshed, the tensile stress at the tooth root is the greatest. At the beginning of meshing, the maximum tensile stress occurs at the tooth root of the driven gear. When disengaging, the tooth tip of the driven gear and the tooth root of the driving gear are in contact. Due to the existence of the resistance moment, the tooth tip also experiences tensile stress. The tensile stress at the tooth root at the beginning of meshing is greater than that at the end of meshing. During the operation of the gear, the bending stress at the tooth root is not always uniformly distributed. At the beginning and end of meshing, the stress value at one side of the tooth root in the tooth width direction is high, while the stress value at the other side is relatively low. Therefore, when a helical gear actually breaks, cracks usually first occur at the tooth root, and the cracks do not extend along the tooth width as in spur gears, but rather extend towards the tooth tip at an angle close to 45°.

The curve of the maximum first principal stress at the tooth root of the gear pair over time is shown in Figure 6.

International Journal of Science and Engineering Applications
Volume 15-Issue 04, 92 – 94, 2026, ISSN:- 2319 - 7560
DOI: 10.7753/IJSEA1504.1017

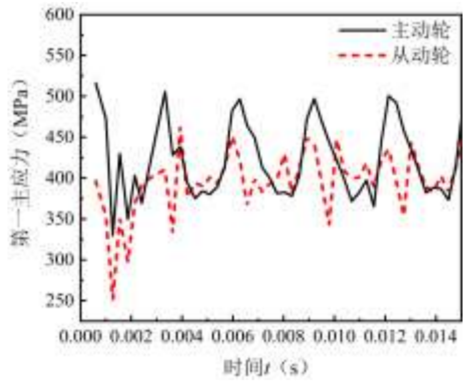


Figure.6 The first principal stress time domain curves of gear pair

At the beginning of meshing, due to the existence of contact clearance, the meshing of gears is not smooth. When the gears overcome the initial clearance, the meshing gradually becomes smooth and the curve changes in a relatively regular pattern. The tensile stress at the tooth root of the driving gear is greater than that of the driven gear. The tensile stress at the tooth root of the driving gear varies between 329 MPa and 516 MPa, while that of the driven gear varies between 252 MPa and 463 MPa. When the gears start to mesh from the tooth tip, the bending stress at the tooth root gradually increases. When the contact line of the tooth surface is fully meshed, the tensile stress at the tooth root reaches its maximum. Subsequently, as the position of the contact line gradually moves down, the tensile stress at the tooth root begins to decrease, showing a periodic cycle.

The time-domain curve of the first principal stress was transformed into a frequency-domain curve by applying the Frequency Spectrum module of Ncode through Fourier transformation. The transient analysis time was 0.015s, divided into 49 sub-steps, so the sampling frequency was set at 3402Hz. The first principal stress frequency-domain variation curve obtained is shown in Figure 7.

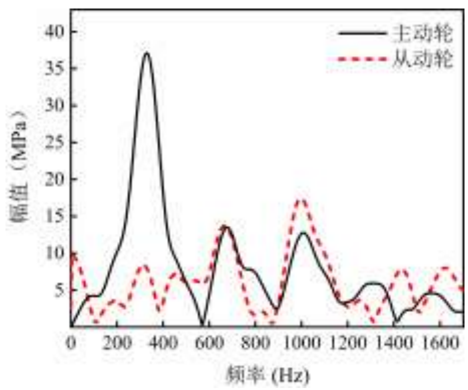


Figure. 7 The first principal stress frequency domain curves of gear pair

The calculation formula for the gear meshing frequency is

$$f = \frac{n \cdot z}{60}$$

After calculation, the meshing frequency of the gear at the current rotational speed (1100 r/min) is 330 Hz. As shown in Figure 3.6, the maximum tensile stress of the driving wheel occurs around the meshing frequency of 330 Hz, while the tensile stress peak of the driven wheel is around the three times frequency of 990 Hz. The tensile stress of the gear reaches its peak at the meshing frequency, the double frequency, and the triple frequency, and there are also peaks near the four times frequency and the five times frequency. To prevent resonance damage to the gear root, when designing the gear, in addition to avoiding the gear's natural frequency being close to the meshing frequency, the gear's natural frequency should also be offset from the meshing multiple frequency.

5. REFERENCES

[1] Čular I, Vučković K, Žeželj D, et al. Analytical approach for low and high cycle bending fatigue life prediction of carburized gear steel specimens[J]. Engineering Failure Analysis, 2020, 108: 104328.

[2] Liu H L, Liu H J, Zhu C, et al. Study on gear contact fatigue failure competition mechanism considering tooth wear evolution[J]. Tribology International, 2020, 147: 106277.

[3] Morales-Espejel G E, Rycerz P, Kadiric A. Prediction of micropitting damage in gear teeth contacts considering the concurrent effects of surface fatigue and mild wear[J]. Wear, 2018, 398: 99-115.

[4] Tang Zhonglian, Qi Xiumei, Gao Chuankuan. Influence of Tooth Surface Roughness on Contact Fatigue Stress in Gear Transmission [J]. Lubrication Engineering, 2011, 36(04): 53-57.

[5] Wang Shuren, Yan Yutao, Yin Weili, et al. A Computational Model for Frictional Fatigue Wear in Gear Meshing [J]. Journal of Northeastern University (Natural Science), 2008(08): 1164-1167.

[6] Weber C, Tobie T, Stahl K. Investigation on the flank surface durability of gears with increased pressure angle[J]. Forschung im Ingenieurwesen, 2017, 81(2): 207-213.

[7] Winter H, Weiss T. Some factors influencing the pitting, micro-pitting (frosted areas) and slow speed wear of surface hardened gears[J]. Journal of Mechanical Design, 1981, 103(2): 499-505.

[8] Wang Xiaoqun, Huo Dongju. Study on the Influence of Tooth Root Transition Curve on Gear Bending Fatigue Strength [J]. Machinery Design & Manufacture, 2008(08): 114-116.

[9] Govind V J, Sachin K, Kumar K P. Crack growth modeling in spur gear tooth and its effect on mesh stiffness using extended finite element method[J]. Engineering Failure Analysis, 2018, 94: 109-120.

[10] Zhang Jiancheng, Zhang Qingxia, Mao Zhiyong. Gear Fatigue Life Analysis Based on Driving Simulation Tests [J]. China Mechanical Engineering, 2014, 25(8): 1001-1006.

[11] Lin C, Li P, Liu W. Analysis and calculation of bending fatigue life of curve-face gear[J]. Journal of the Brazilian Society of Mechanical Sciences and Engineering, 2020, 42(1): 1-11.

[12] Liu Xinbo, Zeng Xiaosong, Liu Yong. Prediction and Analysis of Bending Fatigue Crack Propagation Behavior in Spur Gears [J]. Journal of Mechanical Transmission, 2016, 40(09): 179-183.