Simulation and Life Prediction of Gear Meshing Process of Gearbox of A Crawler Vehicle

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Abstract. A crawler vehicle gearbox was easy to invalidation, stress change of the gear tooth meshing process was difficult to accurately measure, and the life prediction was not accurate. This paper established a finite element model of gear meshing, the analysis of the gear teeth began to come into contact with the complete separation of highly nonlinear dynamic process, and the simulation results of test, and established the life prediction model by nCode Designlife software, and estimated the gear fatigue failure position and life. Compared with the actual results verified the feasibility of dynamic analysis method of gear meshing, for gear strength check, the reference for the optimization design and fatigue life prediction.

1. Introduction
Gears in the transmission process, due to the friction and wear, meshing impact and the influence of such factors as easy to invalidation, all invalidation modes in tooth, the tooth root bending fatigue invalidation accounted for the largest share, followed by the tooth surface contact fatigue failure. Second gears of a crawler vehicle were involute spur gears, using the theory of the ball or cylinder contact analysis before the tooth contact state at a certain moment, already could not comprehensively reflect the changing process of tooth surface contact state. In this paper, by using ABAQUS/Explicit dynamic Explicit finite element method, the contact stress and fatigue bending stress of the second gear of a crawler vehicle transmission with time were analyzed, and the life of the gear was predicted.

2. Gearbox load spectrum obtained
In order to simulated the stress changes on the tooth surface and the tooth root during the gear meshing process under actual conditions, dynamic simulation was carried out on the established virtual prototype of a crawler vehicle. The simulation time was set at 1s and the step length was 0.001s, and the load spectra of the torque and angular velocity of the driving gear and passive gear under the second-grade E-class road surface were obtained. See figure 1 and figure 2.

Through wavelet analysis, filter out the vibration produced by interference signal, the second driving gear angular velocity and the passive gear torque as shown in figure 3 and figure 4 shows, from the diagram, in 0.35 ~ 0.45 s gear because of the influence of road roughness mutations made torque, due to the mutation load was the main cause of gear failure, therefore, The angular velocity and torque of the driving gear and passive gear within 0.35~0.45s were taken as the boundary conditions of finite element.
3. Finite element model establishment and result analysis

The driving system virtual prototype was simulated from 0.35–0.45s to obtain the angular velocity and torque of the driving gear and passive wheels added to the boundary conditions of ABAQUS. At the same time, the active wheel speed was set as 0.14rad/s, and the simulation duration was 0.1s. Finite element simulation analysis was conducted on gear meshing. The stress cloud diagram as a function of time was shown in Figures 5–8.

It could be obtained from the analysis in Figure 5–8 that in the transmission process of gears, the contact force generated when the gears first contact was large, which led to the bending and deformation of the tooth root of passive gear. As the driving gear rotates, the meshing position...
changed from the tooth root to the tooth end of the driving gear and from the tooth end to the tooth root of the passive gear, causing the contact stress position to change along the tooth surface, and the bending stress of the tooth root occurs in the whole process of meshing. Therefore, in the process of gear meshing, tooth root was prone to fatigue failure, tooth surface due to collision, friction and other reasons, easy to produce tooth surface peeling.

It could be seen from the stress cloud diagram that the stress of the gear tooth was very small and basically zero in the non-meshing condition. In the case of meshing, the stress value increases rapidly and reached the peak in a very short time. This showed that the stress in the meshing process of gear teeth was similar to the dynamic stress under the impact load.

The number of teeth of the second gear of a crawler vehicle transmission was 20, the number of teeth of the passive gear was 28, the modulus was 9mm, the manufacturing accuracy was level 6, and the load impact was medium impact. According to the Hertz contacted stress calculation formula (shown in Formula 1):

$$\sigma_{\text{max}} = \sqrt{\frac{\omega_n}{\pi \rho_{\text{red}}} \frac{E_1}{1 - \nu_1^2} + \frac{E_2}{1 - \nu_2^2}}$$

where, $\nu_1, \nu_2$ was poisson's ratio of two cylinders, $E_1, E_2$ was the elastic modulus of two cylinders, $\rho_{\text{red}}$ was the radius of comprehensive curvature, $\omega_n, \omega_n$ was the normal force on the length of the unit contact line of the cylinder.

It was calculated that when the driving gear and passive gear started to contact, that was, the contacted stress of the passive gear near the tooth tip and the active gear near the tooth root was the maximum, and the value was 989.5 MPa.

**Figure 9.** Root bending stress diagram of driving gear under tension  
**Figure 10.** Bending stress diagram of passive gear root under pressure

Fig 9 and 10 from the stress nephogram and bending stress graph could be seen that, in the process of the whole tooth mesh, tooth side by extrusion, the other side by stretching, stress concentration in the fillet part, and the tooth root were subjected to bending stress and tooth started contacted as the tension side of the driving gear and driven gear tooth root bending stress from zero to maximum pressure side, but as time extended, stress decreases, until it was zero, this suggests that the gear meshing process, the tooth root was the part most prone to fatigue failure. Calculation formula of tooth root stress of spur gear (shown in Formula 2):

$$\sigma_F = \frac{F_t}{b m_n} K_A K_Y K_{FP} Y_{Sa} Y_{\beta}$$

Determine parameter values by referring to the manual, $K_A = 1$, $K_Y = 1.514$, $K_{FP} = 1.17$, $Y_{Sa} = 1$, $Y_{FS} = 4.28$, was substituted into equation (2), it could be obtained that under the load of 3700 Nm
meshing torque, the maximum bending stress of the tooth root of the passive gear was 357.4 MPa, and the maximum bending stress of the tooth root of the active gear was 443.6 MPa.

For the maximum stress simulation results, Hertz contact ED theory calculation results and bending stress calculation results were compared, as shown in table 1.

| Passive gear | Driving gear |
|--------------|--------------|
| $\sigma_{\text{max}}$ | $\sigma_{F}$ | $\sigma_{\text{max}}$ | $\sigma_{F}$ |
| Theoretical value | 989.5 | 357.4 | 989.5 | 443.6 |
| Simulation value | 1031.6 | 361.5 | 996.7 | 454.6 |

$\sigma_{\text{max}}$ (MPa) was the contact stress of tooth surface.
$
\sigma_{F}$ (MPa) was the bending stress of tooth root.

The simulation results were compared with the theoretical values. Considering the existence of friction force and the fact that both torque and speed were load spectra simulating the actual road conditions, both the contact stress and bending stress values were greater than the theoretical values. This research method was closer to the actual value and had certain reference value.

4. Gear life prediction

ABAQUS/Explicit simulation results in higher fatigue analysis software nCode Designlife life prediction and fatigue failure parts in the gear was 20Cr2Ni4A alloy steel material, after carburizing and quenching and low temperature tempering heat treatment, surface hardness was not under HRC57, core hardness was HRC34 ～ 45, reliability was 99%, the tensile strength limit was 1175 MPa, modulus of elasticity for the MPa.

To conservative projections for gear life, the heat treatment of residual compression stress in this should not be considered, from access to the complete separation of gear dynamic process of cyclic loading, according to the linear cumulative damage theory, the simulation given gear fatigue failure of the parts in advance as the root, and the driving gear than passive gear failure in advance, the small gear failure in advance.

According to the conversion formula (3) between the second gear speed and stroke of a crawler vehicle, the driving distance of the second gear when fatigue failure occurs could be converted.

$$S = N \frac{S'}{1000}$$

Where, $S'$ was the driving distance corresponding to the load time history; $S$ was the mileage corresponding to the fatigue life of parts; $N$ was the cycle number of fatigue life.

By a crawler vehicle transmission ratio, the second driving gear turned a circle, the vehicle moved 0.57 m, but the passive gear turned a circle, the vehicle moved 0.8 m, according to software analysis, the survival rate 99%, the driving gear tooth root after $2.523 \times 10^6$ cycles fatigue failure occurs, the passive gear tooth root after $4.4 \times 10^6$ cycles fatigue failure occurs, calculated by the type (3), the driving distance before invalidation of the second driving gear and passive gear:

$$S_1 = N \frac{S'}{1000} = 2.523e6 \frac{0.57}{1000} = 1438.11 km$$
$$S_2 = N \frac{S'}{1000} = 4.4e6 \frac{0.8}{1000} = 3520 km$$

According to the research and statistical analysis, the second usage accounted for 20% of the usage of the gear, the convert of the crawler vehicle mileage, under the second gear to run about 1600 ~ 1900 km dedendum fatigue failure occurs, consistent with actual situation, and as a result, a crawler vehicle can be determined through simulation analysis method of overhaul time, has certain engineering application value.
5. Conclusion
By ABAQUS/Explicit dynamic analysis method, to the height of the gear meshing process by simulating the nonlinear problem, and compared with the theoretical value and the simulation value by tooth surface contact stress and tooth root bending stress, and using the fatigue life of simulation compared with survey data. The results showed that the dynamic analysis was carried out on the gear meshing process to closer to the actual situation, the result more accurate, the fatigue life of the resulting results consistent with the research results, to predict the gearbox life provides an important reference.

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