Dynamic Simulation and Analysis of 2Z-X (A) Four Stage Series Planetary Gear Train

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Abstract. In order to understand its dynamic characteristics in operation, a four-stage 2Z-X (a) planetary gear train was simulated based on ADAMS. The simulation uses the solid contact model to analyze the meshing force vibration of the gear train under the condition of constant speed and variable speed, and uses FFT to understand the frequency characteristics of the gear train meshing. The research provides the basis for the follow-up design and optimization.

1. Introduction
Planetary gear train has compact structure, stable transmission and high transmission efficiency. It is widely used in mechanical transmission, especially in automatic production equipment with high precision requirements. But in the transmission process of gear as an elastomer, it is inevitable that the gear will be deformed under load, coupled with the logarithmic change of meshing teeth and gear error, which will generate dynamic meshing force in the gear transmission, and the gear train will generate vibration and noise under the excitation of meshing force. This research is mainly aimed at the dynamic simulation of a four stage series micro planetary gear train transmission system used in a precision machine, which provides the basis for the subsequent design and development.

The reducer is composed of four-stage 2Z-X (a) planetary gear trains in series. The kinematic diagram of the mechanism is shown in Figure 1, and the parameters of each stage gear are the same. The module of each gear is 2mm, the number of sun gear teeth at all levels is 20, the number of internal gear teeth at all levels is 100, and the number of planetary gear teeth at all levels is 40.

Figure 1. Kinematic diagram of gear train mechanism.

The three-dimensional model of the gear train is created in SolidWorks software, as shown in Figure 2.
Figure 2. Three dimensional model of planetary reducer.

2. Principle of simulation analysis

The impact function method is used to calculate the contact force by default in ADAMS. According to the impact function, the contact force is composed of the elastic force generated by the invasion of two components and the damping force generated by the relative velocity of two components.

\[
F_{\text{impact}} = \begin{cases} 
0 & q > q_0 \\
K(q_0 - q)^\gamma - C \frac{dq}{dt} \text{step}(q_0 - d_1, 1, q_0, 0) & q \leq q_0 
\end{cases}
\]

(1)

Where, \( q_0 \) is the initial distance between two teeth; \( q \) is the actual distance between two teeth during collision; \( dq/dt \) is the gear collision speed; \( K \) is the contact stiffness coefficient of two gears; \( C \) is the maximum damping coefficient; \( e \) is the stiffness index; \( d \) is the invasion depth when the damping is maximum, and step is the ADAMS function.

The contact force caused by the collision and engagement of gear teeth can be regarded as the problem of two axis parallel cylinder impact contact. The solution to this problem can be obtained from Hertz contact theory.

Under the action of normal load \( F_N \), a rectangular contact surface with width of \( 2a \) and length of \( B \) is formed due to the local elastic deformation on the contact surface of gear teeth. According to Hertz contact theory, the contact surface size Hertz half width \( a \), contact stress \( \sigma_H \) and contact deformation \( \delta \) are respectively formula (2) ~ (5).

\[
a = \sqrt{\frac{4F_N}{\pi b} \left( \frac{1 - \mu_1^2}{E_1} + \frac{1 - \mu_2^2}{E_2} \right) \left( \frac{1}{R_1} + \frac{1}{R_2} \right)}
\]

(2)

\[
\sigma_H = \sqrt{\frac{F_N}{b} \left( \frac{1}{R_1} + \frac{1}{R_2} \right)} \left( \frac{1 - \mu_1^2}{E_1} + \frac{1 - \mu_2^2}{E_2} \right)
\]

(3)

\[
\delta = 0.58 \frac{F_N}{Eb} \left( \ln \frac{4R_2 a^2}{F_N} - 0.429 \right)
\]

(4)
\[
\frac{1}{E} = \left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)^{-1}
\]  

(5)

R1 and R2 are the equivalent radius of curvature of the two gears at the contact point; \(\mu_1\) and \(\mu_2\) are the Poisson's ratio of the two gear materials; \(E_1\) and \(E_2\) are the elastic modulus of the two gear materials; \(E\) is the comprehensive elastic modulus; \(B\) is the tooth width; \(\delta\) is the deformation caused by the contact between the load acting point and the intersection of the tooth centerline and the meshing line.

### 3. Pretreatment based on ADAMS

The three-dimensional model of the gear train is created in SolidWorks software and imported into Adams to add the constraint between components, as shown in Table 1.

| Component 1            | Component 2                        | constraint  |
|------------------------|-------------------------------------|-------------|
| Ground                 | Internal gear of each gear train    | Fixed       |
| Ground                 | Sun wheel of each gear system       | Revolute    |
| Internal gear of the gear train | Planetary gear of the gear train | Contact     |
| Sun wheel of the gear train | Planetary gear of the gear train | Contact     |
| Planet carrier of the gear train | Planetary gear of the next gear train | Revolute |
| Planet carrier of the gear train | The sun wheel of the next gear train | Fixed       |

The materials of the parts in the gear train are as follows.

Gear material is 20CrMo, Its elastic modulus \(E_1=2.1\times10^5\) N/mm². Poisson's ratio \(\mu_1 = 0.278\). Material density \(\rho_1 = 7.84\times10^{-6}\) kg /mm³. The material of the shaft is 20CrMnTi. Its elastic modulus \(E_2=2.12\times10^5\) N/mm². Poisson's ratio \(\mu_2 = 0.298\), Material density \(\rho_2 = 7.86\times10^{-6}\) kg /mm³.

### 4. Analysis of simulation results

Add drive and load according to the rated working condition of the planetary gear train. Use step function to define the drive and load, where the drive function is step (time 0, 0, 0.1, -10000d), and the load function is step (time 0.3, 0, 0.4, 100), as shown in Figure 3, and the output speed of planetary wheels at all levels is shown in figure 4.

![Figure 3. Drive and load settings.](image-url)
The output speed of the first stage planetary gear train fluctuates slightly up and down in $1666/\text{s}$. The output speed of the second stage planetary gear train fluctuates slightly up and down at $277/\text{s}$. The output speed of the third stage planetary gear train fluctuates slightly up and down at $46.29/\text{s}$. The output speed of the fourth stage planetary gear train fluctuates around $7.5/\text{s}$. The speed of each stage is consistent with the theoretical calculation.

It can be seen that there are a series of high frequency components in the output speed of the first stage planetary gear train through FFT, as shown in Figure 5. The frequency of these vibration components is an integral multiple of 462Hz, and the amplitude decreases with the increase of frequency.

Because the fourth stage is directly connected with the actuator, the operation of this stage directly affects the movement stability and accuracy of the equipment. The speed spectrum of this stage gear train is shown in Figure 6. The main vibration component frequencies are 25.63Hz and 51.27Hz.
5. **Analysis of meshing force at constant speed**

Gear meshing force is a dynamic excitation generated in the operation of the gear train, which is directly related to the vibration and noise of the mechanical equipment. Therefore, it is necessary to understand the dynamic characteristics of the meshing force of the gear train under the design conditions. Because the meshing vibration between the sun wheel and the planet wheel will be directly transmitted to the next stage, it is necessary to focus on the analysis.

The magnitude of engagement force of the first stage sun wheel and planetary wheel is shown in Figure 7, and the change of engagement force before and after loading can be clearly seen. In order to further understand the law, we can observe the component of meshing force on a certain coordinate axis, as shown in Figure 8, we can clearly see that the meshing force presents a sinusoidal law. In order to further understand the frequency characteristics of meshing forces, the fast Fourier transform can be made for the meshing forces of different gear trains, as shown in figure (9) ~ (12).

![Figure 7. Amplitude of meshing force of the first gear.](image1)

![Figure 8. Y-axis component of meshing force of the first gear.](image2)

![Figure 9. Spectrum of meshing force of the first gear.](image3)
By observing the frequency spectrum of meshing forces at all levels of gear trains, it can be found that the vibration frequency of meshing forces gradually approaches to the low frequency region with the decrease of rotating speed.

6. Analysis of meshing force under variable speed

In order to analyze the vibration of the gear train at different speeds, the input speed is set to 30000t * time, the load setting is the same as before, and the simulation time is 1s. That is to say, the law of the vibration frequency of each order during the acceleration of the gear train from 0 to 30000 is obtained. The amplitude of the engagement force between the first stage sun wheel and the planet wheel is shown in Figure 13. It can be seen that the frequency of the engagement force changes more and more
quickly. In order to further analyze the law, 3D FFT transformation can be carried out to obtain the waterfall map of the meshing force of this gear train, as shown in Figure 14.

It can be seen that the wave crest of vibration shows the phenomenon of extending to the right and up. Therefore, with the increase of rotating speed, the meshing vibration frequency of the first stage gear train becomes larger and larger, and the vibration amplitude also increases.

The meshing forces of the four-stage gear trains are all changed by three-dimensional FFT, and the results are compared together, as shown in Fig. 14–17. It can be seen from the comparison that as the speed of each planetary gear train decreases, the vibration peak gradually approaches to the low frequency region.

![Figure 13. Y-axis component of meshing force of the first gear.](image1.png)

![Figure 14. 3-D Spectrum of meshing force of the first gear.](image2.png)

![Figure 15. 3-D Spectrum of meshing force of the second gear.](image3.png)
7. Conclusion
Based on ADAMS, a four-stage 2Z-X (a) planetary gear train is simulated. The research focuses on the specific analysis of the rotational speed and meshing force vibration of the gear train. The research is divided into fixed speed and variable speed. The results are not only in time domain, but also in frequency domain. The research provides the basis for the follow-up design and optimization.

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