Simulation and Test of Spindle Mode of Planetary Variable Speed Device

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Abstract. Spindle is the main bearing part of high speed tracked vehicle transmission device, which needs to carry large power and torque. Modal testing can provide basis for the verification and modification of the dynamic model of transmission structure, response prediction, vibration control, qualitative analysis and dynamic coupling analysis of transmission system. In this paper, the mode of the spindle is simulated by finite element method, and the free mode of the spindle is tested by hammering method. The results show that the modal mode simulated by finite element method is consistent with the mode tested, and the error range between the measured modal frequency and the simulated modal frequency is less than 3%. The simulation and test of spindle mode have important theoretical significance and engineering application value for realizing the optimal design of spindle.

1. Introduction

The function of the high-speed tracked vehicle transmission device is to transfer the engine power to the active wheel to realize the functions of acceleration, steering and braking of the vehicle under complex road conditions. The spindle is the main bearing part of the transmission device, which needs to carry large power and torque [1]. The transmission spindle is made of super high strength steel for aerospace, which can easily cause self-excitation and forced vibration of the spindle under the action of large self-rotation force [2]. At the same time, there is coupling between the spindle and the tracked vehicle to bear the load, which is easy to occur the fracture of the spindle and lead to the loss of mobility of the tracked vehicle [3]. In order to make the spindle of planetary variable speed mechanism have high stiffness, small vibration and noise excellent performance such as low sound and high reliability, in-depth study of the dynamic performance of the spindle, and then realize the optimization design of the spindle, which has important theoretical significance and engineering application value [4].

Modal testing can provide the basis for the verification and modification of the dynamic model of the transmission structure, response prediction, vibration control, qualitative analysis and dynamic coupling analysis of the transmission system [5]. Domestic and foreign scholars have carried out a lot of research on the modal testing and analysis of the spindle. Yu [6] carried out modal analysis of the spindle system of ultra-high speed grinding machine tools. The results show that the error of modal test and simulation mainly comes from the simplification of support model. Zhou [7] and others used BP neural network to fit the functional relationship between the low-order natural frequency of the...
electric spindle and the random variables, and established the reliability limit state equation of the motorized spindle frequency.

Although domestic and foreign researchers have done a lot of research on spindle modes, they mainly focus on fixed mode testing in the field of machine tools, and have less research on spindle free mode of variable speed mechanism. In this paper, free mode testing and simulation analysis are carried out to verify the accuracy of spindle free mode testing and analysis.

2. Principle of modal analysis
Modal analysis is the basis of modern structural dynamics, that is, the physical coordinates in the vibration differential equations of linear time-invariant systems are transformed into modal coordinates, and the equations are decoupled into a set of independent equations described by modal coordinates and modal parameters.

2.1. Single degree of freedom system
For viscous damping systems, the vibration differential equation is as follow:

\[ m \ddot{x} + c \dot{x} + kx = f(t) \]  

The equation of free vibration is:

\[ m \ddot{x} + c \dot{x} + kx = 0 \]  

Its regular form is:

\[ m \ddot{x} + \sigma \dot{x} + \sigma_0^2 x = 0 \]  

Where \( \dddot{x} \) is acceleration; \( \dot{x} \) is velocity; \( x \) is displacement; \( \sigma = \frac{c}{2m} \) is attenuation coefficient (attenuation index); \( \sigma_0 = \sqrt{\frac{k}{m}} \) is undamped natural frequency (natural frequency).

Equation (1) introducing damping ratio (dimensionless damping coefficient):

\[ \xi = \frac{\sigma}{\sigma_0} = \frac{c}{2\sqrt{mk}} \]  

The differential equation of motion is deformed as follows:

\[ \dddot{x} + 2\xi \sigma_0 \dot{x} + \sigma_0^2 x = 0 \]  

Its general solution (free vibration response) is expressed as:

\[ x = Ae^{-\sigma t} \sin(\omega_0 t + \theta) \]  

2.2. Frequency response function
In Fig. 1, \( f(t) \), \( h(t) \), and \( x(t) \) are input, transfer, and output functions in the time domain (\( F(\omega) \), \( H(\omega) \), and \( X(\omega) \) are input, transfer, and output functions in the frequency domain.

\( x(t) \) is the output excited by \( f(t) \), and the expression is as follows:
Figure 1. System transfer function diagram

\[ x(t) = \int_{-\infty}^{\infty} h(\tau) f(t-\tau) d\tau \]  

By Fourier transform on both sides of formula (7), the displacement frequency response function \( H(\omega) \) of single degree of freedom system can be obtained. The vibration of single degree of freedom system is as follows:

\[ H(\omega) = \frac{X(\omega)}{F(\omega)} = \frac{1}{k - m\omega^2 + jc\omega} \]

\[ = \frac{1}{k} \left[ \frac{1 - \lambda^2}{(1 - \lambda^2)^2 + (2\xi\lambda)^2} + j \frac{-2\xi\lambda}{(1 - \lambda^2)^2 + (2\xi\lambda)^2} \right] \]  

Among them, \( \lambda = \frac{\sigma}{\sigma_0} \) frequency ratio.

Similarly, there is a velocity frequency response function \( H_v(\omega) \):

\[ H(\omega) = \frac{j\omega}{k - m\omega^2 + jc\omega} \]  

Acceleration frequency response function \( H_a(\omega) \):

\[ H(\omega) = \frac{-\omega^2}{k - m\omega^2 + jc\omega} \]

The frequency response function reflects the transmission and amplification characteristics of the system to different frequency excitations, as well as the dynamic characteristics of the system in the frequency domain.

3. Modal Simulation and Test of Spindle of Planetary Transmission Mechanism

3.1. Modal finite element Modeling of Spindle

The finite element model of the spindle shell element is established. The model includes 766629 nodes and 520040 elements, of which the element size is 5mm and the simulated ambient temperature is 2222°C. The raw materials are made of ultra-high strength steel for aerospace. The specific chemical composition of the material is shown in Table 1, and the mechanical properties are shown in Table 2.
Table 1. Chemical composition of spindle raw materials

|        |  C   |  Mn  |  Si  |  P   |  S   |  Cr  |  Ni  |  Mo  |  V   |
|--------|------|------|------|------|------|------|------|------|------|
| Standard requirements | 0.38-0.43 | 0.60-0.90 | 1.45-1.80 | ≤0.010 | ≤0.010 | 0.70-0.95 | 1.65-2.00 | 0.30-0.50 | 0.05-0.10 |

Table 2. Mechanical properties

|                | Tensile strength Rm (MPa) | Yield strength Rp0.2 (MPa) | Elongation rate A (%) | Section shrinkage rate Z (%) |
|----------------|---------------------------|---------------------------|-----------------------|-----------------------------|
| Standard requirements | ≥1860                     | ≥1515                     | ≥8                    | ≥30                         |

3.2. Spindle Measurement Steps

The transfer function modal test technique is used to measure the coordinates of the structural specimens by means of multi-point excitation, and the frequency response (i.e., transfer function) with force as input and response acceleration as output. When the transfer function method is used to carry out structural modal test, it mainly includes two main links: one is the frequency response test of the structure, the other is to determine the modal parameters by the system identification.

Spindle free mode testing tools are mainly force hammer, acceleration sensor and LMS dynamic data acquisition system. LMS software is used to analyze the modal mode. Modal test first establishes the component model, arranges the measuring point and the excitation point; records the excitation signal of the excitation point and the three direction acceleration signal of the measuring point after the force hammer excitation.

Eight measuring points are arranged on Fig.2 (a) and (b). Among them, the hammering point is arranged in the red mark of the Fig.2 (b), and the hammering direction is radial and axial respectively.

![Figure 2. Layout of the measured sensor in the planetary frame](image)

3.3. Analysis of Simulation and Test results

The modes of the spindle are calculated by Ansys software, in which the 1-6 modes are rigid body modes, and the rigid body modes are six rigid body motions of the principal axis, including three directions of translation and three directions of rotation, which is a rigid body model with a vibration frequency of 0. The seventh order begins to be the vibration mode, and the comparison between the simulation mode and the test mode is shown in Fig.3. Through the comparison, it is concluded that the simulation mode is consistent with the test mode.
The comparison between the analytical modal frequency and the test modal frequency is shown in Table 3, and it is concluded that the error range between the measured modal frequency and the simulated modal frequency is less than 3%.

| Order | Simulation modal frequency /HZ | Measured modal frequency /HZ | Error | Order | Simulation modal frequency /HZ | Measured modal frequency /HZ | Error |
|-------|--------------------------------|------------------------------|-------|-------|--------------------------------|------------------------------|-------|
| 7     | 240.85                         | 239.76                       | 0.45% | 14    | 1932.40                        | 1901.16                      | 1.62% |
| 8     | 241.13                         | 239.76                       | 0.57% | 15    | 1934.50                        | 1901.16                      | 1.72% |
| 9     | 649.11                         | 636.81                       | 1.89% | 16    | 2290.80                        | 2272.37                      | 0.80% |
| 10    | 649.76                         | 636.81                       | 1.99% | 17    | 2775.10                        | 2727.34                      | 1.72% |
| 11    | 1226.90                        | 1204.70                      | 1.81% | 18    | 2779.10                        | 2727.34                      | 1.86% |
| 12    | 1228.00                        | 1204.70                      | 1.90% | 19    | 3014.90                        | 3044.07                      | 0.97% |
| 13    | 1465.30                        | 1458.59                      | 0.46% | 20    | 3750.10                        | 3687.02                      | 1.68% |

4. Conclusions
In this paper, the modal test of the spindle of the planetary variable speed mechanism is carried out by hammering method, and the matrix of the spindle modal test is compared with that of the analysis by finite element method, and the sensitivity and correlation of the modal are analyzed.

1. After processing the test data and comparing with the finite element simulation results of Ansys software, the simulation mode is consistent with the test mode.
2. The error range between the measured modal frequency and the simulated modal frequency is less than 3%.

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