

Research on dynamic characteristic of planetary gear system based on bond graph method

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Abstract. The dynamic characteristics of planetary system directly affect the stable state of the whole transmission system. Firstly, an accurate dynamic model of planetary gear system is established by using bond graph method. Formerly, the state equation of the gear train dynamics is obtained based on the bond graph model. On this basis, the simulation model of the gear system was acquired by using the numerical simulation method, and four dynamic indexes including zero-pole diagram, Bode diagram, Nyquist diagram and Nichols diagram of the gear dynamics system were gained. Finally, the analysis results show that the accurate mathematical model is described by the bond graph method and the dynamic characteristics is comprehended of planetary gear system, while will provide a basis for the stability research of the transmission system.

Keywords: planetary gear system, bond graph method, dynamics equation of state, dynamic characteristics.

1. Introduction

The dynamic characteristics of planetary gear system will directly affect the healthy and stable running state of the whole transmission system [1]. Due to the existence of ulnar clearance, time-varying meshing stiffness and comprehensive transmission error in planetary gear system, the rules of dynamic characteristics of planetary gear system are extremely complex, which brings great challenges to the kinematics and dynamics design, as well as the reasonable design and accurate control of system stability [2, 3].

The dynamics of planetary gear transmission system, focusing on finding accurate and efficient analysis model, and exploring the dynamic response and excitation mechanism of the system [4, 5]. The dynamic characteristics and system stability play a very important role in advanced design and fatigue analysis. However, the traditional gear train modeling method starts from the perspective of simplification and equivalence, thus some important factors affecting the system are omitted [6]. As a result, the dynamic characteristics of the gear system are much different from the actual situation, and lost its guiding significance for practical application.

2. The bond graph model of planetary gear system

This paper takes NGW gear train as the research object. According to the bonding diagram modeling principle, the bond graph model of planetary gear system is obtained, as shown in Fig. 1.

3. Dynamics equation of state for gear system

State variables $\mathbf{Y} = [p_2, q_6, p_{11}, p_{17}, p_{22}, p_{26}, q_{30}, p_{35}, p_{41}, p_{46}, p_{51}, p_{56}, p_{62}]$. According to gear train bond graph model, the state equation used to describe the motion state of the epicyclic unit was obtained. When $\dot{p}_{17} = e_{17}$, $\dot{p}_{22} = e_{22}$, $\dot{p}_{41} = e_{41}$, $\dot{p}_{46} = e_{46}$, $\dot{p}_{51} = e_{51}$, $\dot{p}_{56} = e_{56}$, as shown in Eq. (1-6):

$$\dot{p}_2 = Se_1 - \frac{R_7}{(m_1)^2 I_2} p_2 - \frac{1}{m_1 C_6} q_6 + \frac{m_2 R_7}{m_1 I_{11}} p_{11} - \frac{R_7}{m_1} Sf_8, \quad (1)$$

$$\dot{q}_6 = \frac{1}{m_1 I_2} p_2 - \frac{m_2}{I_{11}} p_{11} + Sf_8, \quad (2)$$

$$\dot{p}_{26} = Se_{25} - \frac{1}{m_3 C_{30}} q_{30} - \frac{R_{31}}{(m_3)^2 I_{26}} p_{26} - \frac{R_{31}}{m_3} Sf_{32} + \frac{m_4 R_{31}}{m_3 I_{35}} p_{35}, \quad (3)$$

$$\dot{q}_{30} = \frac{1}{m_3 I_{26}} p_{26} - \frac{m_4}{I_{35}} p_{35} + Sf_{32}, \quad (4)$$

$$\begin{aligned} \dot{p}_{11} = & \frac{m_2 R_7}{m_1 I_2} p_2 + \frac{m_2}{C_6} q_6 - \frac{(m_2)^2 R_7}{I_{11}} p_{11} + \frac{m_5 (m_6)^2 R_{67}}{I_{17}} p_{17} + \frac{m_5 m_6 m_8 R_{67}}{I_{22}} p_{22} \\ & - \frac{m_5 m_6 R_{67} + m_{10} m_{11} R_{65} + m_{14} m_{15} R_{66}}{I_{35}} p_{35} + \frac{m_{10} (m_{11})^2 R_{65}}{I_{41}} p_{41} + \frac{m_{10} m_{11} m_{12} R_{65}}{I_{46}} p_{46} \end{aligned} \quad (5)$$

$$\begin{aligned} & + \frac{m_{14} (m_{15})^2 R_{66}}{I_{51}} p_{51} + \frac{m_{14} m_{15} m_{16} R_{66}}{I_{56}} p_{56} + m_2 R_7 Sf_8 + m_5 \dot{p}_{17} + m_{10} \dot{p}_{41} + m_{14} \dot{p}_{51}, \\ \dot{p}_{35} = & - \frac{m_6 R_{67}}{I_{17}} p_{17} - \frac{m_8 R_{67}}{I_{22}} p_{22} + \frac{m_4 R_{31}}{m_3 I_{26}} p_{26} + \frac{m_4}{C_{30}} q_{30} + m_4 R_{31} Sf_{32} - \frac{m_{11} R_{65}}{I_{41}} p_{41} \\ & - \frac{m_{12} R_{65}}{I_{46}} p_{46} + \frac{R_{67} - (m_4)^2 R_{31} + R_{65} + R_{66}}{I_{35}} p_{35} - \frac{m_{15} R_{66}}{I_{51}} p_{51} - \frac{m_{16} R_{66}}{I_{56}} p_{56}, \end{aligned} \quad (6)$$

where, m_i is the conversion coefficient of the converter, $i = 1, 2, \dots, 22$; Se_1 and Se_{25} are potential sources of input; Sf_8 and Sf_{32} are steady-state transmission errors.

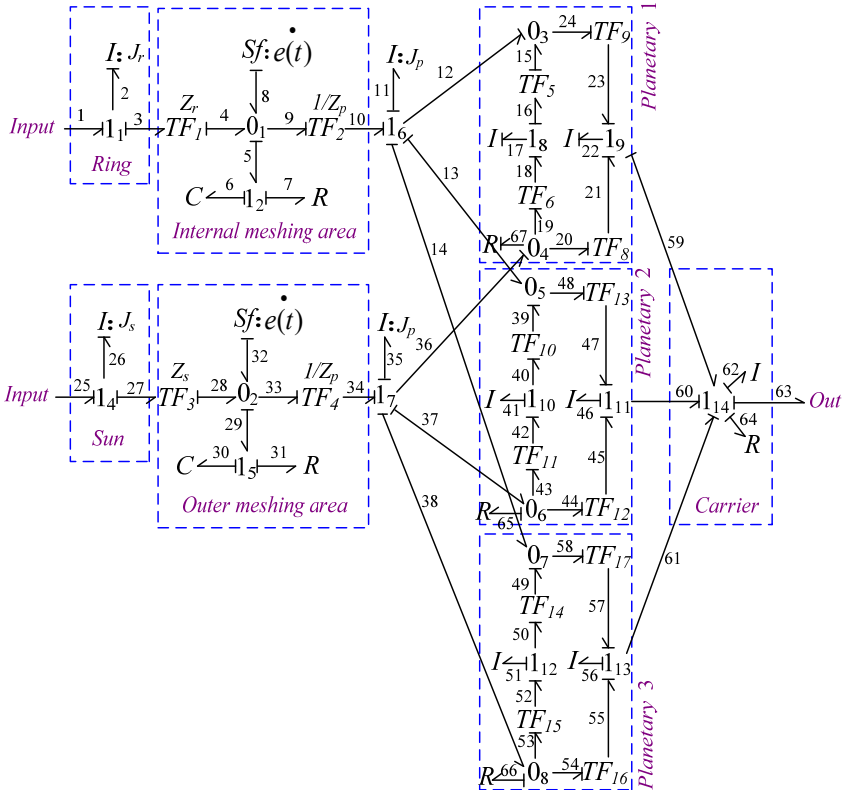


Fig. 1. The bond graph model of planetary gear system

4. Numerical simulation and discussion

4.1. Simulation condition setting

The simulation model is shown in Fig. 2. MS_{e1} and MS_{e2} are as the input end, MSf_8 and MSf_{32} are used to represent the derivatives of steady transmission errors, as show in Table 1.

Table 1. Simulation parameter

Number of teeth	$Z_s = 19, Z_p = 31, Z_r = 81$	Radius of indexing circle	$R_s = 0.0285 \text{ m}, R_p = 0.0465 \text{ m}, R_r = 0.1215 \text{ m}, R_h = 0.075 \text{ m}$
Moment of inertia	$I_s = 0.0015 \text{ Kg}\cdot\text{m}^2,$ $I_p = 0.0094 \text{ Kg}\cdot\text{m}^2,$ $I_r = 0.1510 \text{ Kg}\cdot\text{m}^2,$ $I_h = 0.0491 \text{ Kg}\cdot\text{m}^2$	Meshing stiffness	$K_g = 1.8e8 \text{ N/m}$
		Torsional stiffness of shaft	$K_s = 2.0e8 \text{ N/m}$
		Moment of inertia of shaft	$I_f = 0.0036 \text{ Kg}\cdot\text{m}^2$
Meshing damping	$C_g = 2000 \text{ N}\cdot\text{s}\cdot\text{m}^{-1}$	Support damping	$C_v = 1800 \text{ N}\cdot\text{s}\cdot\text{m}^{-1}$
Derivative of steady state transmission error		$\dot{e}(t) = 0.0027 \times \cos(28t + 21\pi)$	

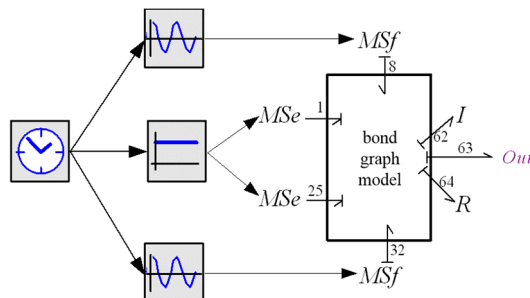


Fig. 2. Simulation model of planetary gear system

4.2. Analysis of simulation results

4.2.1. Gear ring input

It was be comprehended from Fig. 3(a) that the open loop transfer function of the system is a non-minimum phase system. The Bode diagram shown in Fig. 3(b), the phase margin $\gamma(\omega_c) = \infty$, and the amplitude margin $K_g = 35.9 \text{ dB} > 0$ (7.09 rad/s). The Nyquist diagram shown in Fig. 3(c), the open-loop frequency characteristic curve of the system does not enclose the point $(-1, j_0)$. Nichols diagram shown in Fig. 3(d), the vertical axis is the logarithm of amplitude and the horizontal axis is the phase Angle, which reflects the changing relationship between amplitude and phase Angle. The closed-loop system of planetary gear train is unstable under such simulated conditions.

4.2.2. Sun gear input

Similarly, it is be concluded from Fig. 4 that the closed loop system of the planetary gear train is unstable.

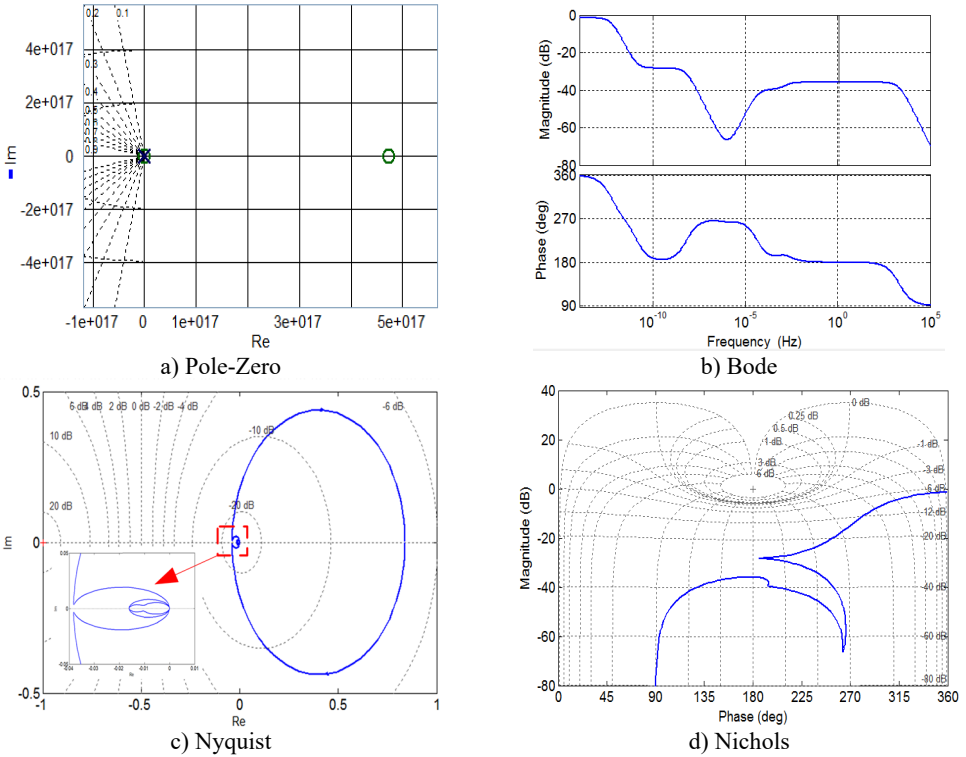


Fig. 3. The stability of planetary gear system

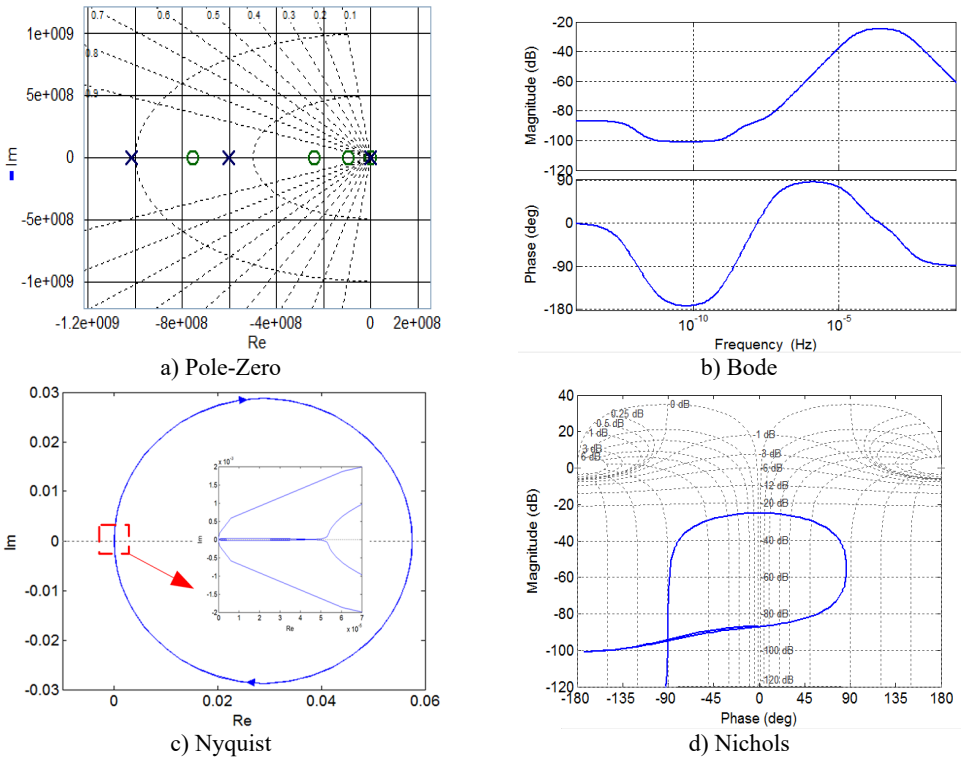


Fig. 4. The stability of planetary gear system

5. Conclusions

The bond graph model of the planetary gear system was established, which solved the problem that traditional modeling method could not accurately reflect the dynamic characteristics of the gear system. Then, state space and state equation of gear train dynamics are derived on the basis of bond graph model. Zero-pole diagram, Bode diagram, Nyquist diagram and Nichols diagram, are calculated by using numerical simulation method, and the dynamic rule of the gear system is reflected.

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