Dynamic behavior of planetary spur gear train affected by tooth clearance and manufacturing deviations based on Jacobian-torsor deviation model

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Abstract

The assemble deviation is inevitable for planetary spur gear systems due to the fabrication deviation of parts, which has a significant impact on their vibration and noise performance. The deviation for the assembled structure is thought as the part of meshing error and is introduced into the dynamic model to investigate the dynamic performance of gear system. However, the relationship between the deviation of parts such as pinion and axis and dynamic performance is rarely considered. The results have limitation on the application of dimension control in the manufacture of parts.

In this study, the deviations of parts are described by using the torsor vectors with three translational components and three rotational components. The Jacobiano-torsor deviation model for gear system is proposed to predict the final deviation for the assembled gear system. The tooth side clearance can also be obtained by combining the empirical formula of tooth clearance and the assembled deviation. Then, the manufacturing deviations and tooth clearances are projected to the tooth meshing deformations in the meshing direction and transformed into equivalent meshing deformations. The dynamic model for the planetary spur gear train is established by considering the equivalent meshing deformations. The relationship between the dynamic behavior and the manufacturing deviations of parts is discussed.

The equivalent meshing deformations for the axial position error and the axial angle error are proposed as

$$\begin{cases} E_{spcpi} = E_{cpi}\sin\left(\alpha_{sp} - \psi_{pi}\right)\cos\beta_b \\ e_{Aspi} = -A_{pi}\sin\left(\alpha_{sp}\right) - A_s\sin\left(\psi_{pi} + \alpha_{sp}\right) \end{cases} \tag{1}$$

where the axial pitch error E_{cpi} and axial angle error A_i can be calculated by the assembled deviation of gear system with the torsor $T = \begin{bmatrix} u & v & w & \alpha & \beta & \gamma \end{bmatrix}^{\mathrm{T}}$. They are expressed as

$$\begin{cases} E_{cpi} = a = \dfrac{\left|\left(\mathbf{Ri}^{\mathrm{T}} \times \mathbf{i}^{\mathrm{T}}\right)\cdot\mathbf{t}\right|}{\left|\mathbf{Ri}^{\mathrm{T}} \times \mathbf{i}^{\mathrm{T}}\right|} \\ A_i = l_i\theta_i = l_i \arccos\left(\left(\mathbf{Ri}^{\mathrm{T}}\right)\cdot\mathbf{i}^{\mathrm{T}}\right) \end{cases} \tag{2}$$

where $\mathbf{i}^T = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix}^T$, $\mathbf{R} = \begin{bmatrix} 1 & -\gamma & \beta \\ \gamma & 1 & -\alpha \\ -\beta & \alpha & 1 \end{bmatrix}$, and $\mathbf{t} = \begin{bmatrix} u & v & w \end{bmatrix}^T$.

The tooth clearance is introduced as a parameter of the nonlinear meshing deformation function and denoted as

$$f(x,b)=\begin{cases} x-b/2 & (x>b/2) \\ 0 & (-b/2\leqslant x\leqslant b/2) \\ x+b/2 & (x<-b/2) \end{cases} \tag{3}$$

The lump-mass parameter model for the planetary spur gear system is shown in Figure 1 and the dynamic model is established. The moving components each gear contains 3 in-plane degrees of freedom. The clearance b_{spi} is introduced as a nonlinear multiplier, and the manufacturing error e_{spi} is introduced as an additive term.

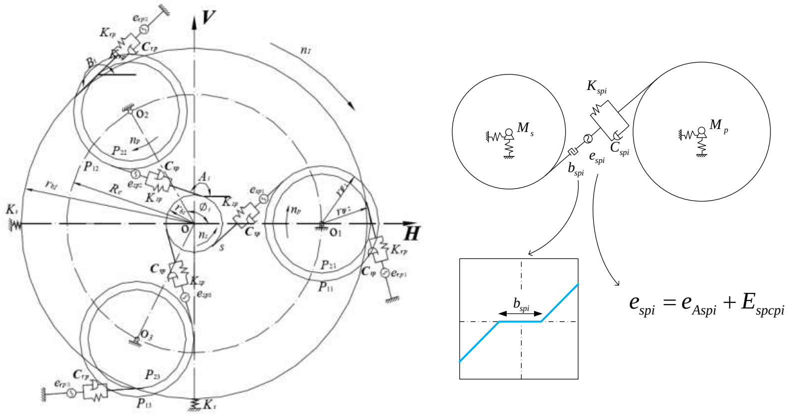

Figure 1. The lump-mass parameter model for the planetary spur gear system.

The dynamic response is calculated with different axial clearance b and parallelism tolerance t. The spectrum-power density diagram of sun gear rotation response is shown in Figure 2 and the vibration response parameter diagram is shown in Figure 3. Obviously, the tooth clearance is positively correlated with the vibration intensity and inversely correlated with the vibration peak frequency. The tooth clearance with the manufacturing deviation significantly increases the vibration amplitude and significantly increases the percentage of low frequency components in the vibration spectrum. Meanwhile, the increase of the tooth clearance may lead to a further decrease of the peak frequency. The relationship between the parallelism tolerance and the vibration intensity is positive and has little effect on the vibration peak frequency. The increase of the tolerance leads to further increase of the amplitude, but there is no obvious further change of the power-frequency distribution.

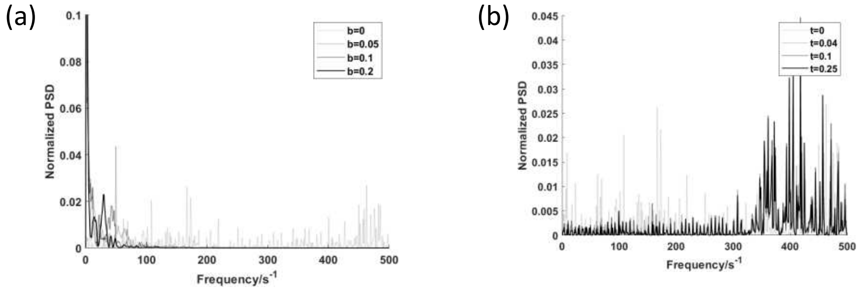

Figure 2. Spectrum-power density distributions of sun gear, (a) different clearances; (b) different tolerances.

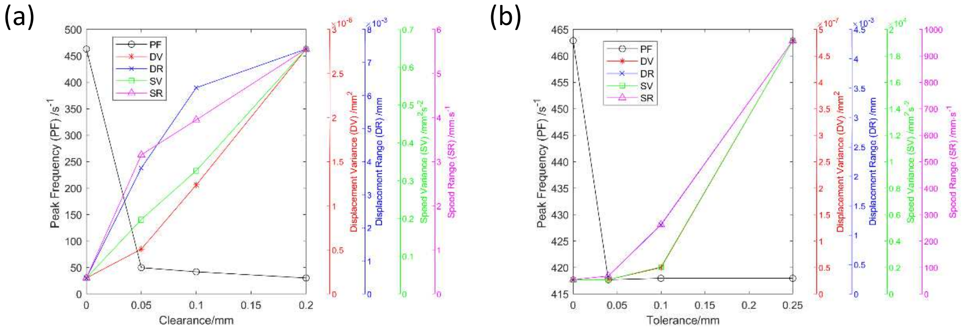

Figure 3. Vibration response parameters of sun gear rotation response, (a) different clearances; (b) different tolerances.