

Highlights

A vibration signal model of planetary gearboxes with unequal load sharing among planets

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A vibration signal model of planetary gearboxes with unequal load sharing among planets

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ABSTRACT

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1. Introduction

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2. Signal model

1. Planetary gearboxes have different configuration with similar layouts.
2. In this paper, we only consider the case when ring gear is fixed. (refer to the paper mentioning the sun or carrier as the fixed central part)
3. The vibration sensor is mounted on the stationary ring gear.
4. The vibration mainly origins the the time-varying stiffness gear meshing. When the planet gears engage with the ring gear or sun gear, the number of the involved tooth varies with the relative rotation between gears. Thus, their contact stiffness changes at the time. If the transmission load proximately remain constant, this system is pure parametric excited. (Add a reference about instability)
5. Specially, the torque load transferred from the central components in planetary gearboxes is split into parallel paths formed by the planets. Because of the inevitable manufacturing and assembling errors of pinholes or bearings, there are usually differences in the position of pinholes and the shape of planets. The load sharing among planets is non-uniform (collect other reasons from the Singh's paper).
6. Another common phenomenon in planetary gearboxes is the difference in meshing phases between ring-planet and sun-planet. When this situation coincides with the load sharing inequality, a complex couple mechanism of time-varying load sharing emerges, rising extra vibrations.
7. For an individual planet, the vibration signal model can be written as

$$J \cdot \ddot{\theta}(t) + k(t) \cdot [\theta(t) - L(t)] = 0, \quad (1)$$


where J is rotational inertia of the machine, $\theta(t)$ is the shaft angular, $\ddot{\theta}(t)$ is the rotational acceleration, $k(t)$ is time-varying meshing stiffness, $L(t)$ is the external load.

8. Taking all planets into account, the above vibration model can be revised as

$$J \cdot \ddot{\theta}(t) + \sum_{i=1}^M \{k_i(t) \cdot [\theta(t) - L(t) + \epsilon_i]\} = 0, \quad (2)$$

where M is the number of planets, $k_i(t)$ is the meshing stiffness of i -th planet, ϵ_i is the angular shift away from its normal position of i -th planet (here we assume $\epsilon_2 \neq 0$). A translational spring-mass system illustrate the above vibration model in Fig. 1.

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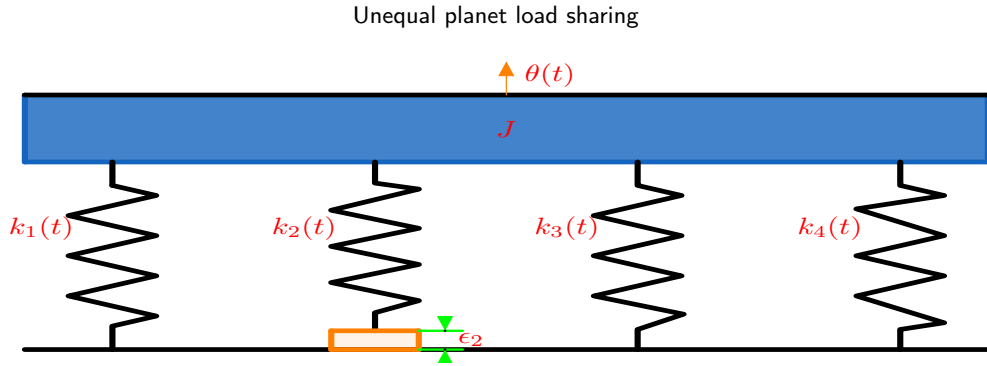


Fig. 1: Vibration system of unequal load sharing among four planets

A typical layout of planetary gearboxes consists of three kinds of gears mounted so that the centers of planet gears revolves around the center of sun and ring gears. As shown in Fig. 2.

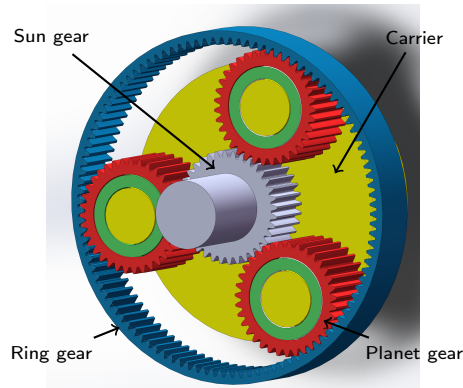


Fig. 2: Configuration of planetary gearboxes

2.1.

3. Factors considered in the signal model

1. Mechanism of time-varying gear meshing stiffness and risen impulses. (Including sun-planet and ring-planet)
2. Transfer path effect led by the carrier rotation.
3. Unequal load sharing risen by the planet angular position error.
4. Differences in the impulsive intensity of planets with sun and ring gears arisen from the position errors of planet. (Amplitude modulation of angular position error)
5. Phase shifts of transfer path effect of planet position error. (Frequency modulation of angular position error)
6. Effects of asynchronous meshing between sun-planet and ring-planet.
7. Effects of asynchronous meshing among sun-planets or ring-planets.
8. Natural vibration influence on spectral structure.

References