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Frequency analysis of a two-stage planetary gearbox using two different methodologies

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ABSTRACT

This paper is focused on the characterization of the frequency content of vibration signals issued from a two-stage planetary gearbox. To achieve this goal, two different methodologies are adopted: the lumped-parameter modeling approach and the phenomenological modeling approach. The two methodologies aim to describe the complex vibrations generated by a two-stage planetary gearbox. The phenomenological model describes directly the vibrations as measured by a sensor fixed outside the fixed ring gear with respect to an inertial reference frame, while results from a lumped-parameter model are referenced with respect to a rotating frame and then transferred into an inertial reference frame. Two different case studies of the two-stage planetary gear are adopted to describe the vibration and the corresponding spectra using both models. Each case presents a specific geometry and a specific spectral structure.

1. Introduction

Planetary gear sets are used commonly in many industrial, automotive, aerospace and wind turbine gearbox applications. The complexity of the dynamic behavior of the planetary gearbox (PG) has been actively investigated using a model based on experimental approaches. Two different kinds of models have been implemented so as to describe the vibrations generated: PG (*i*) lumped-parameters models, and (*ii*) phenomenological models. The lumped-parameter models found in the literature describe the vibrations of all degrees of freedom referenced with respect to a rotating frame fixed to the carrier plate. However, phenomenological models describe directly the vibrations as measured by a sensor fixed outside the fix ring gear, which is subjected to periodic variation in vibration amplitudes when the planets pass through this fixed sensor. The two types of models should highlight the amplitude modulation (AM) of vibration time histories and modulation sidebands in the frequency domain induced by the time-varying vibration transmission path.

For the first type of models, the first study is that of Cunliffe et al. [1] who developed a model with a 13-degree of freedom to analyze the frequencies and mode shapes with a single fixed carrier. Then, Saada and Velex [2] developed the equations of motion of the system by the Lagrange method. Later, many research works have been done by Lin and Parker [3,4] and Chaari et al. [5,6] to describe the modal analysis and the dynamic response of the system.

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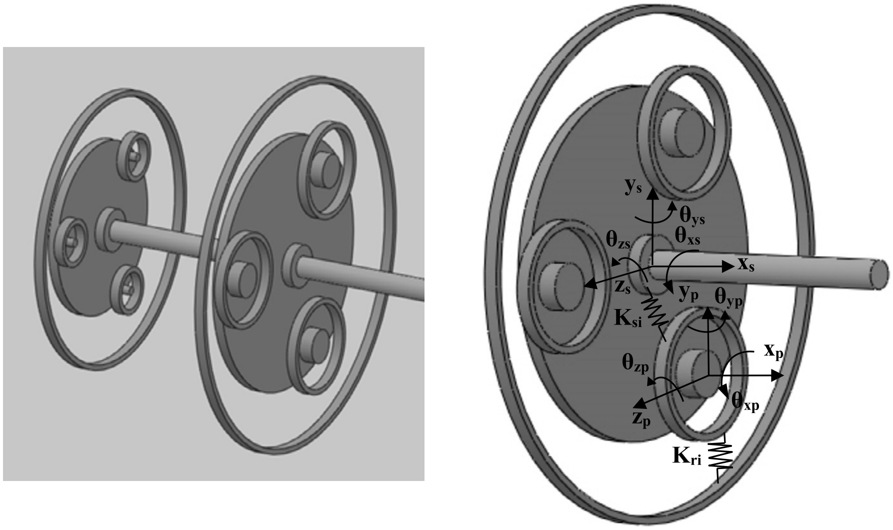

Fig. 1. Lumped parameter model of two stages planetary gear.

Concerning the second kind of models, McFadden and Smith [7] were the first to highlight the asymmetric distribution of modulation sidebands around mesh frequency and harmonics. Their model was able to predict the frequency content of the vibration signal issued from a planetary gear set. Later, McNames [8] explored the relative amplitudes of the dominant peaks using continuous-time Fourier series. Inalpolat and Kahraman [9] developed a simplified analytical model to describe the amplitude modulation of planetary gear sets. This model shows that there are different classes of planetary gear sets that exhibit different sideband behaviors. They also validated these trends through an experimental study. Molina [10] developed a phenomenological model that takes into account the variable vibration transmission paths through the ring gear and through the sun gear and carrier plate, something that had been missed in all previous publications. Samuel and Pines [11] described a technique based on the selection of an appropriate window function used to analyze the vibration signals collected from multiple sensors located around the ring gear. For the case of complex PG sets, one can find works dedicated to compound or multistage PG, which are based on lumped-parameter models. Thomas [12] developed an analytical model for investigating the transmission error and load distribution of a double helical gear pair. Zhang et al. [13] established a translational–rotational coupled dynamic model of a two-stage closed-form planetary gear set to predict the natural frequencies and vibration modes. Recently, Karray et al. [14] presented a complex configuration of a gearbox used in a bucket wheel excavator gearbox to investigate its modal properties. These models are difficult to implement and require a lot of care when computing the dynamic response. Phenomenological models can be an interesting alternative way to describe in a simple manner this behavior.

In this context, this paper will be concerned with developing two models of two-stage helical planetary gear using both the lumped-parameter model and the phenomenological model. Two case studies of two-stage planetary gear that differ from the point of view of geometry and assembly characteristics are developed, and the results obtained from each model will be presented and compared.

2. The lumped-parameter model

The studied gearbox is composed of a two-stage helical PG. Each stage is comprised, as shown by Fig. 1, of a ring gear (r) coupled with a sun gear (s) by N planets (P_n) and mounted on a carrier (c). All of these elements are considered as rigid bodies supported by elastic bearing. Meshing phenomena are approached by linear springs acting on the lines of action [15].

First, the equation of motion of each component is derived separately, and then assembled to obtain the overall system matrices of an N-planet helical PG train. Each component has six degrees of freedom: three translations (u_j, v_j and w_j) and three rotations (φ_j, ψ_j and θ_j, $j = \mathrm{c, r, s}, 1 \ldots n$). These coordinates are measured with respect to a frame $(O, \vec{s}_1, \vec{t}_1, \vec{z}_1)$ fixed to the carrier and rotating with a constant angular speed Ω_c. The rotations (φ_j, ψ_j and θ_j) are replaced by their corresponding translational displacements as:

$$\rho_{jx} = R_{\mathrm{b}j}\varphi_j, \quad \rho_{jy} = R_{\mathrm{b}j}\psi_j, \quad \rho_{jz} = R_{\mathrm{b}j}\theta_j, \quad j = \mathrm{c, r, s}, 1, .., n \tag{1}$$

where $R_{\mathrm{b}j}$ is the base circle radius for the sun, the ring, the planet, and the radius of the circle passing through planet centers for the carrier. Circumferential planet locations are specified by the fixed angles α_i, which are measured relatively to the rotating basis vector $\vec{s}_1$, so that $\alpha_1 = 0$.

Then the coupling between the two stages is done using an additional torsional stiffness joining the rotational degree of freedom of the carrier wheel of the first stage and the sun gear of the second one and an additional linear spring joining the axial degrees of freedom of the same carrier and the sun.

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