Journal of Sound and Vibration 333 (2014) 6498–6515

Contents lists available at ScienceDirect

Journal of Sound and Vibration

journal homepage: www.elsevier.com/locate/jsvi

Dynamic characteristics of the herringbone planetary gear set during the variable speed process

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ARTICLE INFO

Article history:
Received 3 April 2014
Received in revised form
14 July 2014
Accepted 18 July 2014
Handling Editor: L.G. Tham
Available online 15 August 2014

ABSTRACT

In this study, a dynamic model for herringbone planetary gears is proposed which can be applied in the dynamic analysis of variable speed processes (including acceleration, deceleration, and large speed fluctuation process, etc.). The dynamic responses of the acceleration process of an example of a herringbone planetary gear set are simulated in cases where the profile error excitations are ignored and included. The phenomenon of tooth separations can be observed as the rotating speed increases in the simulation, and the effect of the profile error excitations on the phenomenon is also investigated. Furthermore, the effects of the profile error excitations on the vibrations and dynamic meshing forces are investigated before and after the appearance of tooth separations. Moreover, the dynamic characteristics of the herringbone planetary gear set are also compared with that of the spur/helical herringbone planetary gear set briefly. Finally, some advice for the design of planetary gear sets is given to avoid the phenomena of tooth separation and tooth back contacts and suppress the vibrations and dynamic meshing forces.

1. Introduction

Because of advantages such as high power density, compactness, multiple and large gear ratios, and load sharing among planets, planetary gears are widely used in power transmission for many applications, such as automobiles, wind turbines, and long-wall shearers. Herringbone gears offer advantages such as a smooth transmission, great transmission torque, and low axial force; hence they replace spur gears to construct the planetary gear set in high-capacity gear transmissions. The planetary gear sets on automobiles, wind turbines, and long-wall shearers usually work under variable speed processes, so it is essential to construct a dynamic model of variable speed process for the herringbone planetary gear set to obtain dynamic characteristics during the variable speed process.

Many dynamic models for planetary gear set are proposed in the studies. Kahraman [1] proposed a simplified purely torsional model of a single stage planetary gear set to obtain torsional natural frequencies and compared it with the natural modes obtained from a more sophisticated transverse-torsional model. Lin [2] proposed an analytical model of planetary gears which includes key factors affecting the vibration such as the gyroscopic effects and time-varying stiffness. Guo [3] proposed a dynamic model of planetary gear set involving tooth wedging and bearing clearance nonlinearity. Kim [4]

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http://dx.doi.org/10.1016/j.jsv.2014.07.024

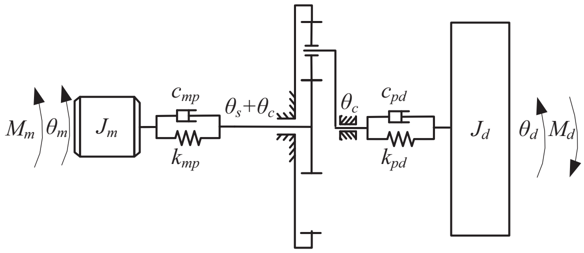

Fig. 1. Electromechanical dynamic model of the planetary gear transmission system (M_i, θ_i, and J_i ($i=m$, d) are the torque, angular displacement, and rotational inertia of the motor (m) and driven part (d), respectively).

proposed a dynamic model of planetary gear set in which the pressure angles and contact ratios are time-varying. Eritenel [5] proposed a three-dimensional helical planetary gear model to study the modal properties and categorized the vibration modes into three types. Bu [6] developed a generalized dynamic model for herringbone planetary gear set to investigate its modal properties and classify its vibration modes into five categories. Sondkar [7] investigated the linear time-invariant (LTI) model, the LTI model with gyroscopic effects included, and the nonlinear time-varying (NTV) model of double-helical planetary gear set in his doctoral dissertation. The assumption is explicitly or implicitly made that the position of the contact lines is determined by the mean angular motions of the gear pair in previous studies [1–7]. The obvious manifestation of this assumption in the dynamic model of planetary gear set is that meshing stiffness is directly expressed as the periodic function of time, but meshing stiffness is actually the periodic function of the gear rolling angle because the mean angular velocity is variable during the variable speed processes. These models [1–7] are proposed to mainly investigate the vibration properties of gear transmission systems and the vibratory translational and angular displacements [1–3,5–7] are usually chosen as generalized coordinates to construct the dynamic models. Therefore, these models [1–7] can just be used in vibration analysis of transmission systems at a stable mean operating angular velocity with small fluctuation, and not for variable speed processes. Moreover, it is an important trend to connect the transmission system and the electric motor to conduct electromechanical dynamic analysis for the optimization of the dynamic characteristics [8], fault diagnosis [9–11], and improvement of the electric motor control systems [12,13], but it is not convenient to connect these models [1–3,5–7] to the electric motor model (steady-state [14] or dynamic model [15]), because these models [1–3,5–7] choose the vibratory angular displacements as the generalized coordinates. At present, some dynamic models [16,17] of planetary gear set in non-stationary operations have been proposed based on these models [2] by modulating the meshing stiffness with the mean angular velocity [18,19]. The variation of the mean rotating speed of gear transmission must be known in advance before using these dynamic models for non-stationary operations [16,17], however, most of the time, the rotating speed is usually not known before the dynamic model is simulated.

In this study, a dynamic model for herringbone planetary gears is proposed which can be applied in the dynamic analysis for variable speed processes (including acceleration, deceleration, and large speed fluctuation process, etc.). The angular displacements are chosen as the generalized coordinates in this model, so it is also convenient to be connected to the electric motor model (steady-state [14] or dynamic model [15]) to conduct electromechanical analysis. The planetary gear set is first transformed into parallel-axis external and internal herringbone gear pairs in the moving coordinate system to obtain the meshing forces, and then Newton's law in non-inertial coordinate system is utilized to obtain the equations of motion of planetary gears. For the transformed parallel-axis herringbone gear pairs, the position of the contact lines is determined by the angular displacement of the driving gear, rather than by the mean angular motions, under this assumption, the meshing stiffness is the periodic function of the angular displacement of the driving gear and not relevant to the meshing frequency. The normal deviation from perfect teeth profile can introduce excitations, namely, profile error excitations, into the dynamic model, which can be projected in the directions of the action line and rotating axis of the herringbone gears. The profile error excitation is usually expressed by a Fourier series with the fundamental frequency equal to the tooth mesh frequency [20–22] or not included in dynamic models [23] and assumed to be invariable along the contact line [20,21]. The profile error excitations are usually variable along the contact line because of the randomness of manufacture error or longitudinal modification [24]. Ajmi and Velex [25,26] consider the profile error excitations depend on the position along the face width and vary with time. Theodossiades [27] considers the error excitation to be determined by the angular displacement of gear pair to study the dynamic characteristics of motor-driven gear-pair systems. Because the proposed model in this study is for variable speed processes, the profile error excitations are assumed to vary with both the angular displacement of the driving gear (not time) and the position along the contact line; furthermore, the value of profile error excitations is assumed to be normally distributed and simulated by pseudo-random numbers [28].

The tooth separation usually appears in gear transmission at high speed and light load and induces impacts to the gear transmission resulting in intense vibration and noise problems and large dynamic loads, which may affect the reliability and life of the gear drive [29,30], so tooth separation is given much attention in this study. In this study, the dynamic responses of the acceleration process of an example of a herringbone planetary gear set are simulated. The phenomenon of tooth separation and its effects on the dynamic responses are discussed; furthermore, the effect of the profile error excitation on the dynamic response is also investigated before and after the appearance of tooth separation. Moreover, the dynamic