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Research paper

A study of nonlinear excitation modeling of helical gears with Modification: Theoretical analysis and experiments

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ABSTRACT

A nonlinear hybrid dynamic model of a helical gearbox is proposed in this study. The model considers the nonlinear coupling effect of time-varying mesh stiffness (TVMS) and transmission error excitation. The effects of tip relief and lead crowning on the TVMS and dynamic characteristics of the helical gear transmission system are studied. Numerical methods are used to obtain the frequency response curves of the vibration acceleration of the helical gear system. The optimal values of tooth modification parameters are then determined in order to minimize the vibration amplitude. The simulation results indicate that the optimized tooth modification parameters can effectively decrease mesh stiffness fluctuation in the alternating areas of the teeth and reduce the vibration acceleration amplitude at its resonance frequency. Finally, the theoretical model is validated against an experimental platform of a high-speed rail traction gearbox transmission system, and the dynamic responses are compared.

1. Introduction

Helical gears are more widely used than spur gears across numerous industries and fields, including transportation, aerospace, military, shipbuilding, and power generation. The mesh stiffness and tooth errors of helical gears are three-dimensional (3D) space problems, and the calculation method of the mesh stiffness considering modification is different from that for a spur gear. The mesh stiffness of helical gears with modification cannot be accurately determined using traditional analytical methods. In addition, because of the presence of axial forces, dynamic modeling of the helical gear transmission system is more complex and requires more degrees of freedom (DOF) to be considered. Choosing an accurate nonlinear excitation modeling method for helical gears with modification is one of the problems that require solving. Selecting appropriate modification parameters can reduce the vibration and noise of the gear system. Therefore, it is necessary to study the effects of different modification parameters on the dynamic characteristics of the helical gear transmission system.

The calculation methods of the stiffness excitation include the material mechanics method, the finite element method (FEM), and the approximate substitution method. In the literature, gears are categorized as spur and helical gears. At present, experimental measurements, FEM, and the material mechanics method are primarily used for calculating the mesh stiffness of spur gears. Raghuwanshi and Parey [1] proposed various experimental techniques for measuring the meshing stiffness

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of spur gears. As a measurement method is time-consuming and its results are affected by the accuracy of the measuring equipment, a FEM was developed for a time-varying mesh stiffness (TVMS). Zhan et al. [2] developed a technique to determine the TVMS by utilizing NX, ANSYS, and a quasi-static algorithm. As the material mechanics method is faster and more efficient than FEM, it is predominantly used to calculate the TVMS of a gear. Fakher et al. [3] calculated the mesh stiffness of spur gears using the material mechanics method considering the bending, shear, radial compression, Hertz contact, and wheel body deformations. For calculating the mesh stiffness of helical gears, Chang et al. [4] determined mesh stiffness using a combination of FEM and local contact analysis of elastic bodies. Ajmi and Velex [5] separated a helical gear into slice models in the tooth-width direction using the section method. The material mechanics method was used to obtain the load distribution of each slice model while ignoring their axial potential energies. The gear mesh stiffness and elastic deformation were calculated based on the deformation compatibility equation. The results were verified using the FEM. The axial deformation of helical gears is neglected when solving their mesh stiffness, therefore, the traditional material mechanics method is not commonly used.

A number of researchers have conducted studies on gear rotor system dynamics modeling [6,7]. Based on the DOF, gear system dynamics models can be classified as pure torsional, flexural-torsional coupling dynamic, or flexural-torsional axial swing coupling dynamic models [8,9]. Of these, the flexural-torsional axial swing coupling dynamic model has a greater number of DOF than the other models, therefore, it can better reflect the actual conditions of a gear system. In the early modeling of gear dynamics, the errors and stiffness excitation were substituted separately into the dynamic model. With advances in research, the dynamic model of a gear system has been transformed from a linear model into a complex nonlinear model [10]. The current nonlinear time-varying model [11] can consider the effect of a TVMS, time-varying bearing stiffness, time-varying error excitation, time-varying impact excitation, and time-varying side clearance. Nonlinear dynamic models of gears, considering the bulk of factors, have recently become widely used. Guo et al. [12] established a gear-bearing-housing system and investigated the vibration propagation of gear dynamics by mathematical modeling and finite element (FE) analysis. Ma et al. [13] studied the effects of varying amounts of profile relief on the mesh stiffness of a spur gear by FEM and substituted it into the dynamic model of the gear system. The effects of different amounts of profile relief on the vibration response of the system were analyzed. Velex et al. [14] and Bruyère and Velex [15] presented a simplified analysis to evaluate the effect of profile modifications minimizing the transmission error. As a FEM has a low computational efficiency and is limited in the number of calculations, analytical and experimental methods were also applied. Chen and Shao [16] performed a dynamic analysis of a crowned gear transmission system with impact damping. Kia et al. [17] studied the torsional vibration effects on an induction current machine and torque signatures in a gearbox-based electromechanical system. Kang and Kahraman [18] and Hotait and Kahraman [19] investigated the dynamic behavior of gear sets by theoretical and experimental methods. Chen et al. [20], using experimental transmission errors, studied the dynamic characteristics of a crowned spur gear transmission system.

The bulk of existing literature only focuses on the effect of one type of modification method on the vibration characteristics of the gear system, and the object is typically a spur gear. Few studies have been conducted on the comprehensive influence of tip relief and lead crowning on the dynamic characteristics of the helical gear transmission system. This study presents a general analytical method for calculating the helical gear mesh stiffness and establishes the stiffness and errors by a nonlinear coupling incentive model that comprehensively considers the tip relief and lead crowning. Considering the structural flexibility of the gearbox, a housing element is introduced into the hybrid dynamic model of the helical gear rotor system by a semi-analytical method. The stiffness and error excitation are substituted into the hybrid dynamic model in order to study the effects of the tip relief parameters and lead crowning parameters on the TVMS and dynamic characteristics of the helical gear transmission system. The amplitude and frequency response curves of the vibration acceleration of the system are obtained by numerical methods and compared with the experimental results.

2. Analytical model of mesh stiffness considering modification

2.1. Improved calculation of contact line position and length of helical gears

The current methods for calculating the contact line length can be classified into two categories. One involves calculating the contact ratio of the helical gear. However, this method cannot determine the instantaneous position of the meshing point in the contact line. The other method is based on the spatial position of the meshing point of the helical gear. In a meshing period, the rotation angle of the driving gear is divided into n equal parts and the cartesian coordinates, $[x, y, z]$, of the meshing point in the contact line are obtained by solving the spiral involute equation and straight-line equation of the end surface. This method requires solving complex equations, and numerous divisions of the rotation angle implies a significant solution time. Therefore, this study proposes an improved calculation method for the helical gear contact line position and length.

The meshing surface of the helical gear is an involute helicoid. According to the meshing principle of the helical surface, the meshing surface of the helical gear can be regarded as the spiral motion of the involute around the z-axis. The spatial position expression of the meshing line of the helical gear is deduced based on the left helix, where involute AB moves to involute $A'B'$ around the z-axis, as shown in Fig. 1, where σ is the rotation angle between involutes AB and $A'B'$, r_b is the base circle radius, point E is any point on involute AB, r_E is the radius of point E, θ_E is the evolving angle of point E, and