

Effect of Component Flexibility on Axle System Dynamics

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ABSTRACT

The prediction and control of gear vibration and noise has become very important in the design of a quiet, high-quality gearbox systems. The vibratory energy of the gear pair caused by transmission error excitation is transmitted structurally through shaft-bearing-housing assembly and radiates off from exterior housing surface. Most of the previous studies ignore the contribution of components flexibility to the transmission error (TE) and system dynamic responses. In this study, a system level model of axle system with hypoid gear pair is developed, aiming at investigating the effect of the elasticity of the shafts, bearings and housing on TE as well as the contribution of flexible bearings on the dynamic responses. The load distribution results and gear transmission errors are calculated and compared between different assumptions on the boundary conditions. The modal characteristics and dynamic responses by assuming simply-support bearing and flexible bearing have also been compared. The results reveal that the system responses are sensitive to the system elasticity and shaft-bearing flexibility. The proposed approach yields a more in-depth understanding to the gear vibration propagation for use in the design of quiet driveline systems.

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INTRODUCTION

The vehicle interior sound quality has long been one of the major concerns for automotive industry. The axle gear whine is one of the major noise sources in a vehicle cabin, especially for four-wheel-drive (4WD) vehicles. The excitation of the axle gear whine is believed to be the gear transmission error (TE) by many researchers. The vibratory energy of the gear pair caused by transmission error is transmitted structurally through shaft-bearing-housing assembly and radiates off from exterior housing. Thus, the understanding of the source of excitation and the primary transmission path is the key to solve gearbox noise problem. Most of the earlier research work neglect the flexibility of other components, such as shaft, bearing, etc., or oversimplify the gear contact process and bearing support stiffness. Thus, the goal of this study is to provide an in-depth understanding of the role of flexible components played in the gear tooth deflection and dynamic responses.

A number of investigations have been made in the past to find out the cause of axle whine and identify the relationship between axle whine with the excitation force of the meshing gear pair. Yoon, et.al. [1]

measured transmission error of a hypoid gear pair and found that the hypoid gear whine noise can be minimized when the input torque is in the target range of the torque load for optimal transmission error. Koronias, et.al. [2] investigated on axle whine vibration and noise by a combined experimental and numerical approach. They compared the vibration modes of the drivetrain with experimental results and established a causal relationship between axle whine and the flexural mode response of system components. Most of the research papers on the practical work of axle whine reduction adopted tooth profile modification and structure modification as the means to achieve sound quality [3, 4, 5].

As the lightweight materials and design become a popular trend in the development of vehicles, the influence of component flexibility becomes increasingly important. The vibration characteristics of the main component on the structure transfer path, such as shafts, bearings, and housing, should be considered in the system model for maximum NVH improvements. Yang [6] and Wang [7] studied the influence of propeller shaft bending modes and analyze the gear-shaft interaction by using component mode synthesis (CMS). The same

method was also applied to investigate the influence of housing flexibility on hypoid gear dynamic responses [8]. Ognjanović and Kostić [9] used a combined approach of theoretical, numerical and experimental analyses to explain the process of spreading disturbance power through the elastic housing and found that the gear unit housing has a dominant role in the transformation of disturbance power and modulation of the sound emitted to the surroundings. Vanhollenbeke [10] investigated the structure-borne and airborne noise radiated from a wind turbine gearbox by using multi-body gearbox model and an acoustic FE model. They found that the flexibility of the housing will make a difference in both mode shapers and eigenfrequencies and thus is not negligible for evaluating the noise, vibration and harshness (NVH) responses of wind turbine gearboxes. Jauregui, et.al. [11] studied the effect of housing stiffness on nonlinear synchronization of gearbox elements. It was observed the dynamic load is increased when gears and rolling bearings synchronize for a soft housing.

Bearing stiffness is also an important contributor to the transmission of vibration from the rotating shaft to the housing. Some of the early models describe bearings as ideal boundary conditions for the shafts, or as purely translational stiffness elements. These models cannot predict flexural or out-of-plane type motion on the flexible housing. This issue is addressed by the analytical model proposed by Lim and Singh [12]. Later, various researchers further refined the models to include the time-varying and speed-varying characteristics [13-14] and extended it to double row ball bearings [15]. However, these models require the displacement or force vector as the input and thus a system model is needed to determine the accurate bearing stiffness in the gearbox. The bearing stiffness in a statically overdetermined gearbox is determined using an iterative process by Čermelj and Boltežar [16]. Its influence on the gearbox vibration is investigated by Razpotnik, et.al. [17]. Guo and Parker proposed a finite element/contact mechanics (FE/CM) model of ball bearing and used finite difference method to solve for the bearing stiffness [18]. They have also applied this modal to predict the vibration response and sound radiation of a parallel axis gearbox [19].

In this paper, the influence of elastic shaft-bearing-housing structure on the transmission error is firstly investigated by comparing the gear pair and axle system model. The axle system model includes detailed contact modeling between gear pairs, the roller bearings and housing. The bearing stiffness matrix are then calculated using the FE/CM model and finite difference method. Finally, the modal frequencies and mode shapes of the geared rotor system model, as well as the dynamic responses are simulated and compared between the model with simply-supported bearing assumption and the model with flexible bearings.

CONTACT ANALYSIS

The hypoid gear mesh model, including mesh point, line-of-action, mesh stiffness and kinematic TE can be derived from the load distribution results calculated by a 3-dimensional quasi-static loaded tooth contact analysis program for hypoid and spiral bevel gears [20]. This program combines the semi-analytical theory with finite element method [21], which can solve the gear tooth contact problem very efficiently.

Gear Pair Model

A hypoid gear pair with multiple contact interfaces is shown in Figure 1(a), and Figure 1(b) shows contact cells on each tooth pair. The position vector of the contact cell in the mesh coordinate system is $r_i(r_{ix}, r_{iy}, r_{iz})$, the contact force is f_i , and the normal vector is $n_i(n_{ix}, n_{iy}, n_{iz})$.

The total contact force is calculated by summing the contact forces on each contact cell (assuming there are N contact cells)

$$F_j = \sum_{i=1}^N n_{ij} f_i, \quad F = \sqrt{\sum_{j=x}^z F_j^2}, \quad (j = x, y, z) \quad (1)$$

The line-of-action vector $L_m(n_x, n_y, n_z)$ can be obtained from

$$n_j = \frac{F_j}{F}, \quad (j = x, y, z) \quad (2)$$

The total contact moment is given by

$$M_x = \sum_{i=1}^N f_i [n_{iz} r_{iy} - n_{iy} r_{iz}] \quad (3a)$$

$$M_y = \sum_{i=1}^N f_i [n_{ix} r_{iz} - n_{iz} r_{ix}] \quad (3b)$$

$$M_z = \sum_{i=1}^N f_i [n_{iy} r_{ix} - n_{ix} r_{iy}] \quad (3c)$$

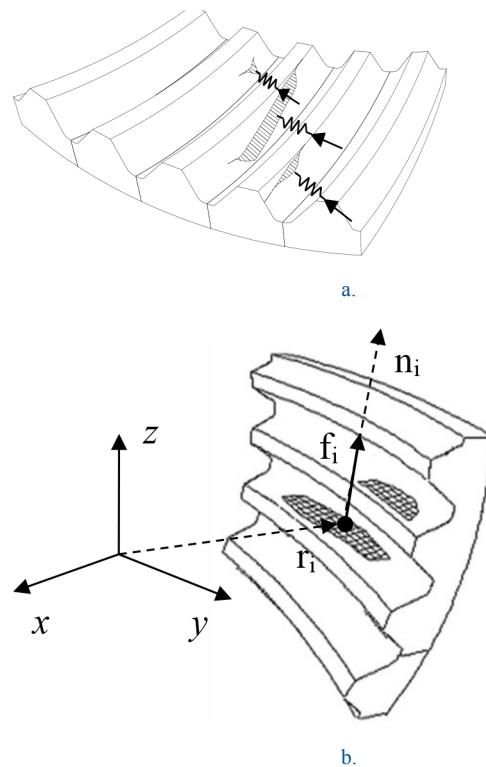


Figure 1. Illustrations of: (a) multi-point coupling of hypoid gear pair, and (b) contact cells on engaging tooth surface.

The mesh point $R_m(x_m, y_m, z_m)$ can be obtained from



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$$y_m = \frac{\sum_{i=1}^N r_{iy} f_i}{\sum_{i=1}^N f_i} \quad (4a)$$

$$x_m = (M_z + F_x y_m) / F_y \quad (4b)$$

$$z_m = (M_y + F_z x_m) / F_x \quad (4c)$$

The translational loaded and unloaded transmission errors e_L and ε_0 are the projections of corresponding angular transmission error along the line of action. Finally, the mesh stiffness is defined by

$$k_m = F / (e_L - \varepsilon_0) \quad (5)$$

The above representations are then combined with the geared rotor system dynamic model to enable TE excitation to be applied reasonably well and also to compute meaningful gear mesh response.

Axle System Model

Figure 2 illustrates the vehicle rear axle system considered. The model includes pinion shaft, ring gear with the carrier, four tapered roller bearings and axle housing. For simplification purpose the bevel gears in the differential are neglected.

The FE/CM model takes account of the macro- and micro-geometry of the gear tooth profile, bearing rollers and raceway. Its efficiency and accuracy has been proven in the previous studies [19, 22]. The advantage of the FE/CM model is that it considers the detailed geometry of components and provides contact information which is not available in an analytical model, while it is more efficient than the full FE model. The output of the FE/CM model is the gear transmission error, tooth load distribution, bearing deflections and loads. Based on these output results, the mesh parameters and bearing stiffness can be calculated.

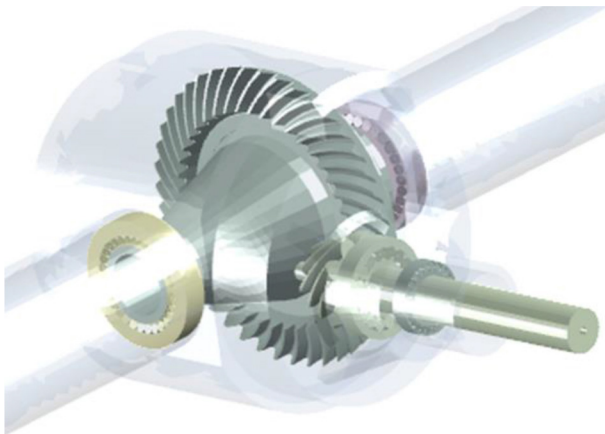


Figure 2. Axle system model.

The bearing model includes the important design details which are absent in most of the previous analytic models, such as the roller and raceway crowning, clearance, roller length, bearing width. The discrepancy due to neglecting these details has been shown in previous study.

The deflections of the bearings calculated from quasi-static analysis of the axle system are used for the determination of bearing stiffness [18, 23]. Small perturbation is added on the bearing deflection vector for each direction at a time. The forces due to the perturbed deflection vector are calculated. The bearing stiffness matrix is calculated based on the forces and displacements

$$K = \begin{bmatrix} \frac{\partial F_x}{\partial x} & \frac{\partial F_x}{\partial y} & \frac{\partial F_x}{\partial z} & \frac{\partial F_x}{\partial \theta_x} & \frac{\partial F_x}{\partial \theta_y} & 0 \\ \frac{\partial F_y}{\partial x} & \frac{\partial F_y}{\partial y} & \frac{\partial F_y}{\partial z} & \frac{\partial F_y}{\partial \theta_x} & \frac{\partial F_y}{\partial \theta_y} & 0 \\ \frac{\partial F_z}{\partial x} & \frac{\partial F_z}{\partial y} & \frac{\partial F_z}{\partial z} & \frac{\partial F_z}{\partial \theta_x} & \frac{\partial F_z}{\partial \theta_y} & 0 \\ \frac{\partial M_{\theta_x}}{\partial x} & \frac{\partial M_{\theta_x}}{\partial y} & \frac{\partial M_{\theta_x}}{\partial z} & \frac{\partial M_{\theta_x}}{\partial \theta_x} & \frac{\partial M_{\theta_x}}{\partial \theta_y} & 0 \\ \frac{\partial M_{\theta_y}}{\partial x} & \frac{\partial M_{\theta_y}}{\partial y} & \frac{\partial M_{\theta_y}}{\partial z} & \frac{\partial M_{\theta_y}}{\partial \theta_x} & \frac{\partial M_{\theta_y}}{\partial \theta_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (6)$$

where x and y denote the bearing in-plane axes and z denotes the axial direction. The symbols θ_x, θ_y are the out-of-plane angular deflections about the x and y axes, respectively. The stiffness matrix is calculated numerically through second order finite difference method by

$$\frac{F(q_0 + \delta q) - F(q_0 - \delta q)}{2\delta q} \quad (7)$$

where q_0 is the calculated bearing deflection vector at given load F_0 about which the stiffness matrix is desired and $\delta q = \{\delta q_x, \delta q_y, \delta q_z, \delta q_{\theta_x}, \delta q_{\theta_y}\}$ is the specified small disturbance vector at q_0 .

DYNAMIC ANALYSIS

The 14-DOF lumped parameter model of the hypoid geared rotor system is similar to the model used in Tao's study [24]. The equations of motion of the geared rotor system in matrix form can be written as

$$[M_1]\{\ddot{x}_1\} + [C_1]\{\dot{x}_1\} + [K_1]\{x_1\} + \Omega[G_1]\{\dot{x}_1\} = \{F_1\} \quad (8) \text{ where}$$

$$\{x_1\} = \{x_D, x_p, y_p, z_p, \theta_{px}, \theta_{py}, \theta_{pz}, x_g, y_g, z_g, \theta_{gx}, \theta_{gy}, \theta_{gz}, \theta_L\}^T \quad (9)$$

$$\{M_1\} = \text{diag}\{I_D, M_p, M_p, M_p, I_{px}, I_{py}, I_{pz}, M_g, M_g, M_g, I_{gx}, I_{gy}, I_{gz}, I_L\} \quad (10)$$

The stiffness matrix $[K_1]$ is the lumped shaft-bearing assembly support stiffness. The damping matrix $[C_1]$ is assumed to be viscous type and derived from damping ratio. The gyroscopic matrix $[G_1]$ can be expressed as given below



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