

Elastohydrodynamic Lubrication Damping of Spiral Bevel Gears at Moderate Loads

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ABSTRACT

Modeling of elastohydrodynamic lubrication phenomena for the spiral bevel gears is performed in the present study. The damping and the friction coefficient generated from the lubricated contact area will have profound effects on the dynamics of spiral bevel gears. Thus the damping value generated from this friction model will be time varying. This makes the use of constant and empirical damping value in the dynamics of spiral bevel gears questionable. The input geometric and kinematic data required for the elastohydrodynamic lubrication (EHL) simulations are obtained using Tooth Contact Analysis. A full numerical elastohydrodynamic lubrication simulations are carried out using asymmetric integrated control volume (AICV) algorithm to compute the contact pressures. The fast Fourier transform is used to calculate the elastic deformations on the gear surfaces due to contact load. The computed pressures and the effective viscosity are used to calculate the time varying damping for the spiral bevel gears. Parametric studies are also conducted by varying the rolling velocities, lubricant properties and torque load to explain the effect of lubrication on the spiral bevel gear dynamics response.

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INTRODUCTION

Spiral bevel gears are widely used in the rear axle of automobile power train systems. Lubrication modelling is mainly studied for two major reasons: 1) to study the effect of lubrication on the gear dynamics 2) to study its effect on the durability of the gears. A theoretical model of mixed Elastohydrodynamic lubrication was proposed by Li et al [1] in which an asymmetric integrated control volume was used. In this work the model developed was validated with experiments available in the literature. The Elastohydrodynamic lubrication was applied to hypoid gears using elliptical point contacts with an angled flow of lubricant by Mohammadpour et al [2]. To study the effect of lubrication on the dynamics of spur gears viscous gear mesh damping must be calculated from the elastohydrodynamic lubrication theory. The variation of the damping value with the gear mesh position was investigated and the influences of various contact conditions including the speed, torque, temperature were studied by Li et al [3]. The EHL theory along with the lubricant rheology was used to determine the sliding friction and also to calculate the sliding power loss calculation in parallel axis gears by Wu et al [4]. An empirical model for determining the time varying coefficient of friction was developed by Feng et al [5]. This model was used to determine the dynamic response of right angled geared rotor system with the effect of friction and lubrication on the high speed gear dynamics. Feng et al [5] concluded that the friction and lubrication has more effect on the spiral bevel gears than on the hypoid gears. The pressure measurements made from the elastohydrodynamic lubrication experiments with thermal effects were compared with the simulation results by I. Jubault et al [6]. The numerical algorithm for EHL was developed using multigrid techniques and multilevel multi integration

technique was used for the evaluation of the elastic potential involved in the deformation calculations by Venner et al [7]. The well-known relations for central film thickness and minimum film thickness were given by Hamrock and Dowson for elliptical contacts and fully flooded conditions [8, 9, 10]. This relation helps to predict the central film thickness based on dimensionless operation parameters.

The basic framework for hypoid gear dynamics was given by Cheng and Lim [13]. The model proposed for hypoid gears included a mesh model and a dynamic model. The mesh model for the hypoid gear was further improved by H. Wang [14] to include the multipoint mesh model and also time varying mesh properties. The mesh model developed by H. Wang was based on the quasistatic loaded tooth contact analysis. A nonlinear time varying mesh model was developed by J. Wang [15]. The effect of lubrication on durability of gears includes contact fatigue modelling of gear tooth and studying the stresses developed on the gear tooth. This would enable to predict the fatigue life of the gear tooth and to determine the strain life of the gear using the critical plane method [16]. The nonlinear model proposed by J. Wang had 14 degrees of freedom and it considered backlash and asymmetric mesh stiffness nonlinearity into consideration. Later Peng [17] obtained a multi-body dynamic model coupled with the vibration model by considering large shaft rotations. Peng also investigated the sliding friction effect in both hypoid and bevel gear dynamics. He concluded that frictional excitations were more in the spiral bevel gears than the hypoid gears due to directionality effect of spiral bevel gears which was not observed in the hypoid gear dynamics.

There are certain experiments which do not correlate well with the simulated results, mainly due to the usage of proportional constant damping in the dynamics calculation. The higher values of damping generally reduces the nonlinear behavior and hence overestimation of damping from proportionality damping may mask the actual nonlinear characteristics existing in the simulated results [3]. In the multi-body dynamic models (Peng [17]) where the deformations are included to determine the stiffness and transmission error; correlation with the experiments would be better if accurate damping models were used in the calculations.

Hence the previous lubrication studies conducted on gears studied the effect of lubrication on the durability of parallel axis gears. Those which studied the effect of lubrication on hypoid gears studied its effect on the transmission efficiency. The effect of lubrication on damping was studied for parallel axis gears. The dynamic models that exist of right angled geared system either assumes a constant damping value or a proportional value to the stiffness of the system. This work is performed to fill this gap. In this work damping values are calculated for the right angled geared system based on the elastohydrodynamic lubricant properties and gear mesh properties. This work is not hitherto reported in the literature earlier. The lubrication modeling done on the right angled gears can be used either to calculate the stresses induced in the tooth profiles of meshing gears or damping properties derived from the EHL contacts can be used in the prediction of the dynamic response. The dynamic response predicted then can be used to predict the vibration levels and noise radiation levels of the gear box housing. In any case modeling the lubricated contacts gives more realistic simulation results for right angled geared rotor system.

GEAR MESH MODEL AND LUBRICATION MODEL

Gear Mesh Model

The spiral bevel gear mesh parameters includes mesh point, line of action, mesh stiffness and quasistatic mesh force. A three dimensional quasistatic loaded tooth contact analysis (TCA) was carried out using the tooth contact analysis program for hypoid and spiral bevel gears [11]. This program combines the Boussinesq's surface integral and the finite element method to calculate the displacements.

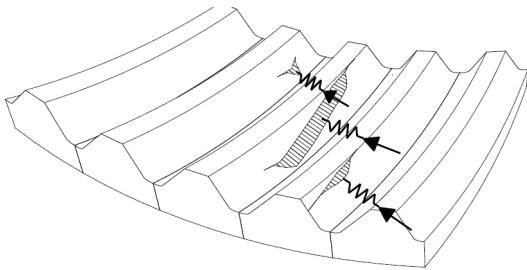


Figure 1. A multipoint mesh model with contact cells of engaging gear tooth

The results of loaded tooth contact analysis consists of the position vector r_i , normal vector n_i , load force f_i of each cell i the loaded transmission error e_L and the kinematic transmission error e_0 for each roll angle. The total contact force is calculated by

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

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

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

The total contact moment for the spiral bevel gears is given by

$$M = \sum_{i=1}^N f_i \cdot (r_i \times n_i) \quad (3)$$

The mesh point (x_m, y_m, z_m) is given by

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

$$X_m = \frac{(M_z + f_x Y_m)}{f_y} \quad (4b)$$

$$Z_m = \frac{(M_y + f_z X_m)}{f_x} \quad (4c)$$

The linear loaded e_l and the unloaded e_0 transmission error is calculated along the line of action vector. The mesh stiffness is calculated by

$$k_m = F_T / (e_l - e_0) \quad (5)$$

The above method is used to calculate the mesh parameters of one interface and it is called as single point mesh model. The elastohydrodynamic lubrication model discussed in this work is based on a single point mesh model.

Elastohydrodynamic Lubrication Model

The lubricant entrainment between the teeth of right angled geared system is given by Reynold's equation [2]

$$\frac{\partial[\frac{\rho h^3}{\eta} \frac{\partial p}{\partial x}]}{\partial x} + \frac{\partial[\frac{\rho h^3}{\eta} \frac{\partial p}{\partial y}]}{\partial y} = 6U \left(\frac{\cos \theta}{\partial x} \frac{\partial[\rho h]}{\partial x} + \frac{\sin \theta}{\partial y} \frac{\partial[\rho h]}{\partial y} \right) \quad (6)$$

where p , h and ρ are pressure, lubricant film thickness and density of lubricant which depends on X and Y . The transient term due to squeeze term in the Reynolds transport equation is not considered because it increases the load carrying capacity of the gears. This equation is solved using finite difference scheme. The changes in the contact pressures are very high which results in a nonlinear change in the viscosity of the lubricant. The accuracy of the