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A numerical study on the performance of straight bevel gears operating under mixed lubrication regime

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ABSTRACT

A model has been developed for predicting the film thickness and friction coefficient under the mixed-lubrication regime in straight bevel gears. Each pair of straight bevel gear teeth is replaced with multiple pairs of spur gear teeth using the Tredgold approximation, and the transmitted load and radii of curvature are accordingly evaluated. The bevel gears' performance is predicted by employing the load-sharing concept with the consideration of elastic, elasto-plastic and plastic deformation for asperities. The effect of parameters such as load, roughness, hardness, and rolling speed on the performance of the gear system is investigated. It has been shown that under the reported operating conditions, increasing the surface roughness to values higher than 0.5 μm results in a shift of lubrication regime from elastohydrodynamic regime to mixed-elastohydrodynamic regime.

1. Introduction

Bevel gears are among the key elements in a power transmitting system with intersecting axes. To improve operational performance and guard against wear and surface fatigue, it is necessary to have a tool for predicting parameters such as the lubricant film thickness and the friction coefficient that realistically take into account the tribological properties of surfaces. The geometrical complexity of bevel gears, however, makes it difficult to obtain the distribution of these contact parameters.

The lubrication regime for gears is in the mixed-elastohydrodynamic lubrication (mixed-EHL), particularly when the speed is relatively low. In these applications, the gear load is shared by a combination of the hydrodynamic action as well as the contact of surface asperities. Gears, like other mechanical elements, have rough surfaces and thus it is important to consider the surface roughness in gear-contact modeling.

Johnson et al. [1] presented the load-sharing concept wherein the total transmitted load is shared between the contacting asperities and the lubricating film. They used the Greenwood–Williamson [2] model to statistically calculate the stress level in an ensemble of asperities. Considering the Johnson's load-sharing concept, Gelinck and Schipper [3] developed a procedure for estimating the asperity contact pressure and reported a successful prediction of the Stribeck curve. In their approach, there is no need for developing a complete solution of the Reynolds equation and the performance parameters are calculated relatively fast. Lu et al. [4] used the model developed by Gelinck and Schipper to obtain the Stribeck curve in conformal contacts. They verified their modeling with experimental results performed on a heavily-loaded journal bearing.

The Greenwood–Williamson model considers the surface asperities deformation to be purely elastic. This limitation was later addressed by Zhao et al. [5] who presented a comprehensive model which included elastic, elastic–plastic and plastic asperity deformations. Faraon [6] studied the influence of operating parameters such as speed and load on the coefficient of friction. He employed the model that was originally developed by Zhao et al. [5] and proposed a mixed-lubrication model capable of predicting the behavior of the Stribeck curve. Beheshti and Khonsari [7] proposed a continuum damage mechanics model to predict the scaling

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factors in the mixed-lubricated contacts. Akbarzadeh and Khonsari [8] presented a model to predict the performance of the spur gears considering the surface roughness. For each point along the line of action, they replaced the contact of gears teeth with contact of two rollers, and calculated the friction coefficient of spur gears under the mixed-lubrication regime. Masjedi and Khonsari [9] developed an equation to find the central film thickness and minimum film thickness in line contact model based on the simultaneous solution to the modified Reynolds equation and surface deformation.

In this study, the contact of bevel gear's teeth under the mixed-lubrication regime is modeled using the Tredgold approximation. Following the procedure developed by Akbarzadeh and Khonsari [8] analysis, contact of the bevel gear teeth is replaced with a set of rollers at each point. Film thickness, friction coefficient and Hertzian pressure at each point along the line of action (LoA) and each layer along face width are calculated. A parametric analysis of the key factors that influence the gear performance is presented.

2. Model

According to the Tredgold approximation [10], the contact of bevel gear teeth can be replaced with a pair of spur gears in which their center line will be laid on the axes of bevel gears as shown in Fig. 1. The number of teeth of virtual spur gears and their equal diameter can be found from the following equations [11]:

$$z_{spur} = \frac{z_{bevel}}{\cos\gamma} \tag{1}$$

$$d_{spur} = \frac{d_{bevel}}{\cos\gamma} \tag{2}$$

where:

z_{spur}	Number of teeth in virtual spur gear
z_{bevel}	Number of teeth in bevel gear
d_{bevel}	Pitch diameter of bevel gear
d_{spur}	Pitch diameter of virtual spur gear
γ	Pitch cone angle.

In this study, for higher accuracy, the face width of each tooth of bevel gear is replaced with multiple pairs of spur gears as shown in Fig. 1. In other words, each layer along the face width is replaced with a pair of spur gears. The load distribution is not uniform along the face width and the load changes linearly so that the maximum load is applied on the heel and the minimum at the toe. Moreover, the number of contacting teeth will affect the load distribution for bevel gears. After replacing the bevel gear with multiple spur gears, the load acting on each layer of bevel gear is applied on its equivalent spur gear. Then, the contact of gears pair is replaced with their equivalent rollers. Thus, it is necessary to determine the equivalent radii of curvatures of bevel gear at each point along the LoA. Considering the equivalent geometry, the variation of radii of curvature as illustrated in Fig. 2 can be calculated by Eqs. 3–5 [12]:

$$\frac{2}{E_{eq}} = \frac{1-\upsilon_1{}^2}{E_1} + \frac{1-\upsilon_2{}^2}{E_2} \tag{3}$$

$$\frac{1}{R_{eq}} = \frac{1}{r_1} + \frac{1}{r_2} \tag{4}$$

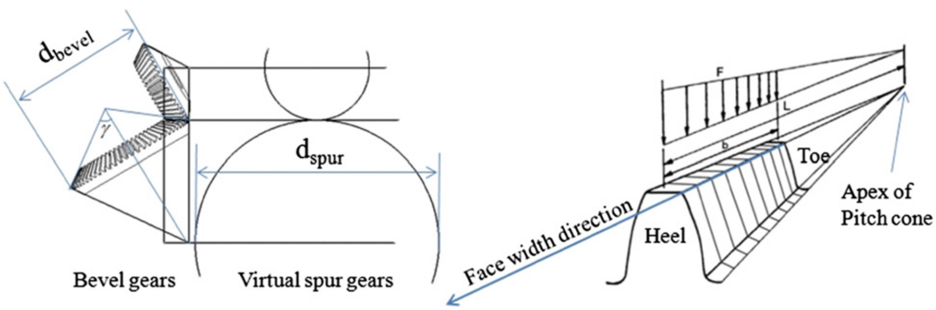

Fig. 1. Replacing the layers of bevel gear faces width with virtual spur gears.