Investigation on Helical Gears under Mixed-Lubrication Regime

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INTRODUCTION
The investigation of wear and related processes is one of the most important issues in the industry [1-7]. Gear is one of the components, which is affected by the wear. Helical gears between all types of gears, has some features cause to use more than other types of gears. These features are high capacity in power transmission and lower noise during its performance. The present study shows a model of contact of teeth of helical gears. The variable parameters in this investigation are surface roughness and thermal effects [7, 8]. For modeling a helical gear, it is assumed each helical gear consists of some narrow spur gears. Theoretical results based on load sharing concept are compared to experimental results. Results illustrate experimental and theoretical results are in agreement.

Results and discussions

Among all types of gears, helical gears are more common due to their high capacity in power transmission as well as lower level of noise. The aim of this study is to present a model for analyzing the contact of teeth of helical gears considering surface roughness. Helical gear is similar to simple gear in terms of outward but there are significant differences in their analysis. On the helical gear to determine the position of the contact point, two non-dimension numbers of ξ and ξ_0 are defined. The physical concept of the first parameter is the ratio of curvature radius to the base circle at the contact point and the second parameter stands for the relative contact location of pinion and gear [9]. Using Eq. (1) the first parameter is obtained:

$$\xi = \frac{z}{2\pi}\sqrt{\frac{r_c^2}{r_b^2} - 1} \qquad (1)$$

where $\boldsymbol{Z}$ is the number of teeth, $\boldsymbol{r_c}$ is the distance of the center of the gear to the contact point and $\boldsymbol{r_b}$ represents the base circle radius which is shown in Fig. 1. Therefore, the radius of the curvature of the contact point can be obtained by Eq. (2):

$$\rho = \sqrt{r_c^2 - r_b^2} = r_b \frac{2\pi}{Z}\xi \qquad (2)$$

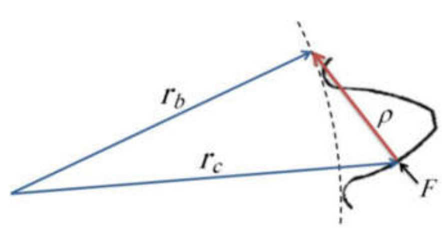

Figure 1. Position of contact point on involute

In this investigation, a model based on load sharing is provided to analyze the lubrication of the helical gear. The advantage of using this method are lower calculating time as well as investigating the effect of temperature and surface roughness [10-13].In the present model, the helical gear will be divided into several simple gears and then each contact point of struggling gear is substituted by an equivalent cylinder. It is worthwhile to point out analysis of lubrication on these equivalent cylinders is performed based on narrow Hertzian assumption.

The lubrication analysis is carried out according to the load-sharing [14-17] which the lubricant film thickness is determined.
In this model, the teeth of gear are assumed to be rigid and as a result, the distribution of loads along the contact line is going to be uniform.

$$I_v(\zeta_0) = \frac{1}{b_0} \sum_{i=0}^{E_\gamma} (\sin b_0 [\zeta_{i,\sup} - \frac{\varepsilon_\alpha}{2}] - \sin b_0 [\zeta_{i,\inf} - \frac{\varepsilon_\alpha}{2}] \tag{3}$$

$$b_0 = [\frac{1}{2}(1 + \frac{\varepsilon_\alpha}{2})^2 - 1]^{-1/2} \tag{4}$$

$$\zeta_{i,\sup} = \zeta_0 + i + \varepsilon_\alpha - \min(\zeta_0 + i, 0) - \max(\zeta_0 + i, \varepsilon_\alpha) \tag{5}$$

$$\zeta_{i,\inf} = \zeta_0 + i - \varepsilon_\beta + \varepsilon_\alpha - \min(\zeta_0 + i - \varepsilon_\beta, 0) - \max(\zeta_0 + i - \varepsilon_\beta, \varepsilon_\alpha) \tag{6}$$

Where:

$$\varepsilon_\beta + \varepsilon_\alpha = E_\gamma \tag{7}$$

Also, length changes of the contact line of a tooth along its contacting cycle can be obtained as follows:

$$L(\zeta_0) = \frac{I_v \times b}{\varepsilon_\beta \times \cos \psi_b} \tag{8}$$

Then the total load on a single tooth during its contacting cycle is calculated as below:

$$f(\xi_0) = \frac{F}{L(\xi_0)} \tag{9}$$

$$F_T = F \times \frac{l(\xi_0)}{L(\xi_0)} = L(\xi_0) \times l(\xi_0) \tag{10}$$

The total force includes of the forces carried by the lubricant and asperities. Using the scaling factors of $\boldsymbol{\gamma_1}$ and $\boldsymbol{\gamma_2}$, the forces carried by lubricant,$\frac{F_T}{\gamma_1}$, and asperities , $\frac{F_T}{\gamma_2}$, are obtained as follows:

$$F_T = \frac{F_T}{\gamma_1} + \frac{F_T}{\gamma_2} = F_H + F_c \tag{11}$$

$$1 = \frac{1}{\gamma_1} + \frac{2}{\gamma_2} \tag{12}$$

Using the load-sharing method, the friction force also can be divided into two different parts. The first one is related to shear force between lubricant film and the second one associated with the aspirates of contacting surface:

$$F_f = F_{f,H} + F_{f,c} \tag{13}$$

$$F_{f,c} = \sum_{i=1}^{N} f_c \times F_{Ci} = f_c \sum_{i=1}^{N} F_{Ci} = f_c \times F_C \tag{14}$$

The final friction coefficient is also calculated by the equation below:

$$F_{f,H} = \frac{u_{sliding \times \eta}}{h_c} \times A_H \tag{15}$$

$$\mu = \frac{F_f}{F_T} = \frac{f_c F_{C+} \frac{u_{sliding \times \eta}}{h_c} \times A_H}{F_T} \tag{16}$$

Where $\boldsymbol{\eta}$ stands for viscosity, $\boldsymbol{A_H}$ is contact area of lubricant film and $\boldsymbol{h_c}$ represents the thickness of the lubricant film.

After determining the required inputs to calculate the thickness of the film lubricant and coefficient of friction, the first step is the initial guess for $\boldsymbol{\gamma_1}$ with several trial and error to optimize them and then using Eqs. (12-16) the friction coefficient will be obtained [17, 18]. Fig. 2 shows the contribution of the hydrodynamic film and the asperities contact in carrying the total load. As the figure illustrate the main load is carried by the lubricant film (approximately 80 percent of the total load is tolerated by the lubricant film) [19].

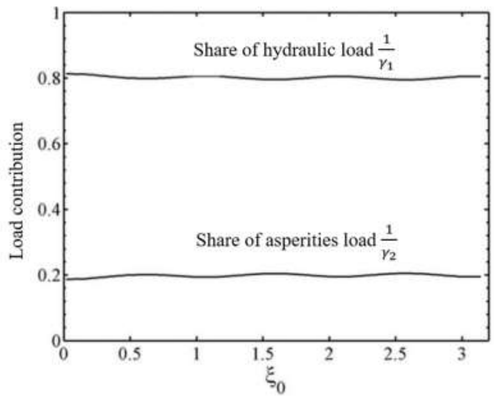

Figure 2. The contribution of the hydrodynamic film and the asperities

Increasing the temperature during gear engagement leads to decrease in lubricant viscosity. Therefore, considering Eq. (15) hydraulic friction coefficient decreases (Fig. 3). Also, Fig. 4 shows the contribution of the hydrodynamic film and the asperities of the friction force of a tooth during a cycle. As the results show, the share of asperities is much more than the share of lubrication [19].

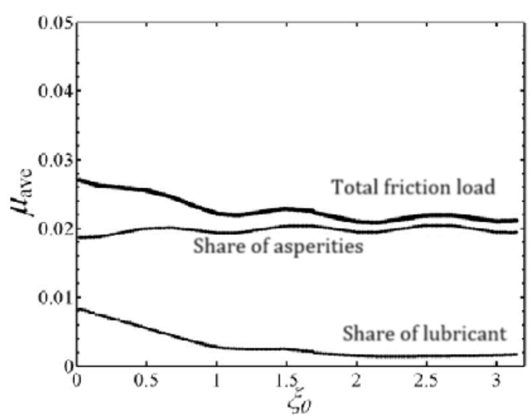

Figure 3. The average friction coefficient a tooth during a cycle

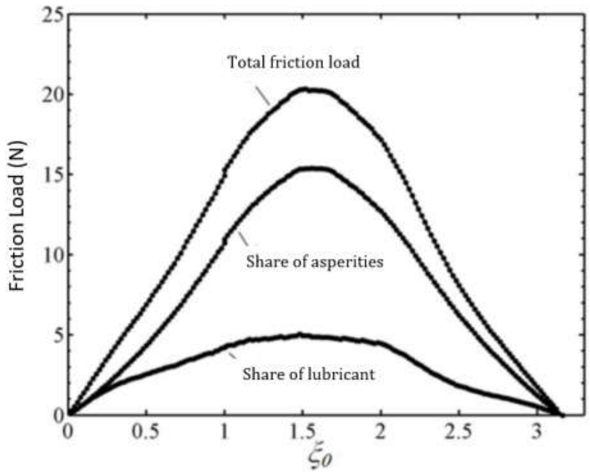

Figure 4. The contribution of the hydrodynamic film and the asperities of the friction force of a teeth during a cycle

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KEYWORDS

helical gears, lubrication, surface roughness, lubrication