Tribology International 94 (2016) 146–154

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

# A study of spur gear pitting under EHL conditions: Theoretical analysis and experiments

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ARTICLE INFO

*Article history:*
Received 6 May 2015
Received in revised form
17 July 2015
Accepted 20 August 2015
Available online 29 August 2015

*Keywords:*
Gear transmission
Elastohydrodynamic lubrication (EHL)
Shear stress
Pitting

ABSTRACT

Based on the elastohydrodynamic lubrication (EHL) theory, the roughness of surface topography, the stress of surface and subsurface under pure rolling contact and sliding contact of spur gear are studied. The formation of micropitting, pitting and their sequence is studied with numerical results based on stress field, especially shear stress, by taking into account the high shear tractions caused by local asperity friction. The morphology of micro-pitting and macro-pitting under different stress levels and the distributions of stress are contrast analyzed. The experiments are carry out to investigate the pitting under EHL conditions by using cylindrical roller test machine. The results show that the shear stress of subsurface is a crucial mechanical factor leading to pitting. And pitting is a gradual fatigue phenomenon under the action of the shear stress.



## 1. Introduction

Gear pitting mainly refers to a fatigue phenomenon observed in gear surface, appearing as the material fatigue spalling to punctate pits. According to statistics, pitting is one of the most common types gear failure [1]. There are many influence factors in micro-pitting, such as surface roughness, slide–rolling ratio and lubricating oil, and so on. While slide–rolling contact is a main form during gear transmission [2,3].

The common method to study the mechanism of gear pitting is by using the slide–rolling contact model based on elastohydrodynamic lubrication (EHL) theory [4]. Hu and Zhu [5] took an universal EHL equation to describe the slide–rolling contact lubrication, mixed lubrication and boundary lubrication conditions. Masjedi et al. [6] and Olver et al. [7] studied the surface roughness condition and the effect of oil film on the load transfer through the EHL equations, which showed that the surface contact fatigue is not only the interaction of metal materials, but more is the effect of oil film on metal. Zhu et al. and Ren et al. [8] forecasted gear pitting fatigue life based on 3-D line contact EHL theory under mixed EHL condition. For one extensibility material, they studied the relationships among the contact fatigue, the equivalent stress of subsurface as well as the Hertz contact stress on the surface. Meshing situations of 15 gear pairs were analyzed and the prediction results were compared with results of theory. Li and Kahraman [9–12] carried out a vast amount of research on a failure pitting model. And they established a pitting model on the fatigue life under transient mixed EHL conditions after considering profiles modification and manufacturing error.

Equivalent cylindrical rollers are widely used to study pitting. Ahlroos [13] used cylindrical roller to simulate the contact fatigue process, and analyzed the influence of the surface roughness, the oil film type and the surface treatment on pitting, he pointed out that the surface roughness plays a decisive role to the pitting. Oila [14] studied roller contact fatigue failure phenomenon by using a cylindrical roller test machine. And Oila put forward that the surface pressure is the crucial factor for the formation of initial crack and the propagation of the crack is mainly affected by slide roll ratio. Aslantas [15] simulated the crack growth and life under the rolling load on the basis of the maximum shear stress theory and fracture mechanics numerical. Hohn [16] studied to the effect of lubricating oil temperature on the gear surface fatigue failure in detail, and discussed the interaction mechanism between the lubricating oil and the surface material. Laine [17] studied the relationship between the lubricating oil film thickness and roughness and its influence to the pitting with three slide-roller contact tester, he also found that tooth wear can delay the formation of pitting.

Pitting can be divided into micro-pitting and macro-pitting. Micro-pitting cannot be distinguished by naked eyes. Those pitting can be distinguished by naked eyes are macro-pitting. The formation mechanism of pitting has no unified conclusion till now. Therefore, it is very necessary and meaningful to study the pitting

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http://dx.doi.org/10.1016/j.triboint.2015.08.037
0301-679X/

of gear contact. In order to predict and analyze the pitting, then to take corresponding measures to prevent and slow down the occurrence of pitting, the theories analysis and experimental investigation on the spur gear pitting under EHL conditions are studied in this work. This paper is organized as follows. In Section 2, the contact process of spur cylindrical gear is depicted, the synthetical curvature radius, average velocity and unit load along contact length of a pair of meshing gear are presented. In Section 3, based on the EHL theory, the roughness of teeth surface topography, the stress of surface and subsurface under pure rolling contact and sliding contact of 40Cr spur gear when $R_a$=0.4 μm are respectively given. In Section 4, equivalent cylindrical rollers are used to study pitting on the surface of 40Cr spur gear, the morphology of pitting is proposed by experimental investigation. Some main conclusions derived from this study are presented in Section 5.

## 2. Contact process of spur gear

The synthetic curvature radius, linear velocity and meshing force will be changed constantly during transmission. Specially, the meshing force is seriously affected by deformations, dynamic effects and transmission errors. In this paper, the pinion and wheel are assumed as rigid body and the transmission errors are ignored. Two finite long equivalent cylindrical rollers are used to simulate the transient slide–rolling contact [16,17], as shown in Fig. 1.

The geometry parameters of meshing points can be calculated by Eq. (1) [18].

$$\begin{cases} g_{yc} = \mp \dfrac{1}{2}d_1 \sin\alpha \pm \sqrt{\left(\dfrac{1}{2}d_1 \sin\alpha\right)^2 - \left(\dfrac{1}{2}d_1\right)^2 + r_c{}^2} \\ R_1 = \dfrac{1}{2}d_1 \sin\alpha \pm g_{yc} \\ R_2 = \dfrac{1}{2}d_2 \sin\alpha \mp g_{yc} \\ \dfrac{1}{R} = \dfrac{1}{R_1} \pm \dfrac{1}{R_2} \\ u_{1,2} = \omega_{1,2} \times R_{1,2} \\ u_s = (u_1 + u_2)/2 \\ s = (u_1 - u_2)/u_s \end{cases} \tag{1}$$

where $d_1$ denotes the pitch circle diameter of the driving gear, $d_2$ denotes the pitch circle diameter of the driven gear, and $\alpha$ denotes the pressure angle at the pitch circle. $g_{yc}$ denotes the distance between meshing point and pitch point in meshing line, $r_c$ denotes the radius of meshing point in driving gear. $R_1$ and $R_2$ are respectively curvature radius of driving and driven gear. $R$ denotes the synthetic curvature radius. $u_1$ and $u_2$ are respectively the rolling speed of driving and driven gear at the meshing point. $\omega_1$ and $\omega_2$ are respectively the angular velocity of driving and driven gear. $u_s$ and $s$ are respectively the averaged velocity and slide–rolling ratio at the meshing point. The upper symbol is applied to the meshing point at the addendum circle of driving gear or dedendum circle of driven gear. The lower symbol is applied to the meshing point at the dedendum circle of driving gear or addendum circle of driven gear.

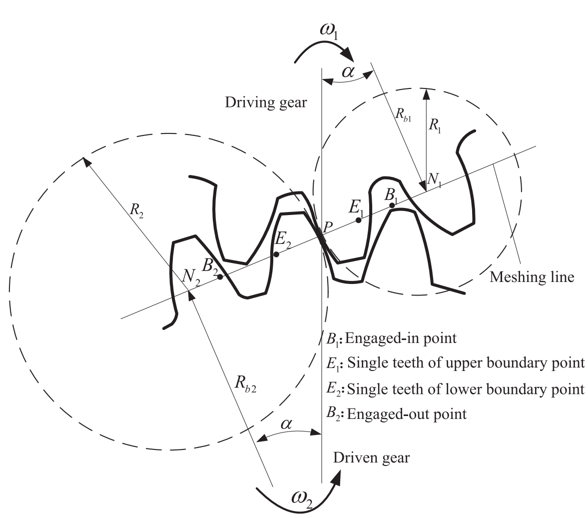


**Fig.1.** Contact process of spur gear.

The meshing force $F_n$ and the equivalent Hertzian contact stress $\sigma_{zeq}$ can be expressed as [19]:

$$\begin{cases} F_n = \dfrac{T_2 z_1/z_2}{nLr_c \cos\alpha} \\ \sigma_{zeq} = \left(\dfrac{EF_n}{2\pi R}\right)^{0.5} \end{cases} \tag{2}$$

In Eq. (2), $F_n$ denotes the meshing force per unit length. $n$ denotes the total number of teeth involved in engagement. $\sigma_{zeq}$ is the equivalent Hertzian contact stress. $E$ is the equivalent elastic modulus, $R$ denotes the synthetic curvature radius, $L$ denotes the contact length.

The parameters of spur involute gear transmission are as follows: gear material is 40Cr, thermal refining and the hardness of surface are HRC25–27, elasticity modulus and Poisson's ratio are respectively $E_1$=$E_2$= 2.06 × $10^{11}$ $P_a$ and $\nu_1$=$\nu_2$=0.3, gear modulus is $m_n$=5 mm, teeth number of the driving and driven wheel are $z_1$ = 25, $z_2$ = 37, pressure angle is 20°, effective tooth width is 50 mm, contact ratio is 1.655, rotation speed of driving gear is $n_1$=1500 r/min, output torque is $T_2$=2.2 × $10^3$ Nm. The distributions of curvature radius and rolling velocity at the meshing point are shown in Fig. 2 and Fig. 3, the synthetic curvature radius $R$ of the meshing points along the meshing line increased first to near pitch point, then decreased. The velocity of the driving gear is gradually increasing whereas the driven gear is decreasing and the averaged velocity is gradually increasing. Slide–rolling ratio $s$ is a negative value from dedendum circle to pitch circle, while it is a positive value from pitch circle to addendum circle. The slide–rolling ratio is zero at the pitch circle. The meshing force is shown in Fig.4. From Fig. 4, there is a larger sudden changing of meshing force from single-tooth meshing regions to double teeth meshing regions, the meshing force of single tooth meshing regions is almost twice as that of the double teeth meshing regions.

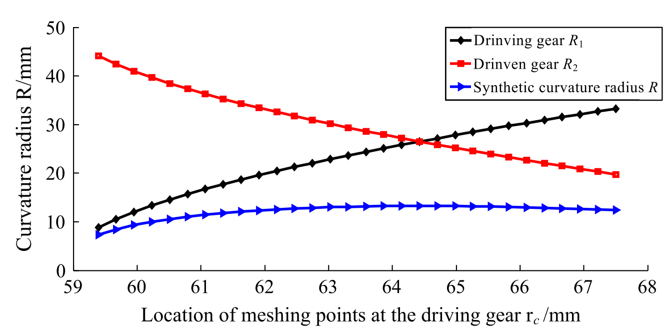


**Fig.2.** Curvature radius changes along the meshing line.