Tribology International 50 (2012) 82–90

Contents lists available at SciVerse ScienceDirect

Tribology International

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

Effect of lubricant selection on EHL performance of involute spur gears

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ARTICLE INFO

Article history:
Received 14 December 2011
Received in revised form
26 January 2012
Accepted 14 February 2012
Available online 24 February 2012

Keywords:
EHL
Rheology
Spur gears
Scuffing

ABSTRACT

As the gear oils usually undergo shear-thinning even in the inlet zone, an accurate EHL analysis requires realistic rheological models. This is necessary for gear oil selection so as to prevent scuffing failure. This paper demonstrates the effect of rheology on the EHL characteristics of spur gears using full transient thermal EHL simulations with Carreau shear-thinning model and Doolittle's free volume based pressure-viscosity relationship. The PDMS oil considered here is found to exhibit severe film thinning with 74% thinner EHL film as compared to a moderately shear-thinning PAO oil which, on the other hand, undergoes a larger thermal reduction.

1. Introduction

In the process of selecting an appropriate gear oil for a given set of operating conditions or determining the limits of safe operation for different gear oils, the lubricant blends developed by lubricant chemists are tested under varying conditions of load and/or speed. One of such tests, known as FZG test, is performed on a gear testing machine to evaluate the capability of the lubricants to avoid/delay scuffing failure and rate different lubricants for the resistance to scuffing. A lubricant which passes a higher load stage is given a higher rating and a comparison of such test results pertaining to different blends suggests the best of all test oils. Since these tests are quite expensive and destructive in nature, it is desirable to replace them by mathematical modeling and simulation so as to minimize time, money and energy.

In this regard, a few attempts have been made in the past to model the transient behavior of gear-teeth lubrication. The involute spur gear is by far the most important and simplest of all the gear systems. However, even the simplest gear system involves time-varying load, geometry, entrainment velocity and slide to roll ratio. Dowson and Higginson [1] were the first to apply their minimum film thickness formula to the gear tooth lubrication problem. A more successful attempt involving transient effects in EHL analysis was made by Wang and Cheng [2,3] who developed a numerical scheme using Grubin-type inlet zone analysis. They were able to predict the minimum film thickness at several points along the line of action and the bulk surface temperatures. An isothermal Newtonian full transient solution was presented by Hua and Khonsari [4] to investigate the effects of geometry on the lubrication performance of involute spur gears without the consideration of dynamic load. In a similar analysis with non-Newtonian circular fluid model incorporating limiting shear strength, Larsson [5] simulated film thickness and pressures along the tooth profile using the multigrid technique described by Venner [6]. Wang et al. [7] incorporated thermal effect in gear tooth lubrication assuming Newtonian fluid model and computed minimum film thickness, friction and temperature along the line of action. A numerical solution to the full transient EHL problem pertaining to spur gears was presented by Kumar et al. [8] for couple stress fluids. Wang and Yi [9] incorporated Ree-Eyring fluid model to study the non-Newtonian transient thermal EHL behavior of gears using multigrid technique. More recently, Li and Kahraman [10] presented transient non- Newtonian mixed-EHL model and showed the differences between the discrete and transient EHL analyses for involute spur gear contact with smooth surfaces and different tooth profile modifications. Subsequently, Li and Kahraman [11] in their study investigated the behavior of high speed spur gear contacts under dynamic conditions. They developed a non-linear time varying vibratory model to calculate the instantaneous tooth forces under dynamic conditions to demonstrate the influence of dynamic loading on gear lubrication.

Despite the aforesaid efforts to model the transient EHL problem, it is required to apply more realistic approaches employing accurate non-Newtonian and piezo-viscous models with due consideration to thermal effect. It has now been clearly established that the rheological model such as Carreau viscosity model successfully characterizes the experimentally observed flow behavior of shear thinning lubricants [12]. Likewise, the free volume based Doolittle's piezo-viscous relation describes the realistic lubricant behavior quite accurately as compared to those used conventionally in EHL investigations [13,14]. Therefore, in the present analysis, an attempt has been made to study the effect of lubricant rheology on the

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doi:10.1016/j.triboint.2012.02.006

Nomenclature

b	half width of Hertzian contact zone, $b=4R\sqrt{W/2\pi}$(m)
b_p	half width of Hertzian contact zone at pitch point (m)
c_p	specific heat of the lubricant (J/kg K)
c_1, c_2	specific heat of the disks (J/kg K)
E'	effective elastic modulus (Pa)
G	gear ratio, $G=z_1/z_2$
h	film thickness (m)
H	dimensionless film thickness, $H=hR/b^2$
$H_{\min}$	dimensionless minimum film thickness, $H_{\min}=h_{\min}R/b^2$
H_o	dimensionless offset film thickness, $H_o=h_oR/b^2$
k	thermal conductivity of the lubricant (W/m K)
k_1, k_2	thermal conductivity of the disks (W/m K)
n	power law index
p	pressure (Pa)
p_h	maximum Hertzian pressure, $p_h=E'b/4R$, (Pa)
P	dimensionless pressure, $P=p/p_h$
r_{b1}, r_{b2}	base circle radii of pinion and gear (m)
r_1, r_2	pitch circle radii of pinion and gear (m)
R	equivalent radius of contact (m)
R_1, R_2	radii of curvature of pinion and gear tooth profiles (m)
S	slide to roll ratio, $S=(u_2-u_1)/u_o$
s	position along the line of action with respect to pitch point (m)
t	time (second)
T	non-dimensional time, $T=t\,u_p/b_p$
u_o	average rolling speed (m/s)
u_p	average rolling speed at pitch point (m/s)
U	dimensionless speed parameter, $U=\eta_o u_o/E'R$
w	applied load per unit length (N/m)
W	dimensionless load parameter, $W=w/E'R$
x	abscissa along rolling direction (m)
X	non-dimensional abscissa, $X=x/b$

Greek symbols

ϕ	pressure angle
β	thermal expansivity of the lubricant (K^{-1})
ρ_o	inlet density of the lubricant (kg/m^3)
ρ_1, ρ_2	densities of the lower and upper surfaces (kg/m^3)
ρ	lubricant density at the local pressure and temperature (kg/m^3)
$\overline{\rho}$	dimensionless fluid density, $\overline{\rho}=\rho/\rho_o$
θ_o	ambient temperature (K)
λ	ratio of characteristic times at contact and pitch points, $\lambda=(b/u_o)/(b_p/u_p)$
θ	temperature of the lubricant (K)
$\overline{\theta}$	dimensionless fluid temperature, $\overline{\theta}=\theta/\theta_o$
θ_{S1S2}	surface temperatures (K)
μ_o	inlet viscosity of the Newtonian fluid (Pa s)
η	apparent viscosity (Pa s)
$\overline{\eta}$	dimensionless apparent viscosity, $\overline{\eta}=\eta/\mu_o$
α	pressure-viscosity coefficient, $\alpha=(d\ln(\mu/\mu_o)/dp)_{p=0}$ (GPa^{-1})

performance of EHL lubricants in spur gears using the aforesaid models in full transient thermal EHL simulations.

2. Scuffing failure

When gears are subjected to heavy loads, the lubricating film may not separate the surfaces adequately leading to localized damage of the tooth surface. This type of failure is known as scuffing and it is characterized by solid phase welding followed by tearing with a consequent roughening of the affected surface [15]. Scuffing can occur at any stage during the lifetime of a set of gears and it may lead to failure in a matter of few hours. While the exact mechanism of scuffing is not fully understood, it is usually associated with the disruption of oil film under high loads. The scuffing failure may occur at very early stages of operation due to any of the following three reasons: (i) inappropriate design, (ii) improper lubricant selection, (iii) poor surface finish. The risk of scuffing damage depends mainly upon the sliding velocity, tooth surface roughness and the lubricant properties. A proper and careful adjustment of any or all of these factors can help avoid/delay scuffing. The chemically active extreme pressure (EP) oil additives are also known to improve the scuffing resistance of steel gears considerably. Another practical measure usually taken is to carry out special surface hardening treatments including nitriding, super finishing and coating with super hard materials such as diamond like carbon (DLC). Super finishing alone was found to raise the minimum scuffing load significantly [16]. Also, the triple combination of nitriding, super-finishing and hard coating has found to give promising results [17]. However; recent investigations have shown that hard coatings are less effective in improving the durability and friction characteristics of polished surfaces [18]. Moreover, the use of EP additives in some particular situations may lead to other forms of failure such as micro-pitting [19]. These additives cannot be used in gas turbine and automatic transmission applications due to incompatibilities with other necessary requirements for the lubricant. In the light of these facts, it becomes quite evident that the careful selection of lubricating oil is the most effective method of preventing scuffing. Generally, low viscosity lubricants have a higher risk of scuffing as the probability of oil film breakdown is quite high. Besides viscosity, the high pressure rheological properties of gear oils greatly affect the oil film thickness and traction coefficient in the elastohydrodynamically lubricated gear teeth contacts. Therefore, the present paper demonstrates the importance of lubricant selection and hence, lubricant rheology on the lubrication characteristics of spur gears.

3. Gear contact geometry, entrainment velocity and load

Gear contact involves a number of time varying parameters which makes the analysis of gear lubrication very difficult. It requires the knowledge of basic kinematics and dynamics pertaining to involute profile spur gears in order to determine the various input parameters for transient EHL simulations. The equivalent radii of curvature of the contacting tooth surfaces within the EHL conjunctions are governed not only by the contact position with respect to the pitch point but the relative distance from the contact point as well. The radii of curvature of the two involute profiles are given as

$$R_1(x,s)=r_1\sin\phi+s-\frac{r_{b1}}{r_1\sin\phi+s}x \tag{1}$$

$$R_2(x,s)=r_2\sin\phi-s+\frac{r_{b2}}{r_2\sin\phi-s}x \tag{2}$$