Predictions of Friction and Flash Temperature in Marine Gears Based on a 3D Line Contact Mixed EHL Model

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1. Summary

In this paper, a numerical solution procedure is developed for the predictions of transient friction and flash temperature in the marine timing gears during one meshing circle based on the 3D line contact mixed lubrication simulation. The obtained results indicate that the polished gear surfaces yield the smallest friction and the lowest interfacial temperature. In addition, It is observed that the conditions of heavy load and low rotational velocity usually lead to significantly increased friction and temperature. In the meantime, by optimizing the gear design parameters, the performance of interfacial friction and temperature can be significantly improved.

2. Keywords

gears, 3D line contact mixed EHL, surface roughness effect, friction, flash temperature

3. Model

3.1 Transient-load sharing

In one meshing cycle of the spur gear, an alternating form of double-teeth loading and single-tooth loading is shown in Fig.1, and the following standard load sharing equation is shown as,

$$\frac{w(t)}{w_0} = \begin{cases} \dfrac{1}{3} + \dfrac{1}{3}\dfrac{t-t_a}{t_b-t_a} & (t_a \le t \le t_b) \\ 1.0 & \\ \dfrac{2}{3} - \dfrac{1}{3}\dfrac{t-t_c}{t_d-t_c} & (t_c \le t \le t_d) \end{cases} \tag{1}$$

3.2 Mixed lubrication model

The unified Reynolds equation[1] of considering the transient velocity change of the spur gear in mixed lubrication region is employed as[2],

$$\frac{\partial}{\partial x}\left(\frac{\rho}{12\eta}h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho}{12\eta}h^3\frac{\partial p}{\partial y}\right) = \frac{\partial(\rho U h)}{\partial x} + \frac{\partial(\rho h)}{\partial t} \tag{2}$$

3.3 machined surface roughness

The fluid film thickness, which involves the effects of changing gear curvature and real machined surface roughness, and expressed as,

$$h(t) = h_0(t) + \frac{x^2}{2R(t)} + v(x,y,t) + \delta_1(x,y,t) + \delta_2(x,y,t) \tag{3}$$

3.4 Friction-temperature

The rheological model of Bair-Winer[3] is employed to predict the friction coefficient in the lubrication region, which is written as,

$$\begin{cases} \dot{\gamma} = \dfrac{\dot{\tau}}{G_\infty} - \dfrac{\tau_L}{\eta} ln\left(1-\dfrac{\tau}{\tau_L}\right)] \\ G_\infty(p,T) = \dfrac{1.2p}{(2.52+0.024T)} - 10^8 \\ \tau_L(p,T) = 0.25G_\infty \end{cases} \tag{4}$$

By the integration of shear stress, the friction force can be obtained. Friction and temperature are interdependent, the flash temperature can be calculated based on moving heat source theory, which is controlled by the second-type Volterra integral equations,

$$T_1(\xi) = T_{b1} + \left(\frac{1}{\pi\rho_1 C_1 U_1 k_1}\right)^{0.5} \times \int_{-x}^{\xi}\left\{\frac{k_f}{h}[T_2(\lambda) - T_1(\lambda)] + \frac{q(\lambda)}{2}\right\}\frac{d\lambda}{\sqrt{\xi-\lambda}} \tag{5}$$

$$T_2(\xi) = T_{b2} + \left(\frac{1}{\pi\rho_2 C_2 U_2 k_2}\right)^{0.5} \times \int_{-x}^{\xi}\left\{\frac{k_f}{h}[T_1(\lambda) - T_2(\lambda)] + \frac{q(\lambda)}{2}\right\}\frac{d\lambda}{\sqrt{\xi-\lambda}} \tag{6}$$

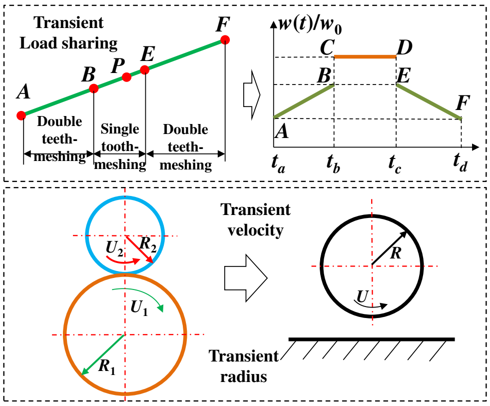

Fig.1 Transient-load sharing, curvature and velocity

4. References

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