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Short communication

Analytical characterization of damping in gear teeth dynamics under hydrodynamic conditions

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ABSTRACT

Using an analytical method, we characterize damping and stiffness in lightly loaded, lubricated gear pairs at different operating speeds and lubricant temperatures. This is accomplished by employing a trace method to approximate and model the hysteresis loop of the lubricant reaction, thus recording the energy transformation mechanism during the gear teeth oscillatory motion. The method can be expanded for use in a variety of problems where hydrodynamic vibro-impacts lead to energy dissipation.

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1. Introduction and governing equations

Lubricant damping properties play an important role in the dynamics of gear pairs. For lightly loaded operating conditions, the critical viscous damping ratio, ζ, can be generally estimated as being inversely proportional to the gear teeth meshing stiffness:

$$\zeta = \frac{c}{2\sqrt{k_m m_e}} \tag{1.1}$$

where c is the viscous lubricant damping coefficient during gear mesh, k_{m} is the average teeth mesh stiffness and m_{e} is the gear pair's equivalent mass. It is understood that the damping ratio implicitly depends on the operating conditions (rotational speed, load and temperature), which influence the rigidity of the lubricant film developed between the teeth [1,7,10].

The aim of this work is to present an analytical method for calculating the variation of the damping ratio in gear teeth contacts under lightly loaded hydrodynamic conditions, reflecting any lubricant effects. To this end, we consider standard involute helical gear teeth geometry without modifications, and we model the gear pair as a lumped parameter torsional system, considering rotational displacements (θ_{p}, θ_{g}) only (Fig. 1). The subscripts p and g refer to the pinion and gear, respectively. The lubricant film can physically act on the drive (h_{d}) and coast side (h_{c}) of the teeth contacts, respectively. Under light loads, it is reasonable to assume

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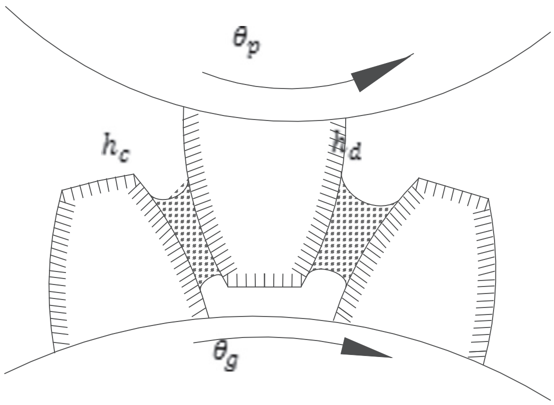

Fig. 1. The physical problem and model used.

that the centres of both gears are not moving axially or laterally (the vibrations of individual shafts in transmissions play no essential role in the system dynamics, [6]) and that the teeth flanks are rigid.

The motion of the pinion can be *a priori* described by

$$\dot{\theta}_p(t) = \dot{\vartheta}_m + \sum_{i=1}^{\infty} \dot{\vartheta}_f^i \sin(i\omega t + \varphi_i) \tag{1.2}$$

In the above equation, $\dot{\vartheta}_m$ is the mean value of the pinion's rotational speed; ω is the excitation frequency (which, in the case of automotive applications for example, depends on the engine type, number of cylinders and crankshaft configuration); $\dot{\vartheta}_f^i$ and φ_i are the fluctuation amplitudes of the rotational speed and the phase of the i^{th} order, respectively.

The corresponding equation of motion for the gear wheel is given in its general form as

$$I_g\ddot{\theta}_g - T_g^w\left(\theta_g, \theta_p\right) + T_g^f\left(\theta_g, \theta_p\right) = -T^p\left(\theta_p, \theta_s\right) \tag{1.3}$$

where I_g is the gear mass moment of inertia and θ_s is the output shaft rotation angle. The torque generated by the lubricant reaction (T_g^w), hydrodynamic flank friction (T_g^f) and hydrodynamic Petrov friction (T^p) have been described in detail in Theodossiades et al. [14]. Only the elements needed for the gear teeth meshing lubricant damping calculation will concern us in this work. Through analytical solution of the Reynolds's equation, the lubricant reaction force (F^w) between the teeth flanks was derived as follows for the iso-viscous rigid prevailing lubrication conditions in the case studied [11,14],

$$F^w = \frac{2L\eta R_{eq} u_e}{h_d h_c}\left[2C_b - (h_c\lambda_d + h_d\lambda_c)\right] \tag{1.4}$$

where L is the gear face width; η is the dynamic viscosity of the lubricant; R_{eq} is the equivalent radius of curvature normal to the line of contact; u_e is the speed of the oil film entraining motion; and $2C_b$ is the total gear backlash, while h_d and h_c correspond to the lubricant film thickness at the drive and coast sides, as shown in Fig. 1. The squeeze-to-roll ratios (λ_d, λ_c) signify the contributions of the squeeze action of the lubricant at the drive and coast sides, respectively, as [11,14],

$$\lambda_j = \begin{cases} \dfrac{3\pi}{2u_e\sqrt{2h_j/R_{eq}}}\dfrac{\partial h_j}{\partial t} & \dfrac{\partial h_j}{\partial t} < 0 \\ 0 & \dfrac{\partial h_j}{\partial t} \geq 0 \end{cases} \quad j = c, d \tag{1.5}$$

The friction introduced by the roller bearings, upon which the gear wheel is mounted on the retaining shaft, can be calculated as [13],

$$F^p = \frac{2\pi\eta \upsilon l R_s}{C_s} \tag{1.6}$$

where R_s is the radius of the output shaft, C_s and l are the clearance and length of the conformal contact surfaces between the shaft and the idle gear, and υ is the entraining velocity of the lubricant.

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