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Inefficiency predictions in a hypoid gear pair through tribodynamics analysis

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ABSTRACT

A tribo-dynamics model, predicting the conjunctional inefficiency and dynamic response of automotive hypoid gear pairs is presented (dynamics coupled with analytical viscous and boundary friction models). The temperature rise at the centre of the conjunction is accounted for using a thermal network model and Time Temperature Superposition (TTS) method. Newtonian and non-Newtonian lubricant shear behaviour are considered with surface topography measurements of a run-in pinion. Inefficiency calculations are performed for typical automotive drive cycle snapshots. Precisely measured lubricant shear characterstics for lubricants with different viscosity modifiers and evolving surface topography are used in the transmission inefficiency study. The integrated thermal-tribodynamic analysis is shown to distinguish between different viscosity modifier types, an approach not hitherto reported in literature.

1. Introduction

The continuing demand for decreasing road vehicle emissions and stringent regulations imposed on both the OEMs as well as their suppliers has strengthened the demand towards improving the overall fuel efficiency of modern automotive transmission systems. By reducing the power losses associated with friction in such systems, it is possible to reduce the energy losses as well as the resulting vehicular emissions. A component of particular interest, with regards to power train friction is the hypoid gear pair, which is part of the differential of the automotive drivetrain system. The significant sliding at the gear teeth contacts, combined with the increased loading can contribute towards frictional power losses [1]. Hence, the development of axle lubricants with improved tribological performance can lead to improved fuel efficiency. To better understand the contribution of the axle lubricant rheology to the conjunction efficiency of the hypoid gear pair, it is necessary to develop an efficient numerical analysis methodology. In addition, although experimental studies focusing on the inefficiency of the rear axles exist in the literature (i.e., [2–5]), the analysis of the problem from a theoretical standpoint can offer a better fundamental insight into the physical phenomena involved.

With regards to the past literature related to the lubrication of hypoid gear pairs, the study by Simon [6] in the early 1980s can be considered as one of the first attempts in addressing the problem. Quasi-static conditions were assumed, while the kinematics and the elasto-statics of the complex teeth conjunction are described through Tooth Contact Analysis TCA. Although the tooth loads considered were low and thus not representative of realistic vehicular operating conditions, this study set the basis of the methodology to be used when examining the lubrication performance of such components. Later, Xu et al. [7,8] and Kolivand et al. [9] predicted the conjunctional efficiency of highly loaded hypoid gear pairs under quasi-static conditions using numerical models. The flank friction was calculated analytically, using a set of empirical/experimental relationships and numerically through solution of line contact Newtonian elastohydrodynamic (EHD) problem. Mohammadpour et al. [10] extended the solution of the EHD problem in hypoid gears by considering an elliptical contact footprint. The effect of angle flow component of the lubricant entraining velocity was also accounted for. However, the lubricant was considered to undergo Newtonian shear. Consequently, the shear thinning action was not observed. Later, Mohammadpour et al. [11] extended their analysis to account for the non-Newtonian response of the axle lubricant. The impact of the dynamic response of the hypoid gear pair on the conjunctional efficiency was examined by Karagiannis et al. [12] following an analytical approach for predicting flank friction. Mohammadpour et al. [13] extended the aforementioned approach to account for the effect of lateral vibrations of the supporting shafts, which are due to the compliance of their supporting bearings.

The results in Ref. [13] have shown the potenial for tribodynamic

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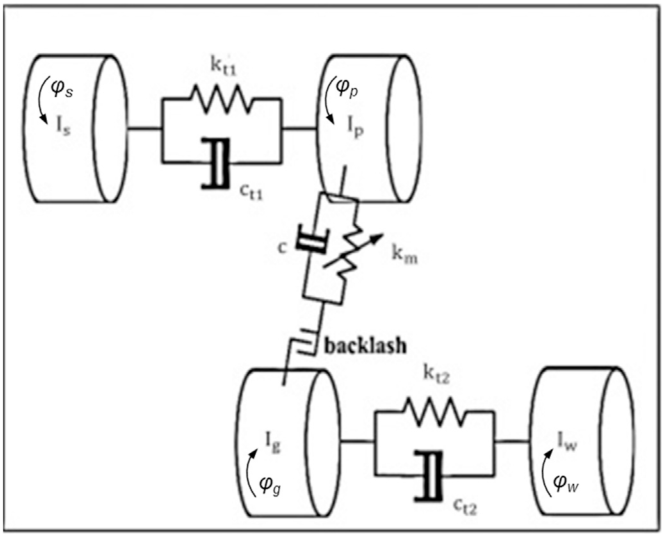

Fig. 1. The 4-DoF torsional gear dynamics model.

analysis to predict the transmission system performance in a realistic emission cycle such as the NEDC. This approach provides a further opportunity for a suitably refined tribodynamic model to ascertain the effect of lubricant chemistry (additves) on transmission system performance in specified manoeuvres, such as various drive cycles. For such an undertaking a detailed rheological model should be created to account for lubricant behaviour under a wide range of loading and shear conditions, including its rheological response at high pressure and shear, linked to lubricant chemistry. The work in Ref. [13] does not include such an approach. It uses empirical lubricant characteristics subject to non-Newtonian shear. The current study, on the other hand includes detailed measured lubricant behaviour with the inclusion of different viscosity modifying species subject to a wide range of loading and shear as described later. This is one of the main contributions of this study.

When analysing gear transmission systems, transient dynamics is required to address system non-linearities and to ascertain the extent of system stability and attainment of desired periodic motions. For example, when utilising a transient dynamic model, it is possible to capture the effect of resonance on the gear pair performance. In addition, this allows accurate prediction of dynamic loads of contacting teeth pairs. The inertial dynamics are normally under-estimated with quasi-static analyses. Therefore, inertial dynamics is required for evaluation of both transmission efficiency and NVH assessment. Thus, in the current study, a 4 DoF gear dynamics model, along with realistic TCA data is employed to account for the impact of the dynamic response of the hypoid gear pair on its conjunctional inefficiency. More importantly, realistic rheological data describing the high pressure and high shear response of fully blended axle lubricants are employed. The thermal effects due to the presence of flank friction are also accounted for by using a time-efficient analytical approach, yielding more realistic predictions of conjunctional inefficiency. The methodology presented in this paper distinguishes between the performance of fluids containing different types of viscosity modifiers (VM).

In the currently expounded tribodynamics model, an appropriate analytical thermal network partitioning model for the generated heat in the contact between the conjunction surfaces is used. The lubricant shear heating effect is also taken into account through Time Temperature Superposition (TTS) method with the use of specifically measured data for each lubricant, based on its TTS coefficient. In addition, in the current study, the contribution due to boundary friction is obtained, based on incorporating the measured surface roughness data for the contacting gear pair, including the combined average radius of curvature of asperity tips, composite RMS roughness values and the combined asperity peak density values. This approach improves the predictions of real system behaviour.

It is shown that the developed analytical approach in the form of a thermal tribodynamics model establishes a link between the rheological properties of transmission fluid's additives in the form of various viscosity modifiers and gear dynamic performance, measured through transmission efficiency. In particular, the model is shown to be sufficiently sensitive to the effect of various viscosity modifiers.

2. Problem formulation

A torsional gear dynamics solver is developed to predict the dynamic response of the gear pair under a wide range of input torque data. The conjunctional friction is considered in the equations of motion. Employing a gear dynamics model into the analysis of the conjunctional friction enables the identification of regions of interest during the operation of the gear pair, such as the first primary resonance. When operating at or near these regions, tooth separation occurs, leading to changes in the frictional losses with respect to an equivalent quasi-static analysis [12,14]. An analytical approach is employed to calculate the magnitude of friction at each time step of simulation. The viscous and boundary components of friction between the meshing gear teeth pairs are taken into account in the friction model. The temperature rise at the central region of the EHD conjunction is accounted for by employing an analytical thermal model. The calculation of the component of boundary friction is performed, using the Greenwood-Tripp model [15]. The Greenwood-Tripp parameters describing the roughness features of the gear teeth surfaces are determined by utilising a 3D Alicona™ optical interferometer and following the procedure recommended by Arcoumanis et al. [16]. A run-in pinion tooth was employed to conduct the surface topography measurements. The instantaneous contact geometry and the instantaneous meshing stiffness are determined through use of TCA data, available in published literature [14] to represent a realistic set of hypoid gear pair design.

The method of study is based on the decomposition of the problem into two separate models including a 4-DoF torsional gear dynamics model and a friction model. Each model is solved separately with the output of the other forming its input, iterating to a single solution. This is a co-simulation approach. The methodologies employed for each model are described in the following sections.

2.1. Gear dynamics model

A 4-DoF torsional gear dynamics model (Fig. 1) is used to predict the dynamic response of the hypoid gear pair. The degrees of freedom considered are the angular displacements of the pinion shaft, the pinion, the gear wheel and the gear shaft. An additional integration in time domain is performed to calculate the Dynamic Transmission Error (DTE) through use of its time history of response [17]. The equations of motion for the 4-DoF lumped parameter torsional gear dynamics model are expressed as:

$$\ddot{\varphi}_s = \frac{1}{I_s}\left[-k_{t1}\left(\varphi_s-\varphi_p\right)-c_{t1}\left(\dot{\varphi}_s-\dot{\varphi}_p\right)+T_s\right] \tag{1}$$

$$\ddot{\varphi}_p = \frac{1}{I_p}\left[-R_p(k_m f+c\dot{x})+k_{t1}\left(\varphi_s-\varphi_p\right)+c_{t1}\left(\dot{\varphi}_s-\dot{\varphi}_p\right)+T_{fr,p}\right] \tag{2}$$

$$\ddot{\varphi}_g = \frac{1}{I_g}\left[R_g(k_m f+c\dot{x})-k_{t2}\left(\varphi_g-\varphi_w\right)-c_{t2}\left(\dot{\varphi}_g-\dot{\varphi}_w\right)+T_{fr,g}\right] \tag{3}$$

$$\ddot{\varphi}_w = \frac{1}{I_w}\left[k_{t2}\left(\varphi_g-\varphi_w\right)+c_{t2}\left(\dot{\varphi}_g-\dot{\varphi}_w\right)-T_w\right] \tag{4}$$

Fig. 1 provides a schematic diagram of the torsional gear dynamics model employed in the present study. As shown, the torsional elasticity of