A transient EHL contact model capturing system-level spur gears dynamic behavior

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Mechanical transmissions are responsible for significant noise generation and power losses in automotive and wind energy applications. New regulations on pollution in terms of noises and the ever increasing demands for improved energy efficiency are pushing the state of the use in gearboxes to lighter, more efficient and more durable designs. On the other hand, little is known about loss mechanisms and their interactions with other crucial performance attributes such as noise and durability. A solution lies in the development and improvement of computational tools that predict the drivetrain system-level dynamics. One of the basic components of a transmission system are gears. In the vast majority of the cases, vibrations in a single rotating gear turn out to be above the audible hearing range (20 Hz to 20 KHz) which makes its dynamic response negligible with respect to NVH (Noise Vibration Harshness) analysis. Yet, when two or more gears are meshing, they excite each other and the surrounding structure with a frequency that depends on the time variation of the meshing conditions. This phenomenon usually happens at a substantially lower frequency. The vibrational behavior of a geared transmission mostly depends on three characteristics: mass distribution, stiffness and damping.

As pointed out by Andersson [1], when a gear is loaded at a contact location, the linear part of the displacement can be separated from the non-linear contact displacement following the approach in [1]. In most of the cases, gears work in lubricated conditions, mainly to reduce friction and wear, provide cooling and remove debris. Indeed, for lubricated surfaces in relative motion with respect to each other, the lubricant is dragged by shear in the convergent gap, separating as such both surfaces due to increased hydrodynamic pressure. Because of the non-conformity of the contact between gears, the local pressure increases drastically up to the order of $1-5$ GPa, resulting into a significant elastic deformation of the opposing surfaces. Moreover, at such high pressures, the liquid lubricant becomes compressible and its viscosity rises locally (i.e. piezo-viscosity), resulting in a local solidification of the lubricant. Hence, this type of lubrication regime is denoted as elasto-hydrodynamic lubrication (EHL).

In this research, the linear load-deflection part is determined by an FE-based approach [2, 3, 4] which is capable of capturing phenomena such as the deformation of the gears body (fundamental in e.g. lightweight gears) and the global deformation of the teeth (e.g. tooth bending) assuming quasi-static loading conditions. The non-linear contact damping and stiffness are considered and determining them is the scope of the presented research. In general, EHL contacts represent a complex multiphysical problem, due to the different nature of the fluid and the solid part of the problem. EHL contacts are typically studied with either Reynolds equation in combination with hertzian contact theory, or more recently, by advanced Computational Fluid Dynamics (CFD) and Fluid-Structure-Interaction. Both turn out to be too computationally expensive to be employed in a multibody environment. Therefore, throughout the last decade, the multibody community started developing analytical or semi-analytical models to describe the most significant phenomena occurring in this type of contacts [5] [6].

This work aims at combining and analyzing different analytical models in order to accurately describe the dynamic behavior of the contact together with the aforementioned FE-based approach for the gears bulk compliance. The input for the contact model is the non-linear contact penetration δ, which is defined as the distance between the undeformed surfaces of the two bodies along a common normal direction (with the negative sign if the body volumes do not penetrate each other). As Figure 1 shows, the penetration can be written as

$$\delta = \varepsilon - h_c, \tag{1}$$

where ε is the deformation of the solids and h_c is the central fluid film thickness. Due to the similarity between

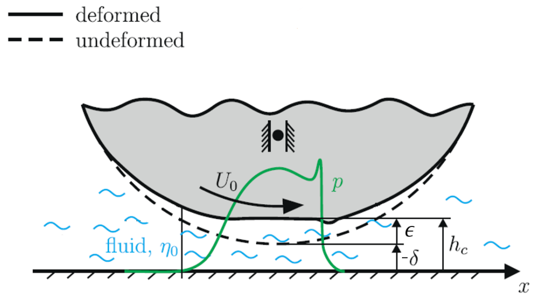

Fig. 1: Contact penetration arrangement [7]

the pressure distribution in EHL contacts and in Hertzian ones [8], ε can be well approximated by the formulation developed by Weber and Banascheck [9] for dry contacts, while h_c is determined by the model proposed by Moes [10] for steady-state conditions. The damping of the lubricant is accounted for by the formula proposed by Wiegert et al. [7], while the damping of the solids is neglected. The contact problem is then written as a set of two equations where the contact force and the central fluid film thickness are the unknowns. The solution can be computed by any Newton-like method for which a semi-analytic formulation of the Jacobian matrix is given.

Finally, the simulation results employing the contact model together with the linear compliance are presented by means of the dynamic transmission error (DTE) and damping coefficient curves during the gears meshing for different operating conditions. The damping coefficient is described statistically, by considering its mean and its variance during one gear revolution. The mean value represents the global effect the damping has on the gears dynamics which is highly influenced by the rotational speed of the gears and by the applied torque. The variable contribution, instead, varies depending on the contact conditions (e.g. surfaces speed, radii of curvature, tooth pairs in contact etc.) affecting the gears dynamics on the tooth-passing scale. The results carried out by the simulations are then compared against experimental test data acquired on a power-recirculation test-rig assessing the accuracy of the modeling technique. The DTE error curves are compared both in terms of shape as well as in terms of their spectrum.

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