SIMULATION OF RACK-PINION GEARS IN STEERING SYSTEMS USING ELASTIC MULTIBODY MODELS

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ABSTRACT

In this work the use of elastic multibody systems for noise, vibration, and harshness investigations in electric power steering systems is discussed. The gears in these systems can get highly stimulated in different driving situations and may produce undesired vibrations. Rigid multibody systems may represent the system dynamics very well, but elastic deformations in flexible systems are neglected with this method. Elastic multibody systems represent a good compromise between computation time and accuracy in the elastic deformations. This work presents an implementation of the software package Gear Train Module in the simulation environment at Bosch Automotive Steering. With this implementation it is possible to use elastic multibody systems for dynamic calculations of rack-pinion gears in automotive steering systems.

1 INTRODUCTION

Recently the automotive industry has seen a strong gain in the use of electric components. Also, in the field of steering systems hydraulic concepts have effectively been replaced by more efficient electric power steering (EPS) concepts. Instead of hydraulic pressure, the driver's steering input is supported by an auxiliary steering torque, generated by an electric servo drive. A summary of different EPS concepts can be found in [1]. The mostly used design in upper class cars is the EPS *axle parallel* (EPSapa) concept, where a belt driven ballnut gear transforms its rotary motion into a translative motion of the rack, see Figure 1. A great advantage of these systems is the low energy consumption. Additionally, they allow extra features like supporting lane departure warning systems, automatic parking assistance and even highly automated driving.

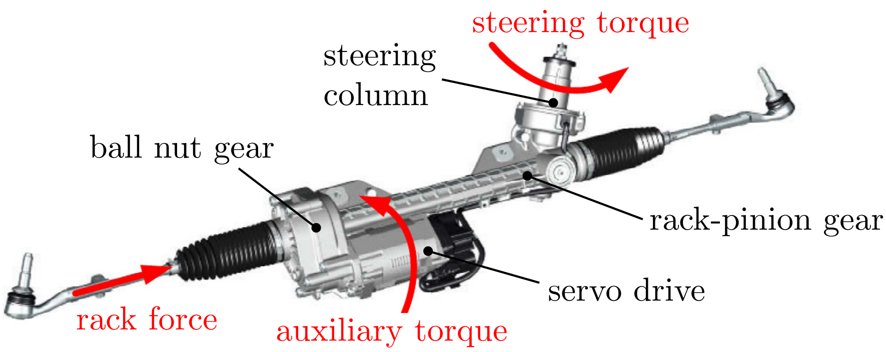

Figure 1: Acting forces on an electric power steering (EPSapa) [1].

However, one important challenge in developing EPS systems is the handling of arising vibrations from additional drive and gear components. Especially in gearing parts, different undesired phenomena may occur like toothing rattle or clunk. They may appear to the driver as audible noise or even as haptically sensible vibrations on the steering wheel and must be prevented to ensure a high driving comfort. Since these phenomena have a highly dynamic character, their identification requires efficient transient simulation approaches, like multibody systems (MBS). However, phenomena related to structural elasticity are often not represented correctly in rigid MBS. Also, finite element (FE) analyses are not appropriate for these tasks because despite offering very accurate results, they have the drawback of very big computation times. In such cases, elastic multibody systems (EMBS) offer a good compromise between fast computation and sufficient accuracy related to the elastic deformation. In [2] the use of EMBS on flexible gear wheels was shown.

This work discusses the use of the EMBS method to calculate accurate contact forces in the rack-pinion gear of an electric power steering. The software package *Gear Train Module* (GTM) [3] was used and extended to set up and simulate rack-pinion models.

2 ELASTIC MULTIBODY SYSTEMS WITH CONTACT

The concept of EMBS in the floating frame of reference formulation is described in detail in [4]. The equation of motion for a free body is

$$\boldsymbol{M}^i \underbrace{\begin{bmatrix} \dot{\boldsymbol{v}}^i \\ \dot{\boldsymbol{\omega}}^i \\ \ddot{\boldsymbol{q}}^i \end{bmatrix}}_{\dot{\boldsymbol{z}}^i_{II}} = -\boldsymbol{h}^i_e - \boldsymbol{h}^i_\omega - \boldsymbol{h}^i_p - \boldsymbol{h}^i_d, \tag{1}$$

where $\boldsymbol{M}^i$ is the symmetric mass matrix, containing mass, inertia and, coupling terms. The vector of generalized accelerations $\boldsymbol{z}^i_{II}$ holds the translational and rotational accelerations $\dot{\boldsymbol{v}}^i$ and $\dot{\boldsymbol{\omega}}^i$ of the rigid body motion and the elastic acceleration $\ddot{\boldsymbol{q}}^i$ with $\dim(\boldsymbol{q}^i) = n_i$. They are obtained via model order reduction from the original FE system with $\dim(\bar{\boldsymbol{q}}^i) = N_i$ by $\boldsymbol{q}^i \approx \boldsymbol{V} \cdot \bar{\boldsymbol{q}}^i$, containing the projection matrix $\boldsymbol{V} \in I\!R^{N_i \times n_i}$. The force terms on the right hand side consist of inner forces $\boldsymbol{h}^i_e$, volume forces $\boldsymbol{h}^i_\omega$, surface forces $\boldsymbol{h}^i_p$, and discrete applied forces $\boldsymbol{h}^i_d$.

There exist different approaches for the consideration of mechanical contact in MBS, see [5]. One detailed approach is the general 3-D node to surface contact search, where the FE meshes of the bodies are used for mutual intersection checks. In case of intersection, a penalty force is calculated based on the penetration depth. Benefits of this contact model are, that malpositioning of gears can be handled natively and elastic deformations of the bodies are respected by transforming the contact elements into the deformed configuration. However, because of the high computational effort of a node to surface contact search it is mandatory to execute a coarse collision detection upfront. In the special case of gear contact, based on the current tooth positions additionally a preselection of the contact sets can be done to limit the computational effort even further.

3 MODELING AND SIMULATION OF A RACK-PINION GEAR

The following shows the setup and results of an impact test, that was simulated in GTM with a rack-pinion model from [6]. The FE mesh of the model, with the components rack (red) and pinion (white), is illustrated in Figure 2. To classify the EMBS results, comparisons to corresponding rigid MBS and FE simulations are made.

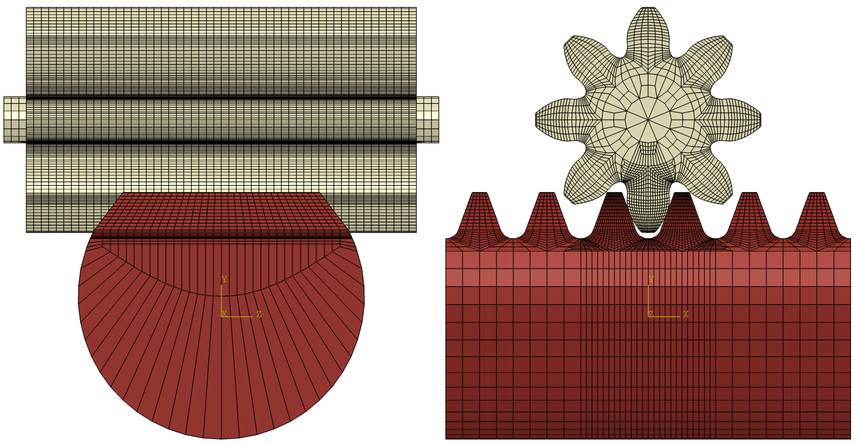

Figure 2: Rack-pinion gear test model [6].

For this test, the rack is fixed in all spatial directions, the pinion has a rotary degree of freedom around its main rotation axis with a starting angular velocity of 100 rad/s. The penalty contact stiffness is set to $3.5\mathrm{E}^7$ N/m. In [2] it was shown that modes up to 80 kHz are needed to provide an adequate result in the contact force. Both bodies were reduced by modal truncation to 200 elastic degrees of freedom.

The resulting cumulative contact forces and angular velocities of all investigated methods are shown in Figure 3. It can be seen, that the FE and EMBS results match very well. They contain significant oscillations due to elastic deformations during contact, while they are missing in the rigid model. The resulting simulation time for the EMBS method on an Intel Core i7 computer is shown in Table 1. It provides results 2.5 times slower than the rigid MBS, but is about 15 times faster then the FE simulation.

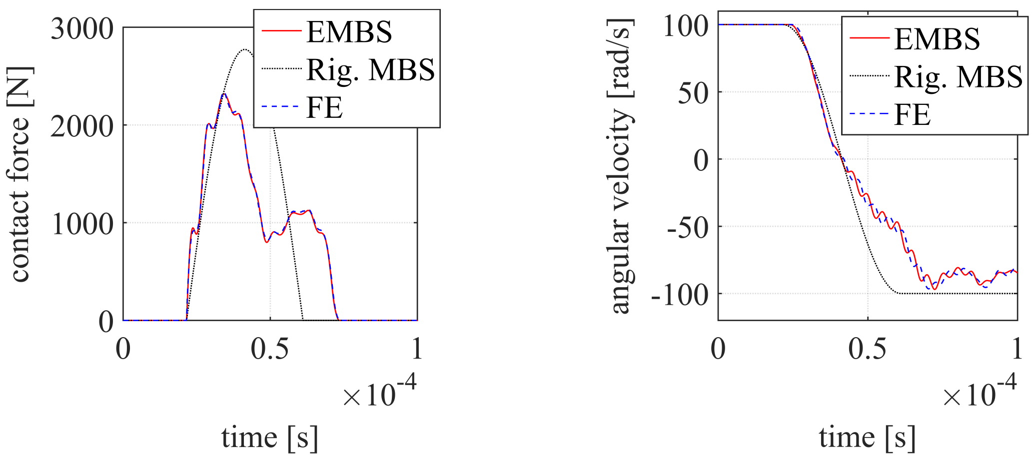

Figure 3: Cumulative contact forces and angular velocities in the different configurations.

Table 1: Comparison of simulation time (in seconds).

	EMBS	Rigid MBS	FE analysis
calculation time	210	80	3180

4 CONCLUSION

This work showed the benefit of using EMBS in gearing calculations within steering systems. With EMBS the elasticity of the gears can be considered in rack-pinion gearing calculations, enabling the identification of structural vibrations. Anyway, the computational effort is much smaller compared to a finite element analysis.

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