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A linear frequency domain solver workflow for fast simulation of transmission systems

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Abstract

When key design parameters are varied in the early design phase of transmission systems, a fast computation of the dynamic properties is essential. This is especially true for automotive transmissions modelled as multi-body systems consisting of mechanical components like rotating rigid and elastic shafts interconnected by joints like gear contacts and supported by bearing joints.

A standard transient time integration for such systems based on the BDF integration method for a floating frame of reference approach is described in [1]. In case of simulating typical application cases like periodic steady-state motion, the computation may take minutes up to several hours depending on the model complexity, as transient oscillations from fast moving components may decay slowly. Solving the equations of motion in frequency domain [6] based on a linearized model, which assumes the first order Taylor approximation of the nonlinear equations of motion leads to a second order ODE with constant, time independent coefficient matrices, yields the steady-state results within seconds. The equations of motion in time domain of the linearized multi-body system are [2][3]:

$$\boldsymbol{M}\ddot{\boldsymbol{q}}(\boldsymbol{t}) + (\boldsymbol{D} + \boldsymbol{G})\dot{\boldsymbol{q}}(\boldsymbol{t}) + \boldsymbol{K}\boldsymbol{q}(\boldsymbol{t}) = \boldsymbol{f}(\boldsymbol{t}) \tag{1}$$

where **M, D**, **K** and **G** are the symmetric matrices of mass, damping and stiffness, and the skew-symmetric gyroscopic matrix, respectively. Vectors **q** and **f** contain displacements and external loads. While **M** and **G** describe bodies, **D** describes mostly joints, and **K** describes both bodies and joints. Eq. (1) is then solved in frequency domain.

This contribution presents a complete frequency solver workflow from the linearization to its final solution. The setup of the multibody model is analogous to the one in a standard transient time integration. All body types (rigid or flexible) may be used. As in standard time calculation, initial velocities are calculated or corrected. Then, using the kinetostatics solver [5], a loaded configuration of the model is sought for, i.e., external loads are applied and a static equilibrium position is computed resulting in preloaded bodies and joints, yielding in particular gear contacts being closed. This equilibrium position is the starting point for the linearization of the model, where effects from the rotation are included. The joint types are linearized by the method of finite differences or – in cases of relatively simple force laws – analytically [4]. The usual differential equation (1) of second order with constant coefficient matrices resulting from the linearization describes the small motion of the model around the linearization point, where constant rotation of the individual bodies around a fixed axis is assumed. This differential equation is then solved in frequency domain. On an equidistant frequency grid, the external loads are first transformed to frequency domain by FFT. The dynamic stiffness matrix [6] is generated and the corresponding linear equation system is solved for each non-negligible frequency load component. The results can be represented in the frequency domain as well as in recomputed the time domain.

The presented solver workflow is applied to a simple gearbox model as depicted in Fig.1 left side. In the application example three elastic gear shafts supported by tapered roller bearings joints in a rigid housing are connected through cylindrical gear joints. In total the model consists of about 430 DOFs. Angular motion of the input shaft (green) is realized through predefined motion. An oscillating moment is applied to the output shaft (yellow). Resulting vibrations from a linear frequency domain solution are compared against results of a complete time domain solution up to 2000 deg reference angle, see Fig.1 right side. As the gear joints couple both transverse and torsional motion of the three shafts, the position of one transverse component of the gear joint connection node at the left side of the layshaft (blue) is compared as an example. The time domain solution is accepted to be reliable, but it takes much more simulation time till a stationary state is reached: the CPU time comparison is 115.4 seconds versus 7.9 seconds in case of the presented frequency domain solver workflow.

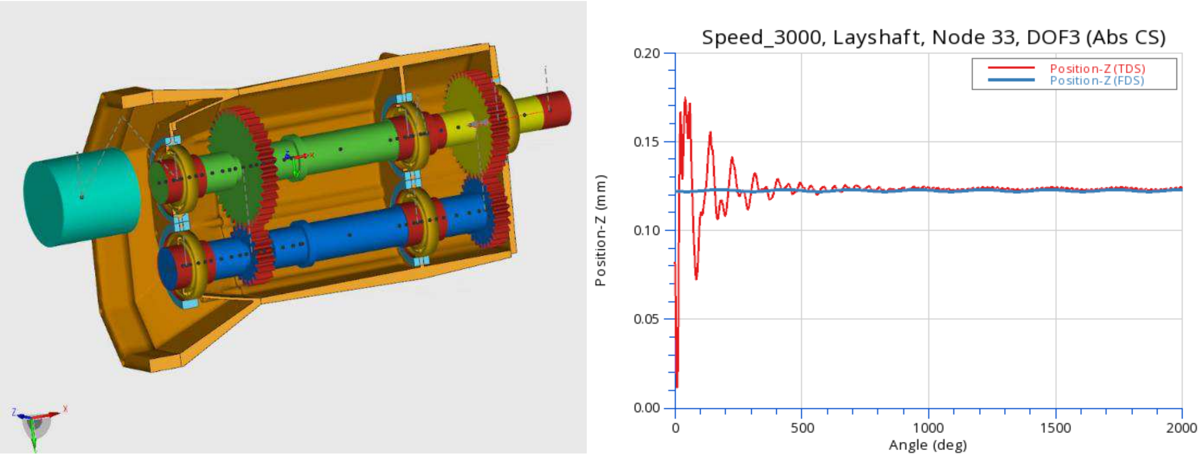

Figure 1: The model of a simple gearbox (left) and a comparison of motion results obtained from a time domain (TDS, red) and frequency domain (FDS, blue) solver (right).

References

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