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Multibody modelling and numerical simulations in drive train dynamics of road vehicles

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Multibody approaches are very powerful tools for the modelling and analysis of real mechanical systems. Design elements of drive trains perform large motion with both rotations and translations and there are many force interactions between particular parts. Therefore the modelling methodology based on the multibody dynamics is suitable for the analysis and optimization of drive trains. This paper introduces the methodology for the creation of particular models and for performing numerical simulations of various operational vehicle states. It addresses two solved case studies of real rally car drive train problems including experimental verification and validation.

The approach used in this work is based on generalized (Cartesian) coordinates, which lead to the mathematical model in the form of a set of differential-algebraic equations (DAEs). Holonomic rheonomic constraints between the coordinates described by vector $\mathbf{q}$ can be written using the vector notation $\mathbf{\Phi}(\mathbf{q},\, t) = \mathbf{0}$ and after their differentiation Jacobian matrix $\mathbf{\Phi}_q$ is obtained. Common mathematical model can be expressed as the set of differential-algebraic equations of index one in the form

$$\begin{bmatrix} \mathbf{M} & \mathbf{\Phi}_q^T \\ \mathbf{\Phi}_q & \mathbf{0} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}} \\ -\boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{g}(\mathbf{q},\, \dot{\mathbf{q}},\, t) \\ \boldsymbol{\gamma}(\mathbf{q},\, \dot{\mathbf{q}},\, t) \end{bmatrix} \tag{1}$$

by the double differentiation of the constraint equations with respect to time. Vector $\boldsymbol{\gamma}(\mathbf{q},\, \dot{\mathbf{q}},\, t)$ represents the remaining terms after the constraints differentiation. Solution of equations of motion (1) can be based e.g. on elimination of Lagrange multipliers [2] and further direct integration of the underlying ordinary differential equation. Vector of Lagrange multipliers λ is introduced in Eq. (1). Matrix $\mathbf{M}$ is the global mass matrix of the multibody system and vector $\mathbf{g}(\mathbf{q},\, \dot{\mathbf{q}},\, t)$ contains centrifugal and Coriolis inertia forces, elastic and damping forces and other externally applied forces including the gravity.

The first solved problem is aimed at sequential manual transmissions which are common sources of impacts and contacts that impose time varying excitation. Design engineers need sufficient computational methodologies and tools in order to predict dynamic behaviour of developing systems. The computational model of the particular driving system with a sequential gearbox was built up in MSC.Adams. The model visual representation is shown in Fig. 1 on the left. The purpose of the model is to simulate shifting between the third and the fourth gear stage because it is the most common maneuvre during a rally car race.

At the input of the driving chain, all the engine parts are modelled using a body with reduced inertia properties (see position 1 in Fig. 1), which include inertia of all relevant rotating parts of the engine. Using torsional elastic coupling of the clutch, body 1 is connected to the inner

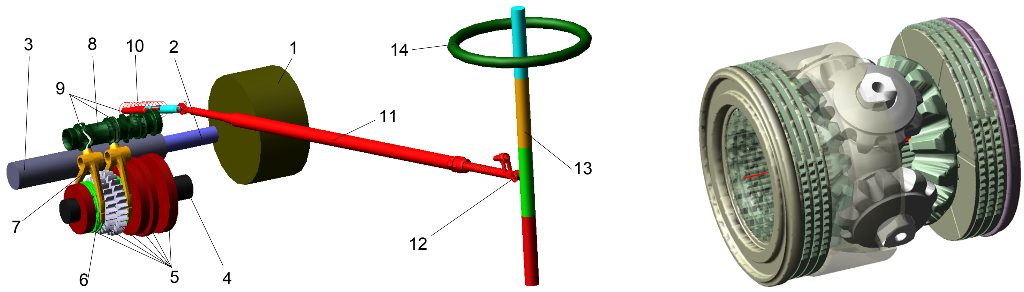

Fig. 1. Visualization of the gearbox model (*left*) and the differential model (*right*)

primary shaft (position 2 in Fig. 1), which goes through whole body of outer primary shaft (position 3) and at the end, both shafts are coupled by gear coupling. Body 3 – outer primary shaft – includes reduced mass properties of all primary gears. Torque flow is transmitted through the third or the fourth gear stage to the secondary shaft by means of a claw clutch between dog-rings (position 6) and secondary gears (position 5). Gear shifting is performed by moving a flexible gear-shift lever (position 13). The translational motion of the rod is transformed by gear coupling to the rotation of a selector (position 8) with guide curves (position 9), which translate corresponding forks (position 7) with dog-rings along the axis of the selector. In this manner, the desired gear stage is chosen. Model's input characteristics are engine angular velocity, engine driving torque and shifting force, which is measured by sensor located between the gear-shift lever and the rod. The aim of the model is to simulate the shifting process and to evaluate the overall shifting time. The results obtained using the computational model were compared to experimental data. The comparison of computed and experimental data for the ideal case (correct shifting without collision), that the presented model of the state dependent shifting force is in good compliance with the experimental data. The resultant driving torque also corresponds to the experimental data.

The second topic deals with approaches to the modelling and dynamical analysis of a special class of automotive differentials called limited-slip differentials. It is also the problem characterized by large motion and contact and friction interactions. The differential is modelled in complex manner with detailed interior structure (see Fig. 1 on the right) and with other elements of the whole drive train. The developed multibody differential model can be utilized in wide range of simulation tasks. In order to generate typical locking characteristics the model was employed in nonlinear dynamical analyses with prescribed input torque and output revolutions, which are motivated by typical experimental tests. The locking characteristics are composed of total torque transmitted by both semi-axles, which is plotted on the horizontal axis, and the difference of theses torques, which is plotted on the vertical axis. The resulting characteristics were successfully compared with experimental results.

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