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ADVANCED COMPUTATIONAL MODELS OF ELASTOMER PARTS IN AUTOMOTIVE TECHNOLOGY

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Summary: In automotive industry, the virtual prototyping means the widest possible use of CAD (computer-aided-design) and CAE (computer-aided-engineering) programs applied for validation and testing of the design before the first prototype. For the success of this method, all substructures of the complex computational models shall be designed similarly. The paper deals with the computational models of elastomer parts. These have acquired an increasingly important role in the design of modern powertrains and automobiles when reducing vibrations and noise, as well as increasing reliability and durability.

Key words: elastomer, rheological models, hysteretic damping, optimization algorithms.

INTRODUCTION

The design and development of new products in automotive technology has historically been based primarily on the experience of developers and designers and their own opinions when designing initial concepts. Any further development efforts were then directed to the production of a physical prototype to ensure the functionality and parameters of the designed product. In most cases, however, the initial concept did not meet the expectations and the initial prototypes revealed many weak points. The design had to be then revised several times, which greatly lengthened the process of development of new products and increased costs.

Currently used method of virtual prototyping makes use of CAD and CAE software applied for validation and testing of the design before the first prototype. An important factor that influences the development process of a virtual prototype is the interdependence of individual software tools and relevant level of applied computational models of individual substructures. Detailed computational models of elastomer parts, which reflex their complex rheological properties, especially frequency and temperature dependences of their parameters, are not part of commercial FEM and multibody software.

1. TORSIONAL VIBRATION DAMPERS

Elastomer parts have an increasingly important role in automotive industry, especially in solving noise and vibration problems reflecting the ever-increasing requirements for reliability and durability. In order to reach ambitious global aims for reducing fuel

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consumption and CO_2 emissions, car producers apply "downsizing" (1). This means they develop smaller engines with fewer cylinders with significantly increased mean effective pressure. The increased pressure and the ignition sequence of small engines leads to strong torsional vibrations in the powertrain, however.

1.1 Rubber torsional vibration dampers for gasoline engines

With torsional vibration dampers, these rotational irregularities are balanced out and the vibrations in the crankshaft are kept away from the belt or chain drives and the auxiliaries. A torsional vibration damper is fixed to its free end to damp the vibrations in the crankshaft. This consists of a rubber spring and an inertia ring which are processed in different ways. The high torque pressed-in torsional vibration damper (see Fig. 1) is a cost-effective solution for higher gasoline engine powers.

Source: (2)

Fig. 1 - High torque pressed torsional vibration damper for gasoline engines

A vulcanized rubber damper (see Fig. 2) meets high demands for thermal load capacity, damping, and concentricity.

Source: (2)

Fig. 2 - Vulcanized torsional vibration damper for gasoline engines

1.2 Rubber torsional vibration dampers for diesel engines

Designing of the rubber damper for diesel engines in passenger cars or commercial vehicles is quite a difficult task because mechanical and particularly thermal load of the rubber spring is considerably higher if compared with gasoline engines.

Fig. 3 shows the design of a rubber dumper which was successfully applied to in-line six-cylinder diesel engine for commercial vehicles thanks to the use of own advanced computational models of rubber parts.

Source: Authors

Fig. 3 - Vulcanized torsional vibration damper for diesel engines

2. DYNAMIC COMPUTATIONAL MODELS OF RUBBER PARTS

Viscoelastic elastomer parts are difficult to model correctly by the finite element method due to their frequency-dependent mechanical properties. In most cases, a linear spring and a linear damper are used to model the elastomer in FEM or multibody software. However, the simulation results may not match well with results of measurements in the time or frequency domain. Multi-parameter elastomer rheological models with a suitable structure can contribute to solving this problem (5).

2.1 Single degree of freedom system with viscous damping

The governing equation of a torsional system with one degree of freedom for the viscous damping model has been formulated as

$$J\ddot{\varphi} + k\dot{\varphi} + c\varphi = \overline{M_0}e^{j\omega t}, \tag{1}$$

where J, k and c represent the moment of inertia, viscous damping coefficient and stiffness, respectively (3). $\overline{M_0}$ and ω are the amplitude of the harmonic loading and the exciting frequency, respectively. The dissipated energy per period is given by

$$E_{dis} = \pi k\omega^2 \left|\overline{\phi_0}\right|^2, \tag{2}$$

where $\overline{\phi_0}$ represents the amplitude of torsional vibrations.

2.2 Single degree of freedom system with hysteretic damping

The governing equation of a torsional system with one degree of freedom for the hysteretic damping model has been formulated as

$$J\ddot{\varphi} + \frac{h}{\omega}\dot{\varphi} + c\varphi = \overline{M_o}e^{j\omega t}, \tag{3}$$

where J, h and c represent the moment of inertia, hysteretic damping coefficient and stiffness, respectively (4). $\overline{M_0}$ and ω are the amplitude of the harmonic loading and the exciting frequency, respectively. The dissipated energy per period is given by

$$E_{dis} = \pi h \left|\overline{\phi_0}\right|^2 \tag{4}$$

and is thus independent on frequency.

In harmonic oscillation both above mentioned damping models are mutually transferable. There is a simple relation between the hysteretic damping coefficient and coefficient of viscous damping

$$h = \omega k\,. \tag{5}$$

Hysteretic damping can also be formulated with the use of loss factor

$$d = \frac{h}{c} = \frac{\omega k}{c}\,. \tag{6}$$

Rubber springs of torsional vibration dampers have damping properties similar to the mentioned hysteretic model. As proved experimentally, in harmonic distortion with particular angular amplitude the dissipated energy in the rubber spring during one oscillation period in the frequency up to several hundreds Hz is almost constant. However, immediate use of hysteretic damping model in the computational models of powertrains is not possible, because exciting moments in crankshafts are periodical and for dynamic computation in the frequency or time domain it is necessary to consider several dozen of their harmonics.

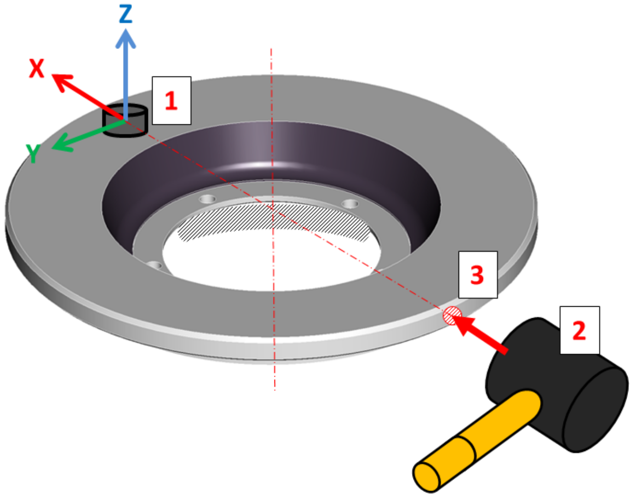
2.3 Multi-parameter elastomer rheological models

Rheological properties of elastomer parts can be determined using suitably designed technical experiments (5), while known simple damping models can only sporadically be applied to obtain first approximation.

Much better approximation of rheological properties can be achieved with multiparameter rheological models whose structure can be designed with the use of linear elastic and damping elements. Applying the method of least squares and efficient optimization algorithms, the parameters of these elements can be set so that the multiparameter model in the specified frequency range approximates rheological properties of the given part with the desired accuracy.

3. EXPERIMENTAL DETERMINATION OF RHEOLOGICAL PROPERTIES OF THE RUBBER SPRING IN TORSIONAL VIBRATION DAMPER

Rheological properties of the rubber spring in the dynamic torsional dampers as shown in Fig. 3 can be determined with the use of a dynamic shear modulus of the rubber mixture G and the loss coefficient d, where both quantities are temperature-dependent. For different temperatures, the rubber springs parameters can be determined e.g. from the measured waveforms of damped oscillations of the seismic ring, while the damper flange is tightly connected to the rigid base frame. For the arrangement of the technical experiment see Fig. 4.



Source: Authors

Fig. 4 - Measurement of the damper ring radial damped oscillations
for constant temperature
(1 - three-axis acceleration sensor, 2 - excitation hammer, 3 - point of impact)

Since it proved to be rather difficult to cause purely torsional vibration mode of the damper ring by the excitation hammer, measurement with the use of radial sliding mode was applied as shown in Fig. 4. The applied three-axis acceleration sensor enables simultaneously check whether the stroke of the excitation hammer caused dominant radial oscillation of the damper ring on the x-axis. Measurement of time histories of the damped oscillations was carried out for different temperatures within the range of corresponding operating temperatures of the rubber damper. For an example of measured waveforms see Fig. 5.

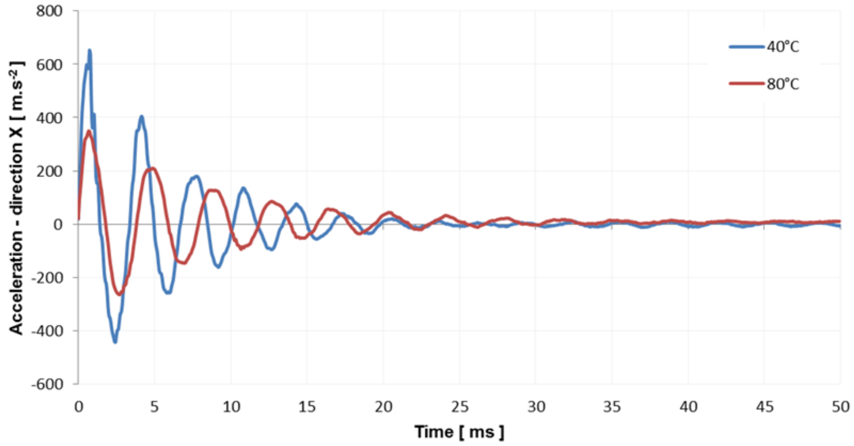


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Fig. 5 - Time course of damper ring radial damped oscillations for two temperatures

If we assume that the amplitudes of damped oscillations are decreasing according to the exponential function

$$A = A_0 e^{-bt}\,, \tag{7}$$

where A_0 represents the initial amplitude and

$$b = \frac{k}{2m}\,, \tag{8}$$

is the damping constant, m is the mass of the damper ring.

The damping ratio is given by

$$\lambda = \frac{A_i}{A_{i+1}} = e^{b\,T_d}\,, \tag{9}$$

where T_d is the damped oscillation period. The damping constant can be expressed as

$$b = \frac{\ln \lambda}{T_d}\,. \tag{10}$$

The angular frequency of the damped radial oscillation of the damper ring can be expressed as

$$\omega_d = \frac{2\pi}{T_d} \tag{11}$$

and the undamped oscillation frequency is given by

$$\omega = \sqrt{\omega_d^2 + b^2}\,. \tag{12}$$

Radial stiffness of the rubber spring damper is formulated by

$$\mathrm{c} = \omega^2 m\,. \tag{13}$$

The viscous damping coefficient of the rubber spring is given by

$$k = 2b\,m\,. \tag{14}$$

By substituting variables from the equations (12), (13) and (14) for different temperatures in the equation (6) we obtain the loss factor dependence on temperature.

Dynamic shear modulus of the rubber spring is determined from the equation

$$G = \frac{c\,t}{S}\,, \tag{15}$$

where t is the mean thickness and S the cross section of the rubber spring.

Computed values for the rubber temperatures according to the Fig. 5 are shown in the Table 1.

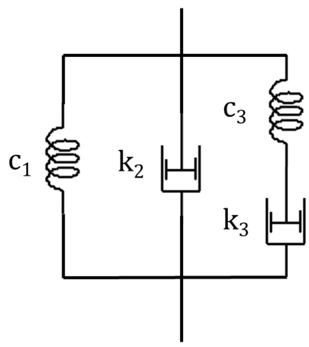
Table 1 – Temperature dependence of the rubber shear modulus and the loss factor

Temperature [°C]	Rubber shear modulus G [MPa]	Loss factor d [-]
40	2,88	0,21
80	2,08	0,15

Source: Authors

4. MULTIPARAMETER RHEOLOGICAL MODEL OF THE RUBBER SPRING IN TORSIONAL DAMPER

Torsional stiffness and loss factor of the torsional damper rubber spring may be set from the above parameters for the given operating temperature. Real and imaginary parts of complex torsional stiffness of the rubber spring are therefore frequency-independent constants, and this corresponds to the hysteretic damping model. The model is however applicable only in dynamic computations of harmonically excited mechanical systems.



Source: Authors

Fig. 6 – Designed rheological model of damper rubber spring

The aim is therefore to design a multiparameter rheological model consisting of linear elastic and damping elements, whose complex stiffness will approximate the components of complex stiffness of the rubber spring with a prescribed accuracy. Such a model will enable designing of a dynamic model of the powetrain in the frequency and time domain at any time waveforms of load effects. Suitable rheological model structure is shown in Fig. 6.

Using the optimization algorithms (6, 7, 8, 9), parameters of this model were computed. Fig. 7 shows approximation of the complex stiffness components of the rubber spring in the torsional damper using the rheological model as shown in Fig. 6.

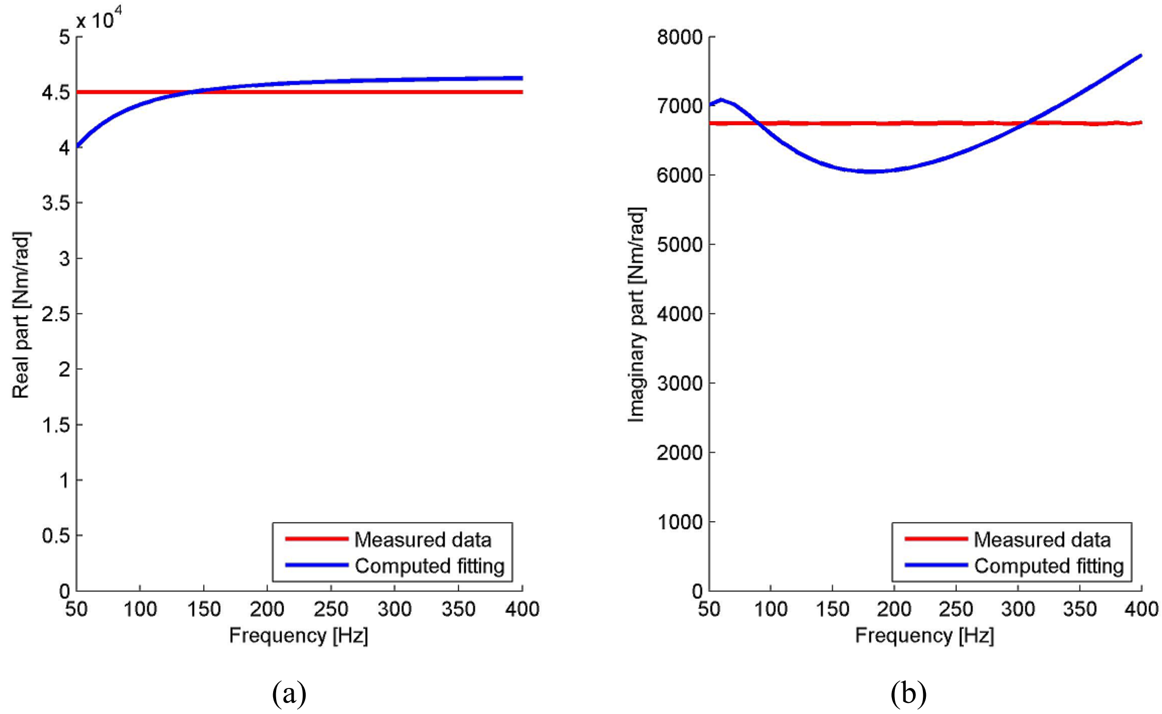


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Fig. 7 – Approximation of the rubber spring complex stiffness at the temperature of 80 °C
(a) real part (b) imaginary part

The designed model of the rubber damper was applied in a complex dynamic model of the crank mechanism in the in-line six-cylinder diesel engine for commercial vehicles. The computations were verified by torsional vibration measurements using contactless laser vibrometers. Some examples of computation results and torsional vibration measurements can be found in Fig. 8 – Fig. 10.

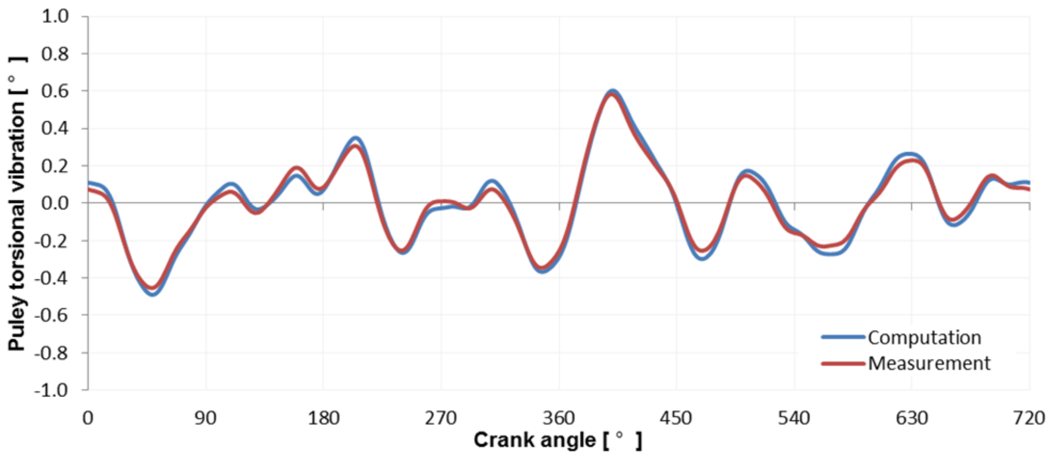

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Fig. 8 – Pulley torsional vibration with rubber damper at engine speed 2200 min^{-1}

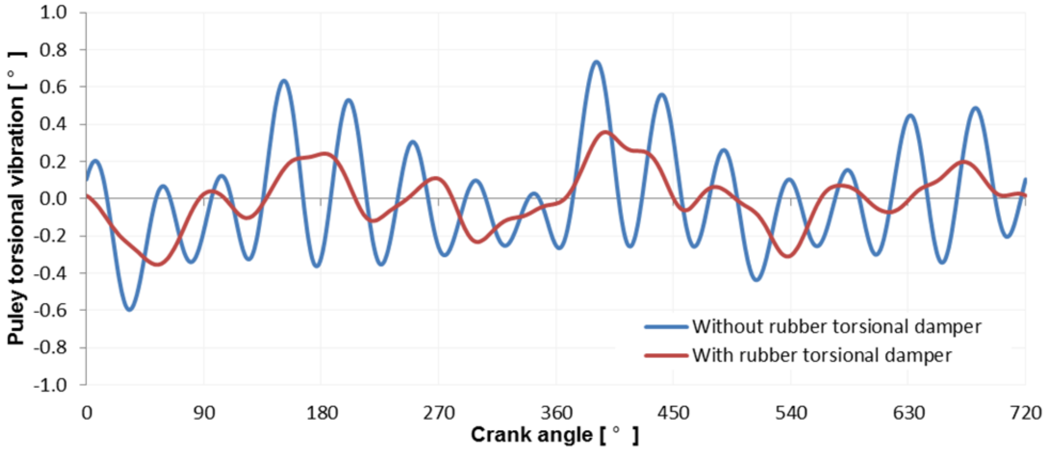

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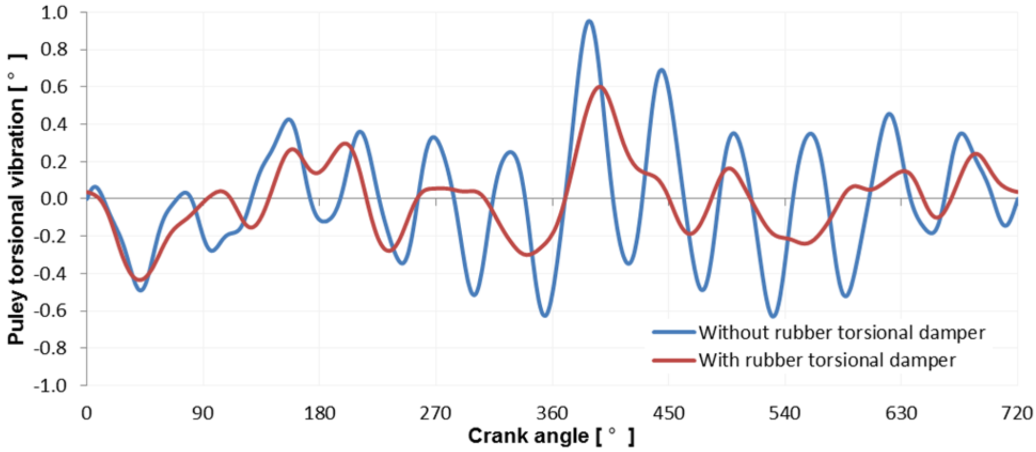
Fig. 9 - Measured pulley torsional vibration at engine speed 1860 min^{-1}



Source: Authors

Fig. 10 - Measured pulley torsional vibration at engine speed 2140 min^{-1}

CONCLUSION

In automotive technology, multiparameter rheological models composed of linear elastic and damping elements can be efficient to create computational models of complex mechanical structures that include elastomer parts. Successful implementation necessitates both appropriate initially designed structure of multiparameter rheological model, and particularly application of effective optimization algorithms.

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