Elastomer elastic element visco-elastic properties’ impact on damping in a vehicle suspension

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Abstract. Suspension structure, its elastic elements made of elastomers, is described in the paper. This allows to considerably decrease a vehicle’s unsprung weight improving its ride quality. The research is aimed at determining elastomer elastic element’s damping impact on a vehicle’s smooth ride quality. The impact estimate was carried out by comparing simulation modeling results of GAZelle NEXT vehicle’s front suspension and an equivalent suspension with an elastomer elastic element. Dual-mass single-axle suspension model was chosen as a calculation model. The modelling was performed in MATLAB/Simulink software package. Standard suspension key parameters, that is spring rate and shock absorber resistance characteristic, were experimentally obtained for each vehicle. Elastomer element elastic response was obtained at static necking and averaged with respect to loading and unloading characteristics. Damping capacity was estimated basing on the damped oscillations process realized on a special pendulum impact testing machine. “Relative” elastomer damping coefficient was calculated. All the measured parameters were reduced “a wheel” by grapho-analytical methods. Vibration acceleration root-mean-square values (RMS) on the car frame and normal awheel response of a vehicle’s motion under various speed conditions over smooth cobbled road served as assessment criteria. Roadbed microprofile mathematic model was realized basing on superposition harmonic function method with various oscillations frequency and amplitude. Analytical findings reveal that a standard suspension with a cylindrical spiral spring and a hydraulic shock absorber outperforms the elastomer elastic element structure in all the given motion modes with respect to the ride quality criteria. The research findings bring about the conclusion that elastomers lack visco-elastic characteristics necessary to provide better ride quality compared to the standard suspension structure.

1. Introduction

Non-metallic elastic elements in vehicles’ suspension are used as extra ones, allowing to appropriately adjust the elastic response, e.g. making it non-linear and increasing its energy capacity. Elastomer elastic elements, Figure 1, when certain requirements to the suspension’s elastic response observed, considerably decrease the vehicle’s unsprung weight and metal expenditure as well as improve the ride quality.
2. Experimental research
Elastomer has viscous friction and its elastic response is of non-linear character [1]. Figure 2 shows experimentally obtained real elastomer elastic response allowing to determine viscous friction value ($F_{\text{тр}}$) and rigidity. Elastomer viscous friction in its physical sense is close to «dry» friction. Elastomer viscous friction impact on the nature of dynamic processes in suspension is difficult to estimate as the type of friction is not determined by clear-cut physical boundaries. This is actually «dry» friction, but its value remains non-constant and depends on elastic element deformation $f$ and force $F$ acting on the elastic element. Moreover, elastomer viscous friction affects oscillatory processes damping rate in suspension and can be determined by “relative” damping coefficient $k_{\text{э}}$, similar to the damping coefficient of hydraulic shock absorber. An experiment was carried out to determine the elastomer “relative” damping coefficient. The experiment allowed to calculate damping logarithmic decrement in elastomer and the “relative” damping coefficient value.

The damping decrement allows to estimate the oscillations damping rate and is mathematically determined as a ratio of two subsequent maximum deviations of the system oscillations damping process, one way from the equilibrium state, Figure 3.

Relative elastomer resistance coefficient equals to

$$k_3 = \frac{2\ln d}{T} M$$

where: $M$ – the vehicle’s sprung weight, coming onto the elastomer; $d$ – damping decrement; $T$ – oscillation period. For the given elastomer element structure relative resistance coefficient equals to $k_3 = 1592$ N s/m.

It would be practical to analyze elastomer friction impact on the suspension critical performance characteristics – vertical acceleration root-mean-square values (RMS) $\sigma_z$, passed on to the sprung weight and normal response forces RMS $\sigma_F$, affecting the wheel in the point of tyre-road contact [6, 7].
3. Theoretical research

Mathematical models are applied to solve such tasks. A simpler mathematical (calculation) model is usually used when researching dynamic processes. Calculation models can become more sophisticated depending on the requirements to applicability and accuracy of analytical experiment result rates. The choice of the vehicle’s calculation model is based on certain assumptions [2-11], about oscillations independence in longitudinal and transverse planes. The vehicle’s transverse and longitudinal oscillations amplitudes are considerably lower than heave oscillations amplitudes, so they can be disregarded for a first approximation and the vehicle’s mathematical model can be linear.

At sprung weight distribution coefficient value ε of each suspension close to unity [3], weight fluctuations are described by independent set of equations, and design diagrams of a vehicle’s front and rear suspensions heave oscillations can be presented by single-mass and dual-mass calculation models.

Dual-mass single-axle calculation model [4], Fig. 4 is suggested for preliminary analysis of elastomer friction impact on the suspension damping properties.

Dual-mass single-axle calculation model allows to obtain consistent results at relatively simple initial motion equations. Sprung weight M, Fig. 4, is considered as a solid object incorporating all the weights elastically connected with it. Unsprung suspension weight \( m_{un} \) is connected with the sprung part by an elastic linkage with \( C_s \) rigidity and a smoother with \( K_n \) damping coefficient. \( K_a \) damping coefficient can take on \( k_a \) or \( K_a \) values corresponding to the hydraulic shock absorber damping coefficient.

Tyre impact on oscillating processes nature is considered \( C_{mm} \) by tyre stiffness reduced values and \( K_m \) tyre damping coefficient.

Differential equation sets describing calculation model weight motion with respect to viscous friction elements are given below [4, 5]:

\[
\begin{align*}
M\ddot{z} + K_n(\dot{z} - \dot{\xi}) + C_n(z - \xi) &= 0 \\
m_{un}\ddot{\xi} - K_n(\dot{z} - \dot{\xi}) - C_n(z - \dot{\xi}) + K_{hm}(\ddot{\xi} - \dot{q}) + C_{hm}(\xi - q) &= 0.
\end{align*}
\]  

where: \( z, \dot{z}, \ddot{z} \) – sprung weight motion, speed and acceleration respectively; \( \xi, \dot{\xi}, \ddot{\xi} \) – unsprung weight motion, speed and acceleration respectively; \( q, \dot{q} \) – microprofile ordinate and rate of ordinate change respectively; \( C_n \) – suspension stiffness equal to \( C_3 \) elastomer rigidity; \( K_n \) – shock absorber resistance coefficient (if equipped with elastomer it is considered to be equal to \( k_a \) relative resistance coefficient).

Shock absorber resistance coefficient should be calculated taking into consideration the suspension kinematics – reduced awheel.

Analytical methods of solving dynamics tasks imply the choice of disturbance forming methods passed on from the bearing surface to the vehicle’s wheels. When a vehicle is in motion, the disturbance affecting the wheel is formed by the microprofile. The road microprofile ordinates change randomly and presenting a law of their change in the form of a mathematical function is a difficult task. There are several disturbance forming methods [3, 4].

The simplest methods of road disturbance modelling are based on deterministic views and realized by analytical dependencies. A tyre is considered to smoothen outlines of irregularities assuming the irregularity profile to have sinusoidal form, symmetrical with respect to its mean value. \( q(s) \) microprofile function within each \( S_0 \) period can be analytically presented as a sum of trigonometric Fourier’s series, comprising an infinite set of the functions’ harmonic components. A possibility to present a continuous function in the form of Fourier integral allows to realize the task of forming microprofile function as a random process along the spectral density of its dispersions. The most significant normative document, classifying roadbeds is considered ISO 8608.
In some papers [3, 4] spectral density dependence of microprofile irregularities distribution is set as angular frequency function while transfer function is formed by a special filter. Forming random microprofile disturbance as a sum of harmonic constituents is an adequate method based on well-known views of the possibility to present a random process with set characteristics by a sum of harmonic constituents with frequency and amplitude selected in the appropriate manner. When solving a practical task of forming random continuous disturbance in the form of harmonic constituents correlation of amplitudes of road irregularities waves and their length is taken into account, identified by spectral density characteristic of microprofile dispersion. When modelling a disturbance, spectral microprofile characteristics are used which are obtained by any of the known methods. Harmonic constituents’ frequencies of a random disturbance process can be selected on a linear scale with the preset discretization rate or corresponding central frequencies of octave or 1/3-octave bands. A harmonic constituent amplitude is determined according to the dispersion value for the preset frequency band. Superposition harmonic function method with various oscillations frequency and amplitude is applied in the paper. These make up the random process spectrum (modelling with a polyharmonic function) [12]. Figure 5 shows the method flowchart.

Integral sum of spectral density function is the dispersions sum of interval ordinates of the frequency band in question. We consider that in the case of microprofile being presented by a superposition of periodical functions with various irregularity length (route frequency) and amplitudes, overall dispersion of microprofile spectral density can be determined as a sum of dispersions of the microprofile route frequency assigned slots, Figure 6. Spectral density overall dispersion constituents are regarded as harmonic process dispersions with the assigned slot medium frequency.

\[
D_j = \frac{S_q(\theta_{j+1}) + S_q(\theta_j)}{2} (\theta_{j+1} - \theta_j) \tag{3}
\]

where: \(D_j\) – function dispersion of microprofile spectral density for the route frequency interval \(\{\theta_j, \theta_{j+1}\}\); \(S_q(\theta_j), S_q(\theta_{j+1})\) – spectral function values at the route frequency values equal to \(\theta_j, \theta_{j+1}\).

Interval frequency mean value is accepted as equal to

\[
\theta_{jcp} = \frac{\theta_j + \theta_{j+1}}{2} \tag{4}
\]

where: \(\theta_{jcp}\) – interval medium frequency \(\{\theta_j, \theta_{j+1}\}\).

The road microprofile contains irregularities of different length, which can cause various frequency disturbances. The given parameters are related by ratios

\[
T_{\omega} = \frac{2\pi}{\omega} = \frac{s}{v} \tag{5}
\]

where: \(T_{\omega}\) – time of overpassing an irregularity [c]; \(\omega = 2\pi f\) – road irregularities frequency, [rad/s]; \(f\) – cyclic frequency, \(f = \frac{v}{s}\), [c\(^{-1}\)]; \(v\) – vehicle speed, [m/s]; \(s\) – irregularity length, m.
Differential equations are solved by numerical methods in MATLAB/Simulink programme. Figure 7 shows a microprofile model realized according to expressions (3) – (5). Process modelling algorithm allows for the disturbance effect frequency diversity at different (constant) vehicle speed, e.g. (40, 60, 80) km/h. Processes of changing vertical acceleration and normal response are time function. Selected process realization should be stationary. Elastomer friction is taken into account by “relative” elastomer damping coefficient.

![Figure 7. Smooth cobbled road microprofile model](image)

Oscillation damping efficiency by elastomer friction is compared to the oscillation damping efficiency by a hydraulic shock absorber of a “standard” suspension. Mathematical model of dual-mass elastomer suspension was performed in MATLAB/Simulink software package.

4. Results and conclusions

Figure 8 and 9 show analytics results – vibration accelerations (RMS) dependences $\sigma_z$, of normal response in a wheel contact patch $\sigma_F$ for elastomer suspension and the same dependences for hydraulic shock absorber suspension.

Comparative analysis of vibration acceleration RMS and normal responses for a standard hydraulic shock absorber suspension and elastomer suspension brings about the conclusion that elastomer elements lack oscillation damping efficiency under various speed conditions. Therefore, in order to obtain better smooth ride quality performance characteristics it is necessary to combine elastomer elements and hydraulic shock absorber. However, such structures have perspective advantages over sprung suspensions from the point of view of energy capacity and need further research and development.

![Figure 8. Normal response RMS awheel: 1 – for hydraulic shock absorber suspension; 2 – for «relative» elastomer damping coefficient.](image)

![Figure 9. Vibration acceleration RMS on the car frame: 1 – for hydraulic shock absorber suspension; 2 – for «relative» elastomer damping coefficient](image)
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