Experimental and calculation method for determining the parameters of a car suspension with an elastic elastomeric element

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Abstract. We present a suspension design the elastic elements of which are made of elastomers. This allows us to reduce the weight of unsprung parts of the vehicle significantly, which affects the quality of ride in a positive way. The aim of the study is to determine the elastic characteristic, the damping coefficient, the natural oscillation frequency of the sprung masses and the specific dynamic energy intensity of the suspension with elastomeric elastic elements. The determination of these parameters is carried out on the basis of experimental data got as a results of static and dynamic loading of a package of elastomeric elements. The elastic characteristic of the elastomeric element is obtained by static reduction and is not linear. Stiffness estimation of an elastomeric element for a specific point of characteristic or for a limited area is suggested. Also, there are significant friction losses during deformation, it is quite difficult to take them into account as a separate parameter when evaluating the elastomer damping properties. It is recommended to estimate their absorption capacity using a "conditional" damping coefficient. Its value is determined by the nature of the damped oscillations process under dynamic loading of an elastomeric elements package implemented on a special pendulum copra. The method of reducing the elastic characteristic and the damping coefficient of the elastomer to the "wheel" using graphoanalytic techniques is described in detail. The values of natural oscillation frequency and the energy intensity of the suspension with elastomeric elastic element are calculated based on the values of the suspension static and dynamic deflections and the forces in the tire contact with the supporting surface. Relying on experience in analyzing the standard car suspension parameters, it is concluded that using elastomeric elastic elements in the suspension design of trucks, buses and all-terrain vehicles is advisable.

1. Introduction
The elastic suspension element reduces the level of vibration loads transmitted from the support surface to the wheels and unsprung masses (body, frame) of a car (vehicle). Despite the fact that suspensions with non-metallic (mainly rubber) elastic elements have been known for at least 100 years, they have not been widely used in any country.

Being made of synthetic materials "rubber" elastic elements used at present have a relatively low durability, but which is more important do not give the necessary elastic response and the suspension energy intensity to the vehicle. In addition, the temperature range of their operation is limited due to the properties of the materials.
Elastomeric elastic elements are partially devoid of these disadvantages. Getting a package of elastomeric elements installed into the suspension (Figure 1), it is possible to reduce the weight of elastic elements and unsprung parts of the vehicle significantly, which affects the quality of ride in a positive way, reduces metal consumption and provides a number of other advantages. Using elastomeric elements is currently hindered by the fact that only certain techniques for calculating the mechanical properties and choosing the size of the elastomer have been elaborated [1], these techniques do not allow us to evaluate the operation of this object as a part of a car suspension. It is necessary to develop some additional techniques to determine all the key parameters of the suspension. These parameters are: elastic characteristics of the suspension (stiffness, static, dynamic and maximum deflections), natural oscillation frequency of the sprung masses of the car, deformation work and suspension capacity, a number of other parameters.

The elastic characteristic of the suspension represents the dependence of the normal force (reaction) $N_x$ acting on the wheel at the point of contact with the support surface from the suspension deflection. Therefore, all forces acting in the suspension (resistances of the shock absorber, elastic element, and others) must be pre-determined and reduced to the “wheel”. To determine the suspension elastic characteristic, it is necessary to get a kinematic diagram of the guiding device and the characteristic of the elastic element.

### 2. Experimental determination of the elastic characteristics

Tests to determine the elastic characteristic of the elastomeric elements package were carried out on a special bench-testing unit under elastic element static loading. The maximum load can be set in accordance with the elastomer strength or the kinematic features of the suspension guiding device (the maximum possible deformation of the elastic element). The design of the package of elastomeric elements eliminates friction between the elements and the guiding device they are assembled on. During the tests, it was found that the elastomeric elastic element has significant internal friction (hysteresis losses). As a result of the tests, the load and discharge curves of the elastic element were determined and represented graphically (figure 2).

![Figure 1. Scheme of the elastomeric elastic element installation into the car suspension](image1)

![Figure 2. The elastic characteristic of the elastomeric element](image2)

A discrete set of experimentally obtained points is approximated by continuous curves using the least mean squares method. When testing elastic elements with internal friction, the elastic characteristic (the dependence of the elastic force $F$ from the deformation value of the elastomeric element $f$) can be determined as the dependence of the average value of the force $F_{cp}$ from the elastic element deformation $f$. 

The elastic characteristic of an elastomeric element is nonlinear, so its stiffness can be determined only at a point or for a limited area considered linear. The average value of the stiffness at the point $C_{cp}i$ of the nonlinear elastic characteristic can be determined in accordance with the expression.

$$C_{cp}i = \frac{F_{cp}i}{f_i}$$ (1)

where: $C_{cp}i$ is the stiffness value at point $i$ of the nonlinear elastic characteristic; $F_{cp}i$, $f_i$ are the average value of the force and the value of the deformation at point $i$ of the nonlinear elastic characteristic, respectively.

$$F_{cp}i = \frac{F_{p1}i + F_{p1+1}i}{2}$$ (2)

where: $F_{p1}i$, $F_{p1+1}i$ are the value of the force at point $i$ in the loading and unloading section of the elastic characteristic, respectively (Fig. 2).

The average value of stiffness for a limited section of the nonlinear elastic characteristic is determined in accordance with the expression

$$C_{np}i = \frac{F_{i+1}-F_i}{f_{i+1}-f_i}$$ (3)

where: $C_{np}i$ is the reduced stiffness on the $i$-th section of the elastic characteristic; $F_i$, $F_{i+1}$ are the previous and the subsequent values of the vertical force on the elastic characteristic acting on the elastomeric elastic element; $f_i$, $f_{i+1}$ are the previous and the subsequent values of the reduced vertical deformation of the elastic element.

The elastic characteristic of the elastomeric element provides the scheduling of the suspension elastic characteristic, followed by the calculation of the reduced values of stiffness and damping coefficient.

### 3. Experimental determination of the damping coefficient

Since the elastomeric element design eliminates the friction of the sleeve material against the outer and inner guiders, the only source of friction can be the friction caused by the loss of energy in the sleeve material during their deformation. To solve the problem of the amount of friction and its sufficiency for oscillation suppression in the car suspension when driving, tests were carried out to assess the elastomeric elastic element damping properties.

To determine the damping coefficient of an elastomeric elastic element we use method based on measuring the parameters of the system vertical damped vibrations.

The scheme of the bench-testing unit for determining the damping coefficient is shown in figure 3.

![Figure 3. Bench-testing unit for determining the damping ratio](image-url)
The elastomer package 1 is attached to the pendulum copra 2, which is connected to the rack 3 by a hinge. The rack 3 is fixed rigidly on the foundation. To provide a load on the elastomer, a load 4 is attached to the pendulum 2, the mass "M" of which must correspond to the reduced load acting on the elastomer package. To create damped vibrations, the pendulum of the raised is raised to a small height and released using a "quick reset" device. Vibrations are recorded with the help of a displacement sensor 5, 6, which registers a damped process by changing the coordinate "\(z\)", movement in the vertical direction. The signal from the sensor is processed by the ADC 7 and recorded using a personal computer 8. Figure 4 shows the result of the experiment, which is the process of damping vibrations in an elastomeric element.

![Figure 4. The process of damped vibrations in an elastomeric elastic element](image)

The internal friction of the elastomer affects the damping properties of the suspension in a certain way. It is difficult to account how internal elastomer friction affects the nature of dynamic suspension processes because the type of friction is not defined by clear physical boundaries. In fact, it can be defined as a "dry" one only by convention, since it does not remain constant, and its value depends on the force acting on the elastic element. Judging from the damping vibrations process, figure 4, it can be assumed that the type of friction in an elastomeric elastic element is a viscous one by its physical nature. Calculating the damping ratio at the present stage of research we assume that the nonlinear nature of the amplitude envelope of a damped nonlinear process is stipulated by nonlinear internal friction, and the elastomer damping coefficient corresponds conventionally to the viscous friction type.

The conditional damping coefficient is calculated in the following order. The damping decrement is:

\[
d = \frac{A_1}{A_{i+1}}
\]  

where: \(d\) is the damping decrement; \(A_1, A_{i+1}\) are the two consecutive values of the oscillation amplitudes.

The vibration damping coefficient is:

\[
h = \frac{\ln d}{T}
\]

where: \(h\) is the vibration damping coefficient, \([\text{sec}^{-1}]\); \(T\) is the oscillation period, \([\text{sec}]\); \(\ln d\) is the logarithmic decrement of the attenuation.

The conditional damping coefficient of the elastomeric element is:

\[
k_s = 2 * h * M
\]

where: \(M\) is the weight of the load reduced to the elastic element axis.

The results of the calculations for determining the damping coefficient of an elastomeric elastic element are summarized in Table 1.
Table 1. Test results of an elastomeric elastic element on a pendulum copra

| Parameter                                                                 | Measure units | Values  |
|---------------------------------------------------------------------------|---------------|---------|
| Load mass corresponding to the reduced load                               | $M$, kg       | 988     |
| Average value of logarithmic decrement of the attenuation according to the results of all tests | $lnd$         | 0.588   |
| The average value of the period of damped oscillations according to the results of all tests | $T$, s        | 0.73    |
| The conditional damping coefficient                                       | $k$, N*s/m    | 1592    |

In cases when acting friction forces [2] need to be taken into account while studying the suspension dynamic properties, they can be defined as the average value of the friction force $F_{tr}$ at the point of the elastic characteristic or for its limited area. Since the elastic characteristic is nonlinear, the friction force can be taken into account as the elastomeric element deformation function $f_1$, Figure 2. In some cases, in order to simplify the analysis of the suspension dynamic properties, it is possible to average the friction force within the most probable value of the elastic element deformation observed in the certain driving modes.

3. Determination of suspension parameters

For most design maps of cushioning systems the stiffness (and the suspension damping coefficient) reduced to the "wheel" differ significantly from the stiffness of elastic elements and the resistance coefficients of damping devices [3].

To determine the suspension parameters reduced "to the wheel", special calculations need to be done. Graphic-analytical techniques are mostly used ones. Figure 5. shows necessary graphical layout for a suspension with a guiding device on wishbones. Figure 5a presents the kinematic scheme of the suspension considered. Figures 5b and 5b show the forces acting on the steering knuckle with the wheel, figure 5g shows forces acting on the lower suspension arm.

![Figure 5](image)

**Figure 5.** Graphic-analytical technique for determining suspension parameters reduced to the wheel: the suspension on wishbones scheme, b – equilibrium conditions for steering knuckle, c – triangle of forces construction for steering knuckle, d – equilibrium conditions for the lower arm
The forces acting on the suspension parts ensure their static balance. The values of the operating forces are determined for a fully loaded vehicle.

To make upper suspension arm equilibrions, the force $\vec{F}_B$ arising in the hinge $B$ must be directed along this lever. The upper lever is connected to the car frame and the steering knuckle with the help of hinges. Therefore, the force acting on the upper lever and the reaction acting on the steering knuckle are directed along the axis of the lever. The forces acting on the lower lever are shown in Fig. 5d. Since the elastic element is hinged at both ends, the force $\vec{F}_{np}$ will be directed along the axis of the elastic element. The normal reaction $R_z$ is applied at the point of contact of the wheel with the support surface and is directed vertically upwards. Since no moments are applied to the wheel and the steering knuckle, the steering knuckle equilibrium is ensured by the following condition: the sum of the vectors acting on the steering knuckle must be zero:

$$\vec{F}_b + \vec{F}_c + \vec{R}_z = 0$$  \hspace{1cm} (7)

The suspension elastic characteristic is the dependence of the normal reaction value $R_z$ from the suspension "deflection" $f_n$, which is actually defined as the wheel (its axis) shift with regard to the car frame, when the elastic element is deformed.

To construct the suspension elastic characteristic it is necessary to define the dependence $R_z = F(f_n)$, that is, to determine the values of the normal force $R_z$ at different values of the elastic element deformation. The graphic approach is considered to be the simplest engineering solution. The method involves the use of the suspension kinematic scheme. In the scheme, several ($6 - 8$) positions of the suspension levers are drawn with different elastic element deformations, larger and smaller static loads. The elastic force value $F_{np}$ is determined for each position of the levers.

The value of the normal reaction $R_z$ is determined for each position of the levers. To do this, one needs to draw a diagram of the forces acting on the steering knuckle. At the equilibrium state of the steering knuckle, the vector sum of all the forces acting on it must be zero, which is graphically displayed as a closed polygon of forces. The directions of the force applied to the upper lever and the normal reaction are known. The direction of the force from the lower lever is determined by a well-known rule. The line that continues the force vector $F_c$ must pass through the intersection of the action lines of the forces $F_b$ and $R_z$, Figure 5a.

The value of the normal reaction for each position of the suspension arms is determined from the force triangle. When constructing a force triangle, the directions of all forces acting on the steering knuckle are known. The line of action of the force $F_c$ is drawn in the direction specified by the suspension kinematic scheme. The force vector $F_c$ must be protracted in accordance with the scale of forces selected. From the end of the vector, a vertical action line of the normal reaction $R_z$ is drawn, and from the beginning of the vector, the line of action for the force $F_b$ is drawn. The intersection of the constructed lines determines the value of the normal reaction $R_z$ for the considered position of the suspension arms, taking the selected scale into account. This is the way to determine the normal reaction value for each of the levers constructed positions and to construct a suspension elastic characteristic which determines the stiffness values reduced to the wheel.

The damping coefficient value reduced to the wheel is determined from the following conditions, figure 6.

The moment of the damping force in relation to the lower lever axis of rotation, created by the elastomer is equal to:

$$T_{k1} = k_1 \cdot V_1 \cdot l_1$$ \hspace{1cm} (8)

where: $k_1$ is the damping coefficient of the elastomer; $V_1$ is the elastomer deformation rate (displacement of p. $O_1$); $l_1$ is the action moment arm of the damping resistance force.

The moment of the damping force reduced to the wheel is equal to:

$$T_{k2} = k_2 \cdot V_2 \cdot l_2$$ \hspace{1cm} (9)

where: $k_2$ is the damping coefficient reduced to the "wheel"; $V_2$ is travel speed of p. $O_2$; $l_2$ is the action moment arm of the damping resistance force.
Figure 6. Determining of the damping ratio

It is obvious that $v_2 = \frac{v_1 l_2}{l_1}$. Reasoning from the equality of moments and the velocity ratio we get the formula of the damping ratio reduced to the "wheel":

$$k_2 = \frac{k_1 l_1^2 l_2}{l_2^2}$$

(10)

At the preliminary stage when choosing main design parameters, the natural oscillation frequency of the vehicle sprung mass can be set as the initial value, so the resulting value of the reduced suspension stiffness can be assessed taking the recommended oscillation frequency of the sprung masses into account.

The natural oscillation frequency of the car sprung masses $\omega_0$ with design static load is used in customary calculations for preselecting suspension stiffness.

Static (conditional) deflection and oscillation frequency are related by the dependence[4]:

$$\omega_0^2 = \frac{g}{f_{ct}}$$

(11)

where: $\omega_0$ is the natural oscillation frequency of the suspension sprung masses, [s$^{-1}$]; $f_{ct}$ is the suspension static deflection of the suspension, [m].

Depending on the type of car, natural oscillation frequency can vary from 1 to 2.5 Hz [4-6]. A lower frequency value provides higher quality of the ride. At the stage of preliminary calculations for different kinds of vehicles the natural oscillation frequency at rated static load can be taken equal to: for passenger cars $\omega_0 = (6...8,5)$ s$^{-1}$; for buses $\omega_0 = (8...11)$ s$^{-1}$; for trucks $\omega_0 = (9...15)$ s$^{-1}$. According to the results of the experiment the natural oscillation frequency of the sprung masses for the considered suspension structure with elastomeric elements made $\omega_0 = 9,5$ c$^{-1}$.

Choosing the value of dynamic deflection, it is necessary to consider that suspension must be energy-intensive enough to exclude "breakdowns", parts of the suspension guider or axles impacting on the structural elements of the body or frame. The energy intensity (capacity) of the suspension is understood as the deformation work of the suspension elastic element from the fully unloaded state to the maximum possible value corresponding to the suspension breakdown. In general, the deformation work is equal to [5]:

$$U = \int_0^{f_{max}} \Phi(f) df$$

(12)

where: $U$ is the the suspension energy intensity; [kN*m]; $\Phi(f)$ is the function describing the curve of the the suspension elastic characteristic.
In numerical terms, the deformation work is determined by the area under the elastic characteristic curve (see Figure 2).

The suspension capacity is limited by the design and the maximum stresses allowed by the elastic element material. Bearing this in mind, one should choose the maximum possible suspension dynamic deflection, being guided by the suspension capacity of the prototype cars and the maximum value of stresses in the suspension elastic element.

In addition, the value of the dynamic deflection at the preliminary design stage can be estimated using the specific dynamic energy intensity \( U_{уд} \) [3]:

\[
U_{уд} = \frac{U_d}{F_{ст}}
\]

where: \( U_d \) is the suspension dynamic capacity, \( U_d = U - U_{ст} \); \( U_{ст} \) is static energy intensity of the suspension; \( F_{ст} \) – static load.

4. Results and conclusions

It is empirically supported that in order to minimize the probability of suspension breakdown when driving on all types of roads, the value of the specific dynamic energy intensity, which depends on the car type, should vary within: for passenger cars and buses \( U_{уд} = (0.04...0.06) \) kNm/kN; for trucks \( U_{уд} = (0.05...0.07) \) kNm/kN; for all-terrain vehicles \( U_{уд} = (0.06...0.08) \) kNm/kN. For the considered suspension structure with elastomeric elements, the value of the specific dynamic energy intensity is \( U_{уд} = 0.068 \) kNm/kN.

As a result of the study, the elastic characteristic of the elastomeric elements package was determined and the material internal friction was evaluated. To predict the ability of elastomers in the car suspension to absorb oscillation energy, we presented the technique describing the attenuation process using a "conditional" damping coefficient. We gave an outline of methods for reducing the suspension parameters to the "wheel", calculating the natural oscillation frequency of the sprung masses and determining the suspension energy intensity. Analysis of the natural oscillation frequency values and the specific energy intensity of the elastomeric suspension allows us to conclude that using the same sort of construction in the suspensions of trucks, buses and all-terrain vehicles is advisable.

Acknowledgment

The research was carried out with financial support from the Ministry of Education and Science as part of the project "Creating high-tech production range of vehicles GAZelle Next with the new electronic architecture of electronic systems", under Agreement No. 075-11-2019-027 from 29.11.2019 (decree No. 218 of the Government of the Russian Federation dated April 09, 2010).

The experimental studies were carried out using the equipment of the NSTU Sharing Center "Transport systems".

References

[1] Kuleznev V N and Shershnev V A 2007 Chemistry and Physics of Polymers M. Colossus p 367
[2] Khachaturov A A 1976 Dynamics of the road-tire-car-driver Mechanical Engineering p 535
[3] Rotenberg R V 1972 Car suspension Mechanical Engineering p 392
[4] Kravets V N 2013 Theory of the car Nizhny Novgorod. state Tech. University n.a. R.E. Alekseeva p 413
[5] Yatsenko N N and Prutchikov O K 1968 Smooth running of trucks Mechanical Engineering p 220
[6] Wong D 1982 Theory of land vehicles Mechanical Engineering p 284