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Energy intensity of parts made from polyurethane elastomers

Энергоемкость деталей из полиуретановых эластомеров

V.G. Artiukh,
E.A. Galikhanova*,
Peter the Great St. Petersburg Polytechnic University, St. Petersburg, Russia
V.M. Mazur,
LLC 'Saint-Petersburg Electrotechnical Company', Saint-Petersburg, Pushkin, Russia
S.B. Kargin,
Pryazovskyi State Technical University, Mariupol, Ukraina

Д-р техн. наук, профессор В.Г. Артюх,
студент Э.А. Галиханова*,
Санкт-Петербургский политехнический университет Петра Великого, г. Санкт-Петербург, Россия
главный специалист технического управления В.О. Мазур,
ООО «Научно-производственное объединение «Санкт-Петербургская Электротехническая Компания», г. Санкт-Петербург, г. Пушкин, Россия,
канд. техн. наук, доцент С.Б. Каргин,
Приазовский государственный технический университет, г. Мариуполь, Украина

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Abstract. Laboratory testing with determination of strength and energy characteristics of materials for elastic elements of shock absorbers helped to establish that polyurethane elastomers, having the highest values of specific energy consumption and a wide range of alternating dissipation and rigidity characteristics were the best materials. It will allow to create a series of efficient shock absorbers for different elements and units of building constructions. As the result of the analysis performed of different types of steel springs a coefficient of spring's quality in volume was obtained, as well as coefficient of spring's quality in weight, coefficients of shape for cylindrical, crew-type and plate springs. These coefficients can help to arrive at the conclusion, regarding the appropriateness of this spring for a particular elements and units of building constructions.

Аннотация. Лабораторные испытания с определением прочностных и энергетических характеристик материалов для упругих элементов амортизаторов позволили установить, что лучшими материалами являются полиуретановые эластомеры, обладающие наибольшими значениями удельной энергоемкости и широким диапазоном изменения диссипативных и жесткостных характеристик. Это позволит создать ряд эффективных амортизаторов для различных элементов и узлов строительных конструкций. В результате проведенного анализа различных типов стальных пружин получены коэффициент качества пружины по объему, коэффициент качества пружины по весу, коэффициенты формы для цилиндрических винтовых и тарельчатых пружин. На основании этих коэффициентов можно сделать заключение о пригодности данной пружины для конкретных элементов и узлов строительных конструкций.

1. Introduction

The use of polymeric materials, including polyurethane elastomers, is continuously increasing in different fields of science such as engineering, transport and construction. It could be explained by the unique qualities of elastomers in terms of their strength, elasticity, energy intensity and other mechanical characteristics. The most upcoming elastomers are injection molded structural polyurethanes, the use of which in construction is expanding very fast.

The energy intensity of elements and units of building structures is the most important characteristic that affects the ability to protect elements from dynamic effects, for example, seismic. The overwhelming majority of such elements have insufficient energy intensity and excessive rigidity.

In metallurgical machinery high levels of parasitic (not technological) loads are often observed. Good example of such drives are drives of continuous wide strip mill (WSM) 1700, where dynamic loads during strip bite by work rolls are in 3...4 times exceed technological loads at steady rolling. One of the most

effective solutions to reduce dynamic loads is significant increase of energy intensity of main drive lines; the most real way of energy intensity increase is installation of energy-intensive (active) parts [1] in drive.

Maximum possible energy intensity, with loads not exceeding standards, able to significantly increase the energy intensity of the entire drive, characterizes this kind of detail.

General issues of machines loading and arising overloads are considered in [1–6]. Elastomer-metallic dampers and their elastic elements began to be scrutinized long time ago, and these researches are ongoing so far [7, 8]. The selection of polymeric (elastomeric) materials for elastic elements of active devices, study their mechanical characteristics are shown in articles [9–12]. Different types of protective devices for metallurgical machines described in [13, 14]. Mechanical and temperature properties of polyurethane elastomers are described in articles [15–21].

It is possible to achieve the maximum energy intensity of detail (assembly, unit) modernizing it in three ways:

1. Selection of detail material with the highest specific energy intensity;
2. Ensuring uniform distribution of stresses in detail;
3. Provision of sufficient size (volume, mass) of this detail.

Below these options of energy intensity increase are considered and the greatest interest is active parts that can affect values of parasitic loads.

2. Methods

Modern machinery is based on numerous classes of materials having different (often substantially different) mechanical characteristics. These are steel, non-ferrous metals and their alloys, plastics, ceramics, elastomers and so on. These materials vary in strength by tenfold and in rigidity by hundreds or thousands times. Common for all construction materials is a lack of information about their energy intensity in any literature. It shows that question of energy intensity of shock absorbers not found not only the proper solution, but also the relevant description. It needs to start this task with selection of material.

Let us consider any simple detail (beam), where there is a simple stress state (such as uniaxial stretching or compression). Specific (per volume unit) energy intensity can be represented as

$$u_v = \frac{U}{V} = \frac{\sigma^* \cdot \varepsilon^*}{2}, \tag{1}$$

where U is energy intensity of detail;

V is volume of detail;

σ^* is stress limit value;

ε^* is deformation limit value.

Keeping in mind that

$$\sigma^* = E \cdot \varepsilon^* \text{ and } \varepsilon^* = \frac{\sigma^*}{E}, \tag{2}$$

Finally

$$u_v = \alpha \frac{\left(\sigma^*\right)^2}{E}, \tag{3}$$

where α is a numeric factor depending on kind of stress condition (quality factor of stress condition). For uniaxial stretching and compression α = 0.5 (i.e., for such loading which provides uniform stress distribution throughout the volume of the elastic element [2]).

Stress limit value can be chosen depending on purpose of calculated detail. For fragile metal materials it is value close to strength limit σ_B, for ductile metal materials it is yield strength σ_t, for high-strength metal materials it is conditional yield strength $\sigma_{0.2}$. Comparative analysis of different materials on energy intensity can be performed by formula (3) with the same coefficient α, for example, taking α = 0.5. Thus we compare samples of various materials but in the same tense condition.

In the vast majority of real structures of shock absorbers as materials which accumulate energy special spring steel grades are used. These are (according to their description in CIS countries) carbon constructional steel grades 65G, 70G, siliceous alloy spring steel grades 60C2, 70C2, 60C2HFA and other. Listed steel grades are characterized by high strength limit values (after thermal treatment) σ_B = 1100...1900 MPa and conditional yield strength $\sigma_{0.2}$ = 700...1500 MPa. For widely spread spring steel grades having $\sigma_{0.2} \approx$ 1200MPa, we get taking into account $\sigma^* = \sigma_{0.2}$

$$u_v = 0.5\frac{1200^2}{2\cdot 10^5} = 3.6\frac{MJ}{m^3} = 3.6MPa\,. \tag{4}$$

For decades, steel was practically the only material for springs and other energy-intensive items. Comparing energy intensity of different steel grades actually we consider only their strength because modulus of elasticity of different steel grades are insensitive structural characteristics, i.e. practically permanent. For all low-alloyed steel grades normal modulus of elasticity value may be taken as $E = 2.0\cdot 10^5$ MPa. Therefore, the improvement of springs and other energy-intensive steel elements went in a way only to increase their strength. This process led to undoubted success. Best spring steels are superior to ordinary low-carbon steels (which are the most widely spread) on strength by 4...5 times and on energy intensity by 20 times, that is a major achievement of specialists working with metal.

However, this way of specific energy intensity improve is not the only one. Apart from steel and considering broader class of materials it needs to take into stiffness of material into account (in addition to strength) stated by normal elasticity module. It is better to search materials with high strength and low stiffness. Such materials can be found among polymers and elastomers in particular that relate to low-modulus materials. Energy-intensive materials can be found in groups of polyamides, lavsanov, ultra-high-molecular polyethylene as well as urethane rubbers-polyurethanes.

It should be noted that specific energy intensity is absent in the list of mechanical characteristics of these materials. So it can be concluded that the listed materials (unlike steels) on “energy intensity” parameter were not selected and not improved. In this regard, low-modulus materials have great chances for meaningful improvements [10]. As an example, we consider such material as molding structural polyurethane CKU-PFL-100 (description in CIS countries) which has normal compressive modulus E_c = 60 MPa. This material is quite widely spread and produced by domestic industry in CIS countries. There are foreign analogues of this material (for example, adiprene L 167).

For polyurethane elastic elements working on compression (with single loading that is typical for buffer devices), it is allowable to consider elastic deformation equal to 20...35 % [11]. Allowable deformation with one-time loading can be found in experiments on samples (e.g., cylindrical samples under their compression). Measuring dimensions of sample before and after loading it is possible to find deformation when after its removal initial sample size fully recovers. For low-hardness polyurethane elastomers (E_c = 10…30 MPa) this value is ε^* = 0.35, for medium-hardness polyurethane elastomers (E_c = 35…60 MPa) ε^* = 0.30 and so on. Let us take the maximum allowable stress σ^* = 0.3E, then specific energy intensity:

$$u_v = \frac{(0.30E_c)^2}{2E} = \frac{0.09E_c}{2} = 2.7MPa; \tag{5}$$

It is clear from shown figures that energy intensity of given polyurethane corresponds to energy intensity of special spring steel grades. It is about specific energy intensity per unit volume. Meanwhile, there are objects in machinery engineering practice (mainly vehicles), for which a very important parameter is their own weight. In this case, to characterize suitability of material it is better to use specific energy intensity per unit weight

$$u_p = \frac{u_v}{\gamma_м} = \alpha\frac{\left(\sigma^*\right)^2}{E\cdot\gamma_м}, \tag{6}$$

where γ_m is specific weight of elastic element material, MN/m³.

For material considered above:

- steel, $\gamma_s = 78\cdot 10^{-3}$ MN/m³;
- polyurethane, $\gamma_p = 11\cdot 10^{-3}$ MN/m³.

Respectively, values of specific energy intensity are:

- for spring steel (σ^*= 1200 MPa; E = 2·10^5 MPa); $u_p = \frac{3.6}{78 \cdot 10^{-3}} = 46m$;
- polyurethane CKU-PFL -100 (σ^*= 0.35E; E = 60 MPa), $u_p = \frac{2.7}{11 \cdot 10^{-3}} = 245m$.

3. Results and Discussion

Figures convincingly show that according to u_p elastomers have a clear advantage over special spring steel grades [11]. There are values of energy intensity of some common plastics below in Table 1. From Table 1 it is clear that all these materials have high energy intensity, therefore, can be effectively used in machines with significant levels of parasitic loads.

Table 1. Energy intensity of elastic elements materials

Elastic element material (description as per CIS countries)	Normal elastic modulus (E), MPa	Maximum (limit) deformation ε*, %	Limit stress σ*, MPa	Specific weight of material γ_M, kN/m^3	Specific intensity of material under compression	
					per unit volume u_v, MPa	per unit weight, u_p, m
Spring steel 65G	2·10^5	0.45	900	78	2.03	26.0
Spring steel 60C2	2·10^5	0.60	1200	78	3.60	46.2
Spring steel 60C2HFA	2·10^5	0.80	1600	78	6.40	82.0
Rubber B-14	14	35	4.90	13	0.855	66.0
Polyurethane CKU-7L	20	35	7.0	11	1.21	110
Adiprene L 100	30	32	9.60	11	1.53	126.0
Polyurethane CKU-PFL-100	60	30	18.0	11	2.70	245
Polyethylene CBMPE	300	10	30.0	9.5	1.50	158

Note. The best options are highlighted.

From Table 1 it is clear that the highest energy intensity (color highlighted) per unit of weight have low-modulus materials: molded structural polyurethanes and polyethylene CBMPE. The most strength spring steel is slightly better compared to rubber B-14 in terms of specific energy intensity. At the same time, high-strength spring steel grades still are not better compared to elastomers in terms of energy intensity per unit volume. Question about choosing particular material for engineered shock absorber will be resolved below, where in addition to specific energy intensity of material other significant factors would be considered [14].

Energy intensity is maximum energy of elastic deformation accumulated by detail with this form of loading. The same item can show varying energy intensities in different loading variants depending on load stress state arising in this detail. Specific energy intensity of different materials on the example of uniaxial stress state with a uniform distribution of stresses was considered above.

For arbitrary stress state specific (per unit volume) energy intensity is

$$u_v = \frac{1}{2}\left(\sigma_1\varepsilon_1 + \sigma_2\varepsilon_2 + \sigma_3\varepsilon_3\right), \tag{7}$$

where σ_1, σ_2, σ_3 are principal stresses;

ε_1, ε_2, ε_3 are principal deformations.

When use the generalized Hooke's law and exclude from formula (7) deformations:

$$u_v = \frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\mu\left(\sigma_1 \cdot \sigma_2 + \sigma_1 \cdot \sigma_3 + \sigma_2 \cdot \sigma_3\right)\right]. \quad (8)$$

From formula (8) it is possible to get special cases for simple stress states: stretch-compression, torsion, bending and other.

Total energy intensity of detail can be found by summing formula (8) throughout volume of the detail. To do this it needs to know all stress states of the detail of analytical expressions for the main stresses depending on detail coordinates of points:

$$U = \int_v u_v \cdot dV\,; \quad (9)$$

$$\left.\begin{aligned} \sigma_1 &= f_1(x, y, z) \\ \sigma_2 &= f_2(x, y, z) \\ \sigma_3 &= f_3(x, y, z) \end{aligned}\right\}. \quad (10)$$

Difficulties in practical application of formula (9) is that for the majority of details (except those that operate at very simple stress conditions) it is difficult to get functions (10) with sufficient accuracy and in a form convenient for integration.

Quite simply result can be achieved when it comes to uniaxial tensile-compression, when it comes to torsion and bend it is more complicated. For such energy-intensive items as disk springs calculation of energy intensity is strongly difficult, for objects of type similar to slotted springs it becomes virtually impossible (if not take numerical methods into consideration that can be used for making special cases).

At the same time, energy intensity of any details or units can be found experimentally or approximately by values of limit loads and deformations. If working characteristic of researched detail (in coordinates “force P and draught λ”) is available and it is for all λ and ensure the absence of residual deformations then target energy intensity represents area of working characteristic limited by values $0 \le \lambda \le \lambda^*$, where λ^* is maximum draught, causing arise of plastic deformation in detail.

$$U = \int_0^{\lambda^*} P(\lambda) \cdot d\lambda\,, \quad (11)$$

where $P(\lambda)$ is a variable force value.

In most cases, expression $P(\lambda)$ is easier than expression (10) and integral (11) is easier to calculate; function $P(\lambda)$ may be approximate.

Finally, energy intensity of detail can be found on the basis of internal loading factors and elastic movements. For linearly deformable systems (consisting of rods) energy intensity of detail can be found by formula

$$U = P^* \cdot \frac{\lambda^*}{2}, \quad (12)$$

Where P^* is force appropriating loading when $\sigma_{max} = \sigma^*$;

λ^* is limit value of generalized displacement in direction of force P^*.

So, for elastic element constituting the beam on two supports and under loading by force in the middle by equation (11) it is

$$U = \frac{\left(P^*\right)^2 \cdot l^3}{96EI_x}. \quad (13)$$

It would be interesting to compare energy intensity of the same detail under different loading. Let the detail has simple form that is a cylinder with diameter d and length l, and $l >> d$, so our detail is beam which can be calculated by methods of strength of materials.

This beam can be stretched, compressed, bent in different ways (such as doubly or cantilever beam) and tighten. It is possible to produce cylindrical coil spring from long beam, and so on. The energy intensity of these beams can be represented as

$$U = \alpha \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot V. \tag{14}$$

More value of α, bigger energy intensity of the beam, more it is suitable for manufacture of shock absorber. Let us start with tension-compression (meaning a case of compression, when there is no loss of stability)

$$U = \frac{1}{2} P^* \cdot \Delta l^* = \frac{1}{2} P^* \cdot \frac{P^* \cdot l}{EF}, \tag{15}$$

where Δl^* is maximum allowable deformation at which there is no permanent deformation.

Here $P^* = \sigma^* \cdot F$. Accordingly,

$$U = \frac{1}{2} \sigma^* \cdot F \cdot \frac{\sigma^* \cdot F \cdot l}{EF} = \frac{1}{2} \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot V. \tag{16}$$

Thus, in this case α = 0.5, which was already mentioned above. Note that stresses in the above example are considered uniformly distributed throughout the volume of the cylinder.

Further, bending of the beam (as beam on two pivot supports) by concentrated force P applied to the middle is considered. In this case

$$U = \frac{1}{2} P \cdot f = \frac{1}{2} P^* \cdot \frac{P^* \cdot l^3}{48 E I_x}, \tag{17}$$

where f is deflection of the beam.

Here P^* has to be found considering conditions of strength at bending.

$$\sigma^* = \frac{P^* \cdot l}{4 W_x}, \tag{18}$$

where W_x is axial section modulus of the beam. At the same time

$$P^* = 4 W_x \cdot \frac{\sigma^*}{l} \cdot$$

Substituting it into formula (18)

$$U = \frac{1}{2} \cdot 4 W_x \cdot \frac{\sigma^*}{l} \cdot 4 W_x \cdot \frac{\sigma^*}{l} \cdot \frac{l^3}{48 E I_x} = \frac{1}{6} \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot \frac{W_x^2 \cdot l}{I_x} \cdot \tag{19}$$

Factor $\frac{W_x^2 \cdot l}{I_x}$ has dimension m³, it is proportional to volume. In other words, volume of the detail can be isolated from expression (19). Thus, for the cylinder

$$W_x = \frac{\pi d^3}{32}; \; I_x = \frac{\pi d^4}{64}. \tag{20}$$

Then $\frac{W_x^2 \cdot l}{I_x} = \frac{\pi d^3 \cdot \pi d^3 \cdot 64}{32 \cdot 32 \cdot \pi d^4} \cdot l = \frac{\pi d^2 \cdot l}{4 \cdot 4} = \frac{1}{4} V$. Formula (19) begins to be in a form

$$U = \frac{1}{24} \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot V \cdot \tag{21}$$

From formulas (16) and (21) it is seen that energy intensity of the same detail (the cylinder) decreased by 12 times at the transition from stretching to bending. If consider this detail as a shaft its energy intensity is equal to

$$U = \frac{1}{2} M^* \cdot \varphi^*, \tag{22}$$

where M^* is torque corresponding to maximum stress $\tau_{max} = \tau^*$. This value is related to the equivalent stress σ^* that is a strength criteria (e.g., energy)

$$\sigma^* = \sqrt{3} \cdot \tau^*, \tag{23}$$

φ^* − the angle of shaft twist which corresponds to torque M^*. This angle can be found by the formula

$$\varphi^* = \frac{M^* \cdot l}{GI_p}, \tag{24}$$

where G is shear modulus of the shaft material

$$G = \frac{E}{2(1+\mu)}, \tag{25}$$

where μ is Poisson coefficient of the material;

I_p is polar moment of inertia of the shaft cross-section.

$$I_p = \frac{\pi d^4}{32} \cdot \tag{26}$$

Considering (25) and (26) it comes from (24

$$U = \frac{1}{2} M^* \cdot \frac{M^* \cdot l}{GI_p}. \tag{27}$$

Here

$$M^* = \tau^* \cdot W_p. \tag{28}$$

W_p is polar section modulus

$$W_p = \frac{\pi d^3}{16} \cdot \tag{29}$$

Putting (28) and (29) into (27) it comes to

$$U = \frac{1}{2} \tau^* \cdot W_p \cdot \frac{\tau^* \cdot W_p \cdot l}{GI_p} \cdot \tag{30}$$

If put in expression (30) values τ^* and G, taken from (23) and (25) it comes to

$$U = \frac{1}{2} \cdot \frac{\sigma^*}{\sqrt{3}} \cdot \frac{\pi d^3}{16} \cdot \frac{\sigma^* \cdot \pi d^3 \cdot l \cdot 32 \cdot 2(1+\mu)}{\sqrt{3} \cdot 16 \cdot \pi d^4 \cdot E} = 0.208 \frac{\left(\sigma^*\right)^2}{E} \cdot V \cdot \tag{31}$$

This result is 2.4 times less than in case of tension or compression; but it is significantly higher than for case of bending. However, it should be noted that considered cross section (circle) is advantageous (optimal) in case of torsion and disadvantageous in case of bending.

If cross-section of square is taken, then as per formula (16) for case of uniaxial stretching or compression it comes to obtaining of α = 0.5 (as well as for other forms of cross-sections). For case of bending from formula (29) it comes to

$$U = \frac{1}{6} \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot \frac{\left(b^3\right)^2 \cdot l \cdot 12}{6^2 \cdot b^4} = \frac{1}{18} \cdot \frac{\left(\sigma^*\right)^2}{E}. \tag{32}$$

Here, quality factor of the stress state $\alpha = \frac{1}{18}$ compared with $\alpha = \frac{1}{24}$ of the circle. For case of torsion of shaft with square cross-section let us take formula (30) wherein W_p should be replaced by W_k and I_p should be replaced by I_K respectively, wherein:

$$W_к = 0.208 \cdot a^3;$$

$$I_к = 0.141 \cdot a^4.$$

Then

$$U = \frac{1}{2} \cdot \frac{\sigma^*}{\sqrt{3}} \cdot 0.208 \cdot a^3 \frac{\sigma^* \cdot 0.208 a^3 \cdot l \cdot 2(1+0,25)}{\sqrt{3} \cdot E \cdot 0.141 \cdot a^4} = 0.128 \frac{\left(\sigma^*\right)^2}{E} \cdot V, \quad (33)$$

that is the quality of the stress state which became almost 2 times less than one of a round shaft.

These values are listed in Table 2; wherein in all cases, quality factors of elastic elements are calculated. This refers not only to the quality of the stress state but also to a real opportunity to implement this stress state for particular material and particular design. Thus, for steel springs it is practically impossible to implement loading case by tensile or compressive force. Such devices would have extremely large (unacceptable for real conditions) stiffness. Practically all steel springs (cylindrical coil, disk, slotted, torsions) work in torsion or bending. Exception is ring springs working in tension and compression which, however, have significant irremovable drawbacks. These exceptions will be discussed below.

Table 2. The specific energy intensity of the elastic elements

Material of the elastic element (description as per CIS countries)	Shape and form of deformation	Coefficient of quality of the stress state α	Specific energy c intensity per unit of volume u_v, MPa	Quality coefficient of the elastic element β	Overall energy intensity u_r, MPa
Steel 65G	Twisting (torsion)	0.208	0.844	0.208	0.844
	Cylindrical coil spring	0.208	0.844	0.085	0.343
	Bending of rectangular beam	0.055	0.223	0.055	0.223
Steel 60C2	Twisting (torsion)	0.208	1.50	0.208	1.50
	Cylindrical coil spring	0.208	1.50	0.085	0.620
	Bending of rectangular beam	0.055	0.396	0.055	0.396
Steel 60C2HFA	Twisting (torsion)	0.208	2.66	0.208	2.66
	Cylindrical coil spring	0.208	2.66	0.085	1.10
	Bending of rectangular beam	0.055	0.704	0.055	0.704
Rubber B-14	Compression	0.500	0.855	0.500	0.855
Polyurethane CKU-7L	Compression	0.500	1.21	0.500	1.21
Adiprene L-100	Compression	0.500	1.53	0.500	1.53
Polyurethane CKU-PFL-100	Compression	0.500	2.70	0.500	2.70
Polyethylene CBMPE	Compression	0.500	1.50	0.500	1.50

Note. The most perspective options of elastic elements are highlighted.

Table 2 shows values of specific energy intensity related to the elastic element. For case of bending bar with rectangular cross section is taken (which gives the same results as the square bar), and in case of torsion round is optimum.

This takes into account the quality factor of the stress state, weight and dimensions of the elastic element. The latter is especially important when one element is replaced by another and available slots for their installation are preserved. In this case, it may be found so-called overall specific energy intensity

$$u_0 = \alpha \cdot \frac{\left(\sigma^*\right)^2}{E} \cdot \frac{V}{V_0}, \tag{34}$$

where V_o is overall volume of elastic element.

Relation between $\frac{V_0}{V}$ can be called a coefficient of form of elastic element, for example, of spring. Spring shape can be considered optimal if this coefficient equals one.

Further, in Table 2 quality coefficient of elastic element is shown

$$\beta = \alpha \frac{V}{V_0}, \tag{35}$$

which takes into account nature of the stress state and shape of the elastic element (spring). For cylindrical coil spring actual volume is equal to

$$V = \frac{\pi d^2}{4} \cdot \pi D \cdot n, \tag{36}$$

where D is diameter of the cylindrical coil spring;

d is diameter of the rod (bar);

n is number of coils.

Overall volume of the spring is

$$V_0 = \frac{\pi\left(D+d\right)^2}{4}\left(n \cdot d + \lambda^*\right), \tag{37}$$

where λ^* is maximum spring draught corresponding to maximum stress τ^*.

Then shape factor of the spring is

$$k_{sh} = \frac{V_0}{V} = \frac{\left(D+d\right)^2 \cdot \left(n \cdot d + \lambda\right)}{\pi \cdot D \cdot n \cdot d^2}. \tag{38}$$

According to the formula (38) value of k_{sh} can be calculated for any given spring. For example, there is a spring with the parameters: D = 128 mm; d = 32 mm; n = 6; λ^* = 40 mm. For this spring

$$k_{sh} = \frac{(128+32)^2 \cdot (6 \cdot 32 + 40)}{3.14 \cdot 128 \cdot 6 \cdot 32^2} = 2.4. \tag{39}$$

In general, for all springs of the same type it is

$$2.0 \le k_{sh} \le 6.0 \tag{40}$$

The highest values of k_{sh} correspond to low stiffness springs, lower levels correspond to high stiffness springs. As a rule, high stiffness springs are used for metallurgical equipment; in this case it can be limited by

$$2.0 \le k_{sh} \le 3.0 \tag{41}$$

Formula (38) can be simplified if value $nd+\lambda^*$ (which represents height of spring in free state) is shown in form

$$nd + \lambda^* = nd \cdot \xi, \tag{42}$$

where $\xi = \frac{H}{n \cdot d}$ is relative height of spring depending on its stiffness.

$$1.1 \le \xi \le 1.5 \tag{43}$$

Then formula (38) has form

$$k_{sh} = \frac{\left(D^2 + d^2\right)\xi}{\pi \cdot D \cdot d} \tag{44}$$

Analysis of results presented in Table 2 allows to finally evaluate material in terms of its effectiveness to produce shock absorbers. Table 2 shows that efficiency of polymer materials in particular of class of polyurethane elastomers significantly increases when quality of stress state and dimensions of elastic elements are taken into account. This advantage is certain. Even the best spring steel grades (obtained as a result of numerous researches and action focused on obtaining the highest energy intensity) yield to polyurethane elastomers for which corresponding selection has not being conducted and characteristics as energy intensity is not available in standard set of their mechanical characteristics. It is understood that upon receipt of relevant order technologists and polyurethane developers for new elastomers can significantly improve parameter such as energy intensity. Nowadays, it is believed that elastomers of polyurethane type have a great future in terms of amortization of metallurgical equipment [3, 5, 6].

Practice of last two decades of development and implementation of shock absorbers done from molded structural polyurethanes fully confirms this. Designed and manufactured in Peter the Great St. Petersburg Polytechnic University compression shock absorbers for frames of housings rollers and rollers supports of roller tables have been introduced in almost all roughing mills in Ukraine and similar technical solutions will be implemented soon for roughing mills in Russia. These dampers have elastic elements made of polyurethane type CKU-PFL, adiprene, vibrathane and others with normal compressive elastic modulus E_c = 5...500 MPa. This rigidity of the material makes it possible to use elastic element in a form of a monoblock (thick-walled cylinder) and allows it for axial compression.

All similar elastic elements have been installed to replace existing steel disk springs or cylindrical coil springs in existing slots. Thus, overall dimensions of new shock absorbers do not exceed the old ones that greatly simplifies the process of replacing. At the same time, due to significantly higher energy intensity (refer to Table 2) dampers with elastic elements made of polyurethanes provide better protection from dynamic loads and increased resource of elastic elements.

In a future, process of replacing metal springs by polyurethane elastic elements will continue to expand. However, steel springs fully will not be pushed out. This is because of many reasons. All these features refer to mechanical characteristics of elastomers from which it should be noted that use of elastomers is limited by many factors such as heat and cold resistance, internal friction, rheological effects and so on.

Therefore, steel springs for a number of objects remain. To the point, there is another issue that is the choice of optimal design of steel springs. Wherein spring material is excluded from consideration. It is necessary to analyze quality of stress state, filling out of overall dimensions and some of the technological and operational characteristics.

From Table 3 it is clear that elastic element in a form of torsion in terms of its parameters is better than cylindrical coil springs. At the same time, torsion bars are rarely used because of inconvenient form that is long round shaft which is not always possible to successfully fit into the size of protected unit. To the point, the element has greater rigidity. If twisted into a spiral these elements become with acceptable size and stiffness, but lost in optimum use of the overall volume.

Table 3. Features steel springs

Spring type	Strain type	Stress state quality coefficient α	Shape factor k_{sh}	Quality coefficient of the elastic element β
Torsion (Elastic shaft)	Torsion	0.208	1.0	0.208
Cylindrical coil	Torsion	0.208	2.0…3.0	0.083
Disk	Plane stress	0.055	1.5…1.7	0.0345
Ring	Tension and compression	0.5	4.0…6.0	0.109
Multi-sheet	Bend	0.083	1.8…2.0	0.054

Steel springs also need to be analyzed in terms of their effectiveness when working in shock absorbers. First of all, it is necessary to answer the question why on practice with a large number of constructions of steel springs in metallurgical machines are used 1...2 types of springs. Mainly these are various sizes cylindrical coil springs, sometimes these are disk springs. At the same time, torsions, slotted and ring springs are used very rarely and multi-sheet springs are used almost exclusively on some vehicles.

Table 3 shows basic characteristics of the most popular springs. Wherein it takes into account quality factor of stress state and form factors. Obtained results of the analysis are:

1. Quality factor of the elastic element (spring) by volume

$$\beta = \frac{\alpha}{k_{sh}}. \tag{45}$$

2. Quality ratio by weight of the elastic element

$$\beta_m = \frac{\alpha}{k_{sh} \cdot \gamma_m}. \tag{46}$$

On the basis of these coefficients suitability of spring for a specific machine or unit can be concluded. Coefficients of various forms of springs are calculated:

- for cylindrical coil springs (with high stiffness) in the formula (39);
- for disc springs it follows.

Actual volume of spring

$$V = \frac{\pi}{4}\left(D^2 - d^2\right)\delta_S, \tag{47}$$

where D and d are outer and inner diameters of the spring;

δ_S is thickness of spring sheet.

Overall volume of the spring (the volume of cylinder into which it fits)

$$V_0 = \frac{\pi}{4} D^2 \left(\delta_S + \lambda^*\right), \tag{48}$$

where λ^* is maximum spring draught corresponding to maximum stress σ^*. Value of λ^* for disc springs can be taken as draught *S* defined in accordance with Russian State Standard GOST (standard in CIS countries).

Then a shape factor

$$k_{sh} = \frac{D^2\left(\delta_S + \lambda^*\right)}{\left(D^2 - d^2\right)\delta_S}. \tag{49}$$

For increased rigidity of springs (which are mainly used in metallurgy) it is $1.5 \le k_{sh} \le 1.8$. Formula (49) can be simplified when put relative draught for spring

$$\eta = \frac{\delta_S + \lambda^*}{\delta_S}. \tag{50}$$

Quantity for hard springs ranges

$$1.4 \le \eta \le 1.6. \tag{51}$$

Then (49) takes the form

$$k_{sh} = \frac{D^2}{\left(D^2 - d^2\right)}\eta. \tag{52}$$

For slotted spring with dimensions:

D and d are outer and inner diameters;

$\delta = \dfrac{D-d}{2}$ – is thickness of the pipe; H is height of the spring;

ς is relative density of slots

$$\varsigma = \frac{F_h}{F_S}, \tag{53}$$

where F_h is area of holes in pipe wall; F_s is area of side of the pipe.

Usually,

$$0.1 \le \varsigma \le 0.3. \tag{54}$$

In this notation

$$V = \pi(D-d)\cdot\delta\cdot H(1-\varsigma), \tag{55}$$

and the overall volume

$$V_0 = \pi\frac{D^2}{4}\cdot H. \tag{56}$$

Thus the shape factor

$$k_{sh} = \frac{D^2}{4(D-d)\cdot\delta\cdot(1-\varsigma)}. \tag{57}$$

For actual size of slotted springs it is.

$$3.0 \le k_{sh} \le 6.0 \tag{58}$$

For a preliminary assessment it can be $k_{sh} \approx 5.0$.

4. Conclusions

1. Laboratory tests for determination of strength and power characteristics of materials for elastic elements of shock absorbers revealed that the best materials are polyurethane elastomers having the largest values of specific energy intensity and wide range of dissipative and stiffness characteristics. It makes possible to create a wide number of effective shock absorbers for different machines.

2. As a result of the analysis of various types of steel springs quality coefficient by volume of spring, coefficient of spring quality by weight, form coefficient for cylindrical coil springs and disk springs are obtained. On the basis of these coefficients conclusions about suitability of certain spring for specific machine or elements and units of building structures.

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Viktor Artiukh,
+7(931)579-70-53; artiukh@mail.ru

Elina Galikhanova,*
+7(905)004-41-99; elinlvs.g@gmail.com

Vladlen Mazur,
+7(921)369-17-60; mazur@spbec.com

Sergey Kargin,
+38(098)480-51-15; vbudar1973@gmail.com

Виктор Геннадиевич Артюх,
+7(931)579-70-53; эл. почта: artiukh@mail.ru

Элина Азатовна Галиханова,*
+7(905)004-41-99;
эл. почта: elinlvs.g@gmail.com

Владлен Олегович Мазур,
+7(921)369-17-60;
эл. почта: mazur@spbec.com

Сергей Борисович Каргин,
+38(098)480-51-15;
эл. почта: vbudar1973@gmail.com

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