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

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.

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

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

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

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

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

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

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

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:

$$W_{\kappa} = 0.208 \cdot a^3;$$

$$I_{\kappa} = 0.141 \cdot a^4.$$

Then

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

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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