UDC 678.4.06:621.8 DOI: https://doi.org/10.15407/geotm2023.166.075

SUBSTANTIATION OF PARAMETERS AND EXPERIMENTAL STUDIES OF VIBRATION ISOLATORS OF HIGH LOAD CAPACITY

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Abstract. When designing buildings and structures for operation in seismically hazardous areas, constructions of increased bearing capacity are traditionally used. At present, in Ukraine and abroad, “active seismic protection systems” have been used, which reduce seismic loads on constructions. Most of the systems belong to the means of kinematic vibration isolation. Protection against vibration seismic loading is provided by devices located between the construction (building) and the supporting construction. Vibration isolators of various designs, dampers, dynamic dampers, etc. are used as such devices.

As vibration isolators, constructions using elastomeric materials, mainly rubber, are most widely used. The use of elastomeric blocks in the systems of vibration and seismic protection of buildings and structures is characterized by high vertical rigidity, low shear rigidity, high energy dissipation, they have high reliability and the absence of sudden failure.

Despite significant advances in the design of elastomeric parts, the simplicity of shapes and extensive experience in their application, constructions with desired physical and mechanical characteristics have not yet been created. In this regard, fairly simple and accurate calculation methods that can be used at the design stage are of great importance.

The calculation of thin-layer rubber-metal vibration isolators under static compression is considered. The calculation was performed using the Ritz method for an axisymmetric problem with the assumption that axial displacements do not depend on the radius of an individual element, but are only a function of the axial coordinate. Experimental verification of the results obtained was carried out using rubber-metal vibration isolators with a diameter of 200 mm and a rubber layer height of 5 mm, 10 mm and 20 mm, which were stacked with a rubber mass of 100 mm thick.

Comparison of the calculated and experimental data shows that up to the value of the element radius to height ratio equal to ten, the stiffness values practically coincide. For thinner elements, the introduction of an appropriate coefficient is required.

Keywords: rubber-metal vibration isolator, thin-layer element, physically nonlinear medium, compressibility, elastic potential.

1. Introduction

Seismic protection is an increase in the stability of buildings and structures using special structural elements to counteract to external dynamic impact without complete destruction and with minimal human casualties.

Seismic vibration isolation systems are designed to reduce the response of building objects and protect them from earthquakes; they can also be used to protect buildings and structures from industrial vibrations, from the dynamic impact of the subway, railway and road transport, as well as from shock waves of an explosive nature.

Seismic isolation is most widely used in Japan, the USA, New Zealand, China, and Italy. Vibration isolation systems based on rubber and rubber-metal vibration isolators have become widely used in the reconstruction and construction of new buildings [1–4]. Most of the publications in these countries are devoted to methods of protecting buildings and structures using special devices [5, 6]. In recent years, the competition between such devices according to the "price-quality" index, at least for residential buildings, leans in favor of the use of rubber elements. Therefore, in the known literature [6–8], the majority of scientific works are devoted to the calculation of the dynamics of building structures with such elements under various natural and man-made influences. The calculations of rubber elements are not given enough attention, and the available publications in most cases are recommendatory [3, 5, 7, 9] or have cumbersome calculations without sufficient experimental confirmation [10, 11].

At the same time, this problem is quite actual, because the effectiveness and durability of protective systems significantly depend on the choice of parameters of protective elements, taking into account the special properties of rubber.

Elastomeric vibration isolators with high bearing capacity (VIHBC) are becoming more and more widely used in construction. They are packages of alternating layers of rubber and metal (fig. 1). Due to restricted displacements in the deformed thin rubber layers of VIHBC, there are zones with a high level of all-round compressive stresses. As a result, rubber at VIHBC exhibits physically non-linear properties.

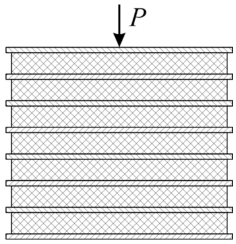

Figure 1 – Scheme of a cylindrical vibration isolator of high bearing capacity

At the moment, in practice, there are no sufficiently simple calculation methods for the deformation of VIHBC, which allow quantitatively and qualitatively to adequately describe the nonlinear effects observed in experimental studies.

The purpose of this work is to build a calculation algorithm for solving static problems of determining nonlinear mechanical characteristics of thin-layer rubber-metal elements.

2. Theoretical part

We consider one layer of rubber, which is a solid cylinder of radius R and height $2h$. Let's introduce a cylindrical coordinate system in the center of gravity of the layer (r, φ, z). With axial compression of VIHBC by force P, applied to the ends of the vibration isolator, radial u_r and axial u_z displacements of its layer are restricted by metal plates vulcanized to the ends of the layer. Considering the axisymmetric problem ($u_\varphi = 0$), we assume that the metal layers of the VIHBC are absolutely rigid and do not deform during its loading.

Using the algorithm presented in [1], let us assume that the axial displacements do not depend on the radius and are only a function of the coordinate z:

$$u_z = f(z). \tag{1}$$

Let's write the Cauchy relation for the axisymmetric problem:

$$\varepsilon_z = \frac{\partial u_z}{\partial z}; \quad \varepsilon_r = \frac{\partial u_r}{\partial r}; \quad \varepsilon_\varphi = \frac{u_r}{r}; \quad \varepsilon_{zr} = \frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r}, \tag{2}$$

where ε_z, ε_r, ε_φ – relative linear deformations in the axial, radial and circumferential directions; ε_{zr} – shear deformation; u_r – radial displacement, m.

The first invariant of the tensor of small strains has the form:

$$e = \varepsilon_z + \varepsilon_r + \varepsilon_\varphi \tag{3}$$

or, taking into account (1) and (2)

$$\frac{\partial u_r}{\partial r} + \frac{u_r}{r} + f'(z) = e.$$

This differential equation can be rewritten as

$$\frac{1}{r}\frac{\partial}{\partial r}(ru_r) = e - f'(z).$$

Integrating it taking into account the boundary condition $u_r = 0$ at $r = 0$, we get

$$u_r = \frac{1}{2}[e - f'(z)]r. \tag{4}$$

Substituting the obtained relation in (2), we come to the expressions for deformations:

$$\varepsilon_z = f'(z); \quad \varepsilon_r = \varepsilon_\varphi = \frac{1}{2}[e - f'(z)]; \quad \varepsilon_{zr} = -\frac{1}{2}f''(z)r, \tag{5}$$

The intensity of deformation is calculated using the formula:

$$e_u = \frac{\sqrt{2}}{3}\sqrt{(\varepsilon_r - \varepsilon_\varphi)^2 + (\varepsilon_\varphi - \varepsilon_z)^2 + (\varepsilon_z - \varepsilon_r)^2 + \frac{3}{2}\varepsilon_{zr}^2}, \tag{6}$$

which, taking into account (5), leads to the following:

$$e_u = \frac{1}{3}\sqrt{(e - 3f')^2 + \frac{3}{4}r^2 f''^2}. \tag{7}$$

The potential energy of the rubber element is expressed as the integral:

$$\Pi = 2\pi \int_0^R \int_{-h}^{h} Wrdrdz, \tag{8}$$

where W is the elastic potential, J/m³.

Following [2], we will adopt the model of an isotropic physically nonlinear medium, whose elastic potential has the form:

$$W = \frac{K}{k}\left\{e + \frac{1}{k-1}\left[(1+e)^{1-k} - 1\right]\right\} + \frac{3}{2}Ge_u^2\left\{1 + \frac{\mu_0}{\mu}\left\{e + \frac{1}{\mu-1}\left[(1+e)^{1-\mu} - 1\right]\right\}\right\}, \tag{9}$$

where K is the volume expansion modulus of the VIHBC material at small deformations, MPa; G – shear modulus of rubber at small deformations, MPa; μ_0, μ, k – constants characterizing the properties of a physically nonlinear material along with K and G [3].

Since for a given load e = const, it is convenient to introduce the following notations:

$$A_k = \frac{K}{k}\left\{e + \frac{1}{k-1}\left[(1+e)^{1-k} - 1\right]\right\}; \tag{10a}$$

$$A_\mu = \frac{3}{2}G\left\{1 + \frac{\mu_0}{\mu}\left\{e + \frac{1}{\mu-1}\left[(1+e)^{1-\mu} - 1\right]\right\}\right\}. \tag{10b}$$

Then the potential can be rewritten as follows:

$$W = A_k + A_\mu e_u^2. \tag{11}$$

The total energy of the system in this case is equal to:

$$U = \Pi - P\Delta, \tag{12}$$

where P is a compressive load, N; $P\Delta$ – its potential, J; $\Delta = \int_{-h}^{h} \frac{\partial u_z}{\partial z} dz = \int_{-h}^{h} f'(z) dz$ – a sag of the rubber element, m.

Considering relations (7), (10)–(12), we obtain for the total energy for this problem the expression

$$U = \int_{-h}^{h} \Phi(f', f'', z) dz, \tag{13}$$

where $\Phi(f', f'', z) = \pi R^2\left\{A_k + \frac{1}{9}A_\mu\left[e^2 - 3\left(2e + \frac{3P}{\pi R^2 A_\mu}\right)\right]f' + 9f'^2 + \frac{3}{8}R^2 f''^2\right\}$.

The condition for the minimum total energy of the system determined by the integral (13) has the form:

$$\frac{\partial \Phi}{\partial f'}-\frac{d}{dz}\frac{\partial \Phi}{\partial f''}=0, \tag{14}$$

whence we get a differential equation to determine $f(z)$

$$\frac{R^2}{4}f'''-6f'+2e+\frac{3P}{\pi R^2 A_\mu}=0. \tag{15}$$

Solving this equation, we get:

$$f'(z)=C_1 e^{\frac{2\sqrt{6}}{R}z}+C_2 e^{-\frac{2\sqrt{6}}{R}z}+\frac{1}{6}\left(2e+\frac{3P}{\pi R^2 A_\mu}\right); \tag{16}$$

$$f(z)=C_1\frac{R}{2\sqrt{6}}e^{\frac{2\sqrt{6}}{R}z}-C_2\frac{R}{2\sqrt{6}}e^{-\frac{2\sqrt{6}}{R}z}+\frac{1}{6}\left(2e+\frac{3P}{\pi R^2 A_\mu}\right)z+C_3, \tag{17}$$

where C_1, C_2, C_3 are the integration constants.

We have the following conditions for determining the integration constants:

1) symmetry condition – at $z=0 \quad u_z=0$;
2) boundary conditions – at $z=\pm h \quad u_r=0$.

Then the corresponding conditions for the function $f(z)$ are:

$$\text{at} \quad z=0 \quad f=0; \tag{18a}$$
$$\text{at} \quad z=\pm h \quad f'=e. \tag{18b}$$

From here we find

$$C_1=C_2=\left[\frac{1}{3}e-\frac{P}{4\pi R^2 A_\mu}\right]\frac{1}{\mathrm{ch}\left(\frac{2\sqrt{6}}{R}h\right)}; \quad C_3=0. \tag{19}$$

Thereby,

$$f(z)=\frac{R}{\sqrt{6}}\left[\frac{1}{3}e-\frac{P}{4\pi R^2 A_\mu}\right]\frac{\mathrm{sh}\left(\frac{2\sqrt{6}}{R}z\right)}{\mathrm{ch}\left(\frac{2\sqrt{6}}{R}h\right)}+\frac{1}{6}\left(2e+\frac{3P}{\pi R^2 A_\mu}\right)z; \tag{20}$$

$$f'(z)=\left[\frac{1}{3}e-\frac{P}{4\pi R^2 A_\mu}\right]\frac{2\,\mathrm{ch}\left(\frac{2\sqrt{6}}{R}z\right)}{\mathrm{ch}\left(\frac{2\sqrt{6}}{R}h\right)}+\frac{1}{6}\left(2e+\frac{3P}{\pi R^2 A_\mu}\right). \qquad (21)$$

The sag of the rubber element is equal

$$\Delta=\int_{-h}^{h} f'(z)\,dz=\frac{2R}{\sqrt{6}}\left[\frac{1}{3}e-\frac{P}{4\pi R^2 A_\mu}\right]\mathrm{th}\left(\frac{2\sqrt{6}}{R}h\right)+\frac{h}{3}\left(2e+\frac{3P}{\pi R^2 A_\mu}\right). \qquad (22)$$

To calculate the sag according to this formula, you need to know *e*, that is, the relative change in the volume of the vibration isolator. However, calculating the volume of the rubber layer after deformation is a difficult task, since it is necessary to specify the shape of the deformed side surface, which cannot be specified in a single way. The choice of the most accurate shape of the side surface of the vibration isolator after deformation is the subject of further research.

Nevertheless, formula (22) can be used to determine the rubber layer sag in engineering calculations. To do this, it is enough to ignore the compressibility of the rubber, that is, to accept $e \approx 0$. Then $A_\mu \approx \frac{3}{2}G$ and the stiffness of the vibration isolator will be equal to

$$C=\frac{P}{\Delta}=3\pi RG\frac{\alpha}{1-\frac{\alpha}{\sqrt{6}}\mathrm{th}\frac{\sqrt{6}}{\alpha}}, \qquad (23)$$

where $\alpha=\frac{R}{2h}$ is the size ratio of the vibration isolator.

3. Experimental part

The purpose of experimental research is to determine the change in stiffness characteristics of rubber-metal elements depending on the ratio of their diameter to height.

Cylindrical rubber-metal vibration isolators, in which the rubber massif is a cylinder with a diameter of *D* and a height of h_p, to the ends of which metal cylindrical plates are vulcanized over the entire surface, have been subjected to laboratory studies. The diameter of the metal plates slightly exceeds the diameter of the rubber massif. Such vibration isolators can be tested both individually and in stacks, stacking them one on top of the other. In order for the stacks to be stable under compression, recesses with a diameter equal to the outer diameter of the upper plates are provided in the lower metal plates, which preserves the alignment of all vibration isolators in the stack.

In total, three standard sizes of vibration isolators were tested (fig. 2). All vibration isolators had a diameter $D = 200$ mm and differed only in the thickness of the rubber layer $h_p = 5$; 10; 20 mm. Before the tests, the vibration isolators were assembled in stacks. The number of vibration isolators with $h_p = 5$ mm was 20 pcs.; with $h_p = 10$ mm – 10 pcs. and with $h_p = 20$ mm – 5 pcs. So, the total thickness of the rubber massif for all standard sizes was 100 mm. All rubber-metal elements were made of medium-filled rubber type 2959. The research was carried out on a mechanical press and a hydraulic one of the PR-250 type. Before the tests, all stacks were subjected to training – three times deformation followed by unloading. The holding time before taking strength and sag readings was 5 min on a mechanical press and 3 min on a hydraulic press.

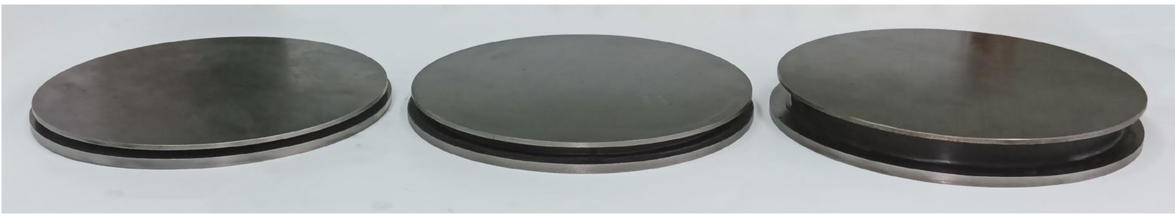

Figure 2 – Different sizes of vibration isolators that were subjected to laboratory tests

Fig. 3 shows the obtained dependences of "compression force – sag" for three types of vibration isolators, where 1 is the thickness of the rubber layer of the vibration isolator of 5 mm; 2–10 mm thick; 3–20 mm. As can be seen from the obtained graphs, the dependence "force – sag" for all types of vibration isolators is clearly nonlinear, and the degree of nonlinearity increases with a decrease in the thickness of the rubber layer of the vibration isolator.

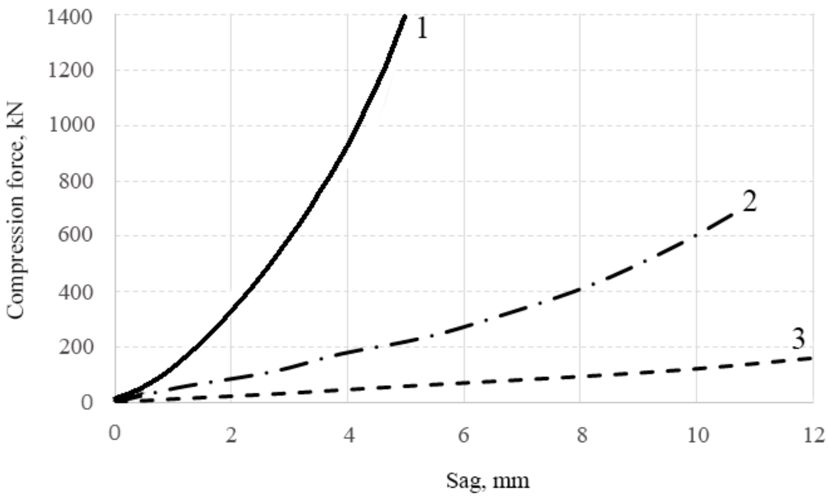

Figure 3 – Experimental dependences "compression force – sag"

Fig. 4 shows the graphs of changes in the compressive stiffness of vibration isolators depending on the ratio of the radius of the rubber layer to its thickness, obtained by calculation (solid line) and as a result of experimental studies (dashed line).

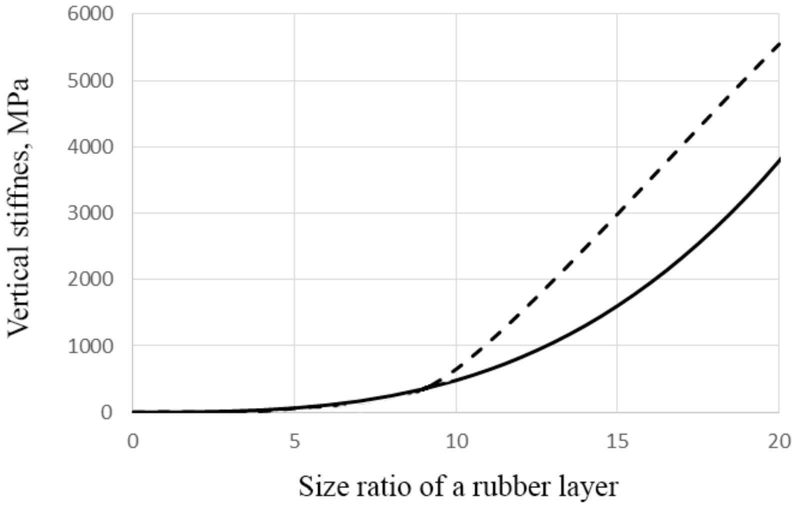

Figure 4 – Dependence of vertical stiffness on the size ratio of the rubber layer of the vibration isolator

A comparison of the obtained results shows that the calculation of the vertical stiffness for rubber 2959 according to formula (23) almost perfectly coincides with the experimental data at the value of $R/h \leq 10$. For thinner layers of rubber, it is necessary to take into account the compressibility or introduce a stiffness amplification factor.

4. Conclusions

1. Using the Ritz method, an expression was obtained for the sag of the rubber element taking into account the change in the volume of the rubber layer, which can be used specifying the shape of the deformed side surface.

2. To calculate the vertical stiffness of the shock absorber, an engineering formula is obtained, which gives a good result that coincides with the experiment, when the ratio of the radius of the element to its thickness does not exceed ten, and should be refined for thinner elements using a stiffness amplification factor or taking into account compressibility.

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ОБГРУНТУВАННЯ ПАРАМЕТРІВ ТА ЕКСПЕРИМЕНТАЛЬНІ ДОСЛІДЖЕННЯ ВІБРОІЗОЛЯТОРІВ ВИСОКОЇ НЕСУЧОЇ ЗДАТНОСТІ

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Анотація. При проектуванні будівель та споруд для експлуатації в сейсмічно небезпечних районах зазвичай застосовуються конструкції підвищеної несучої здатності. В даний час в Україні та за кордоном отримали застосування «системи активного сейсмозахисту», що знижують сейсмічні навантаження на споруди. Більшість систем відноситься до засобів кінематичної віброізоляції. Захист від вібросейсмонавантаженості забезпечується пристроями, розташованими між спорудою (будівлею) та опорною конструкцією. У якості таких пристроїв застосовуються віброізолятори різних конструкцій, демпфери, динамічні гасники і т.п.

У якості віброізоляторів найбільшого поширення набули конструкції з використанням еластомерних матеріалів, переважно гуми. Застосування в системах вібросейсмозахисту будівель і споруд еластомерних блоків характеризується високою вертикальною жорсткістю, низькою жорсткістю на зсув, високою дисипацією енергії, вони мають високу надійність і відсутність раптової відмови.

Незважаючи на значні досягнення в галузі конструювання еластомерних деталей, простоту форм та великий досвід їх застосування досі не створено конструкції із заданими фізико-механічними характеристиками. У зв'язку з цим велике значення мають досить прості та точні методи розрахунку, які можуть бути використані на стадії проектування.

Розглядається розрахунок тонкошарових гумометалевих віброізоляторів при статичному стиску. Розрахунок виконаний з використанням методу Ритца для осесиметричної задачі з припущенням, що осьові переміщення не залежать від радіусу окремого елемента, а є лише функцією осьової координати. Експериментальна перевірка отриманих результатів виконана з використанням гумометалевих віброізоляторів діаметром 200 мм та висотою гумового шару 5 мм, 10 мм та 20 мм, які складалися у стопки з товщиною гумового масиву 100 мм.

Порівняння розрахункових та експериментальних даних показує, що до значення співвідношення радіуса елемента до висоти, що дорівнює десяти, величини жорсткості практично збігаються. Для тонших елементів потрібне введення відповідного коефіцієнта.

Ключові слова: гумометалевий віброізолятор, тонкошаровий елемент, фізично нелінійне середовище, стисливість, пружний потенціал.