Research of rubber-cord characteristic of reduced rigidity compensator in ABAQUS packet

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Abstract. The aim of the article is to study the design of rubber-cord compensator for damping and compensating pipes movement and vibration isolation of other structures in vibroactive systems with the required specifications. As a result of calculations, graphical "force-deformation" curves were obtained in the axial and radial directions, the stiffness of the compensator was determined on their base. The stress-strain state of the rubber-metal element is determined, so one concludes that the compensator under study meets the strength criteria.

1. Introduction
Compensators of various designs find wide application for reducing deformations from the effect of temperature elongations of pipelines and vibrations of technological equipment in various industries and, in particular, oil and gas pipeline transport. A reinforced rubber-cord compensator is the most effective one, when its design there is a demand for stiffness, strength and sealing characteristics. The aim of this article is to study the design of a rubber-metal compensator for damping and compensating pipes movement and vibration isolation of other structures in vibroactive systems with the required specifications.

2. Problem statement
When examining a rubber-metal compensator with reduced stiffness, the task is to determine the following characteristics:

a) static stiffness in the axial and radial directions;
b) strength and stress-strain state;
c) to prove the choice of material characteristics.

The general view of the element of the rubber-cord compensator is shown in Fig. 1:

![Figure 1. General view of the rubber-cord element of the compensator.](image)

When calculating the transverse stiffness, the compensator should be pre-stressed in the axial direction by \( l_z = 1 \) mm. Therefore, the first step is an axial compression of 1 mm, and then a shift in the radial direction by \( l_x = 20 \) mm. The diagram of the compensator load (when calculating axial and radial stiffness) is shown in Fig. 2.
2 is radial displacement by $l_z = 20$ mm.

The scheme of loading the compensator when calculating the strength is shown in Fig. 3.

3. Theory

The main materials of the compensator are rubber and metal plates. Rubber is a material capable of large deformations, so it belongs to the category of hyper elastic materials; models based on the energy potential are used to describe them [1]. A polynomial potential is accepted for calculations, and it is expressed by ratio [2, 3]:

$$U = \sum_{i,j=1}^{N} C_{ij}(\bar{T}_i - 3)^i(\bar{T}_2 - 3)^j + \sum_{i=1}^{N} \frac{1}{D_i} (J - 1)^{2i},$$

where $U$ is potential energy of deformation per unit volume; $C_{ij}, D_i$ are parameters depending on the material; $N$ is degree of the polynomial; $\bar{T}_i, \bar{T}_2$ are the first and second invariants of the deviator of the stress tensor; $J$ is a determinant of volumetric compression.

To set the material of the reinforcing plates a traditional approach was used and it is based on the elasticity modulus of the tested material.

Calculation of the compensator characteristics was carried out on the basis of the ABAQUS package [1]. When creating a solid model, hexagonal elements were used: three-dimensional, eight-nodal cubic elements of the 1st order. Three-dimensional model of the rubber-metal compensator is presented in Fig. 4.
The connection of structural details was carried out without contacts - by merging the nodes of the boundary layer grid. The solution was performed considering the geometric and physical nonlinearity with a direct consecutive increase in the load \([4, 5]\).

It is accepted for calculations: flanges and plates are Steel 45, elastic element is rubber. Rubber deformation does not follow Hooke's law, so one of energy potential models is used for calculation \([6, 7, 8, 9]\):

- polynomial \(N = 1\) (Mooney-Rivlin);
- polynomial \(N = 2\);
- Ogden \(N = 2\);
- Ogden \(N = 3\);
- reduced polynomial \(N = 1\) (Neo-Hooke);
- etc.

To calculate the required characteristics is to obtain curves of two-axis stretching, shear, and volumetric compression. The rubber characteristics for two-axis stretching are shown in Fig. 5. When the stretching deformation is up to 10% for the given rubber at linearly calculation then elasticity modulus is \(E \approx 4.5\) MPa.

As a result of preliminary calculations of the compensator stiffness, the graphical dependences "force-deformation" in the axial (Fig. 6) and radial (Fig. 7) directions were obtained. Based on the graphs, the stiffness in two directions is determined from the equation:

\[
C_{x(x)} = \frac{\Delta F}{\Delta z},
\]

where \(\Delta F\) is change of force (H); \(\Delta z\) is absolute deformation value (mm).
The stiffness is \( C_z = 186 \text{ kN/mm} \) for the axial direction. The stiffness is \( C_x = 3.78 \text{ kN/mm} \) for the radial direction.

To obtain the necessary hardness according to the specifications for the compensator design there may be used rubber with other modulus or the number of reinforcing plates or the thickness of the rubber layer is to be changed. Impact of rubber elasticity to the radial rigidity of the compensator is presented in Fig. 8.

\[
\mu = \frac{\sigma_b}{\sigma_{\text{max}}},
\]

where \( \sigma_b \) is the limit strength material, \( \sigma_{\text{max}} \) is maximum calculated value of stresses.

4. Calculation results

When calculating the stress-deformed state in the compensator, ABAQUS package studies the stresses and strains to be in the rubber solid and metal rings for two cases of loading compensator:
– with a nominal inner pressure $P = 6.3$ MPa;
– with a nominal inner pressure along with maximum deformation in the axial and radial directions. Findings for the first case are presented in Fig. 9, 10 and 11.

For the second case, the calculation is performed, additionally pressing the compensator by 1 mm in the axial direction and shifting by 20 mm in the radial direction. Findings are shown in Fig. 12, 13 and 14.
5. Discussion
Findings show that the maximum stresses on the inner face of the plates are $\sigma_{\text{max}} = 135.7$ MPa for the first case of loading, and they are $\sigma_{\text{max}} = 0.99$ MPa in rubber at points of stress concentration near the inner edges of the plates. The deformations in the rubber solid are most significant in the layer between the flanges and plates, where they have $\varepsilon = 0.13$.

The findings show that the maximum stresses on the inner faces of plates slightly changed for the second case of loading and they are $\sigma_{\text{max}} = 138.2$ MPa. The stresses changed more significantly in rubber and they are $\sigma_{\text{max}} = 1.7$ MPa. The stress values are $\sigma_{\text{max}} = 2.5$ MPa at stress concentration points near the sharp edges of the plates. Deformations in rubber solid are the most significant between the plates they are $\varepsilon = 0.28$.

6. Conclusion
The safety factor of the compensator plates is $\mu = 3.4$.

The limiting stresses for the investigated rubber are $\sigma = 7.5$ MPa. So, the rubber safety factor is from $\mu = 4.4$ to $\mu = 3.0$ on the edges of the plates.
As a result of the calculations, it was determined that the investigated design of the rubber-metal compensator meets the criteria for strength. Stiffness specifications also are carried out, applying rubber with extreme minimal module of elasticity.

7. References

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