ANALYSIS OF THE DAMPING SYSTEM OF THE EQUIPMENT WITH VIBRATORY MOTION BY USING FINITE ELEMENT METHOD

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Abstract: Theoretical researches whose results are presented in this article are the subject of damping system of equipment with vibration moving. The damping system studied is formed from rubber with no insertion, which, in addition to the role of damping, it also has the role of supporting and takeover the shocks and vibration transmitted. In this paper we presents the results obtained using the finite element method, the system deformed state, deformation tensions and displacements of the analyzed system. The results were obtained and interpreted by means of the work program, Solid Works.

Keywords: Analysis, finite element method, system damping, vibratory motion

1. INTRODUCTION

Ensure anti-vibration protection of the assembly’s components, in order to ensure operational sustainability, is a fundamental issue of efficient use of equipment with vibratory motion. If this equipment is not designed, manufactured, installed and adjusted properly, it can have the following disadvantages: high noise level, transmitting the dynamic loads to the foundation or to the adjacent building elements, low performance or even failure of the functional role. An important component in the construction of these equipments with vibratory motion system is also the elastic system composed of rubber with no insertion (Fig. 1), which is intended to absorb vibration transmitted and for supporting. The elastic systems made from rubber are widely used in mechanical engineering, as a result of the special properties of them. Among these properties, the most important are: high damping capacity, construction and simple technology, low cost, safe and quiet. High damping capacity of these rubber systems is due to internal frictions that appear in the mass of the rubber, which can absorb up to 40% of the energy received. Rubber properties are influenced by the environment (temperature, radiation, humidity, chemical agents, etc.). Under the action of environment and vibration in time, the rubber properties are deteriorating [3].

The main properties that determine the working capacity of rubber are elasticity, fatigue resistance and hysteresis losses to the stress variables. Properties of rubber to dynamic loads differ from those at static. Strain cycle for a steady state is represented as a curve called the closed hysteresis loop. The force of deformation in small areas (in small deformations domain) for the non-linear isolators can have a linear behavior providing a particular constant frequency [3].
where \( g \) is the gravitational acceleration, \( x_\text{st} \) static static deformation (arrow).

In order to achieve effective isolation of vibration, it is necessary to avoid functioning at resonance, and the vibration transmissibility of the foundation, \( T \) equals subunit, is, [1] [3]:

\[
[T] = \frac{1}{1 - (\frac{f}{f_0})^2}
\]

where \( f \) is the frequency of the disturbing force.

2. MATERIAL AND METHOD

The finite element method is based on dividing the mathematical model components that do not overlap with a simple geometry, called finite element (fig. 2). The response of each element is expressed in terms of finite number of degrees of freedom of nodal points.

The answer of the mathematical model is approximated by the meshed model obtained. Each element has a set of distinct points called nodes. Nodes serve two purposes: defining the element geometry and number of degrees of freedom. They are normally located in the corners or the ends of, the elements of higher order are nodes placed on the sides or faces or inside the element [1, 2].

A mathematical model for analyzing an item, involves determining a large number of variables and functions \( u'(\xi) \) representing the displacements, strains and stresses of that item, \( \xi \) are the coordinate of the functions \( \xi (x, y, z) \), defined in the domain points where is defined the 3D model.

To establish an approximate solution is sufficient to describe an expression containing \( n \) parameters approximation. The finite element approximation is done at the nodes and has the form, [4]:

\[
\begin{bmatrix}
    u_1 \\
    u_2 \\
    \vdots \\
    u_n \\
\end{bmatrix} = [N][u]
\]

Where \( u_i \) are the nodal parameters of approximations, \( N_i(\xi) \) function interpolation or approximation, which usually have polynomial forms.

Approximate solutions \( u(\xi) \) can be constructed throughout the \( V \) definition field of the structure element or \( V_e \) elementary sub domains – of the finite elements – which means that \( \Sigma V_e = V \). The nodal approximation is made on \( V_e \) sub domains, which are actually the finite elements, the values of approximate functions \( u(\xi)=u'(\xi) \) are the nodal variables and \( x_i \) the coordinates of nodes. It can build a functional \( \pi \) that is the total potential energy of the structure element. Imposing this condition the stationary functional, [4]:

\[
\delta\pi = W = 0
\]

Is obtained the characteristic equation:

\[
W(u) = \int_V R(u)wdV = \int_V w(L(u)-f)dv = 0
\]

where \( L \) (u) is the exact solution, \( f \) the approximate solution, \( R(u) = (L(u)-f) \) the residue function, \( w \) the weighting function, also called correction functions.

The accuracy of the \( u \) solution depends on the choice of weighting function \( w \), which forms [4]:

\[
w = \sum_{i=1}^{\alpha} \beta_i \psi_i
\]

where \( \psi_i \) are a set of linearly independent functions, \( \beta \) arbitrary numerical coefficients.
Assuming that the approximate solutions satisfying boundary conditions of the element structure, the approximation error represented by $R(u)$ residue is weighted by multiplying with the weighting functions $w$, throughout the field $V$. If the functional structure is $\pi(u)$ this may be expressed through the mesh becomes [4]:

$$\pi(u) = \pi[u(a_1, a_2, \ldots, a_n)]$$  \hspace{1cm} (7)

or

$$\frac{\delta \pi}{\delta a_i} = 0, \text{ i=1, 2... n}$$  \hspace{1cm} (8)

Is obtained the system of equations, by solving which is determined the parameters of approximation [4]:

$$\frac{\delta \pi}{\delta a_i} = 0, \text{ i=1, 2... n}$$

Using the program Solid Works we made the 3D model of the element analyzed, in our case the rubber with no insertion, and then is analyzed using the finite element method. The data, by means of which the analysis was performed, are shown in Table 1.

Table 1. Main characteristics of the rubber

<table>
<thead>
<tr>
<th>Material</th>
<th>Dimensions [mm]</th>
<th>Elastic modulus [N/m²]</th>
<th>Poisson’s ratio</th>
<th>Rigidity modulus [N/m²]</th>
<th>Density [kg/m³]</th>
<th>Yield strength [N/m²]</th>
<th>Hardness [Shore]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rubber with no insertion</td>
<td>Outside diameter: θ 75</td>
<td>61x10³</td>
<td>0,49</td>
<td>29x10³</td>
<td>1000</td>
<td>9237370</td>
<td>40…50</td>
</tr>
<tr>
<td></td>
<td>Interior diameter: θ 20</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Height: 95</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3. RESULTS AND DISCUSSION

The distribution of von Misses equivalent stresses in the rubber according to the criteria is shown in Fig. 3. It is noted that the maximum equivalent stresses are found at the contact surfaces where the rubber grip is made to body equipment.

Figure 3. Equivalent von Misses stress distribution

The distribution of the total displacement for the same section is shown in Fig. 4. The maximum displacement occurs at the top of the rubber where the forces act. Is noted that the arrow appear where forces act and compresses the respectively rubber.

Given that the frequency of the disturbing force is $f=960 \text{ s}^{-1}$, to the displacements obtained by finite element method using Solid Works program and to expression (2) we calculated vibration transmissibility, which is displayed in Table 2.

Figure 4. Distribution of total displacements
Table 2. The results obtained

<table>
<thead>
<tr>
<th>No. item.</th>
<th>The force applied [N]</th>
<th>Maximum von Mises equivalent stress [MPa]</th>
<th>The maximum displacement [mm]</th>
<th>The particular constant frequency [s⁻¹]</th>
<th>The vibration transmissibility</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1200</td>
<td>1.26</td>
<td>8,623</td>
<td>5.4</td>
<td>3,16 x 10⁻⁵</td>
</tr>
<tr>
<td>2</td>
<td>1500</td>
<td>1.58</td>
<td>10,78</td>
<td>4.82</td>
<td>2.52 x 10⁻⁵</td>
</tr>
<tr>
<td>3</td>
<td>1800</td>
<td>1.89</td>
<td>12,93</td>
<td>4.4</td>
<td>2.1 x 10⁻⁵</td>
</tr>
<tr>
<td>4</td>
<td>2100</td>
<td>2.21</td>
<td>15,09</td>
<td>4.08</td>
<td>1.8 x 10⁻⁵</td>
</tr>
</tbody>
</table>

![Figure 5. Variation of equivalent stress and total displacements for the analyzed model](image1)

![Figure 6. Variation of the own frequency of the damper and the vibration transmissibility for the analyzed model](image2)

4. CONCLUSIONS

From this analysis the following conclusions are resulted:

✓ Solid Works program can be a good tool to analyze the different models and the results can provide good information about their behavior in actual working conditions;

✓ The software can model and analysis through the finite element method the nonlinear isolators, in order to highlight the deformed areas and tensions encountered;

✓ This analysis examined the model areas where the stresses are greatest, as well as, by increasing the force exerted to the model is submitted to greater strains. Thus appears the greatest risk of rubber aging, by repeated deformations.

✓ The particular constant frequency of the analyzed rubber decreases with increasing forces as it is subject to compression, and the vibrations transmissibility is kept to a subunit under the exerted forces, which is that vibrations are not transmitted to the foundation and not cause damage.

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