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OPTIMIZATION ALGORITHM FOR THE PASSIVE ISOLATION SYSTEM CONSISTING OF COMPOSITE NEOPRENE INTENDED FOR BUILDINGS SUBJECTED TO SEISMIC SHOCK ACTION

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Abstract: *This paper deals with new vibration systems intended for vibrations characterized by low tuning eigen frequency. The damping as well as the elastic characteristics are presented both for the whole system and its components. Thus, three passive isolation solutions where the viscoelastic elements consist on antivibrating rubber designated to attain appropriate performances in vibration isolation intended for equipment inside buildings (e.g. electric generator, air ventilation-conditioning systems) are presented.*
Keywords: *passive isolation, elastic elements, supporting elastic systems, viscous dissipation, structure analysis*

1. INTRODUCTION

The classic passive isolation solution is characterised by directly supporting of the equipment 1, having the weight *G* on the identical elements 2, (see Figure 1) the system being denoted DSS (directly supporting system).

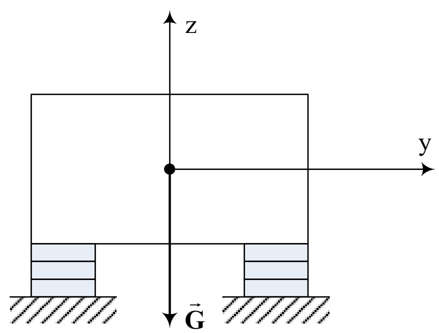

Figure 1: Directly supporting system.

The presented innovative solutions are patented and based on the static deflection amplifying under loading using properties of the elastic linked lever. In this respect, figure 2 illustrates the simple elastic system - SES basing on the effect of a single lever to amplify the deflection under loading.

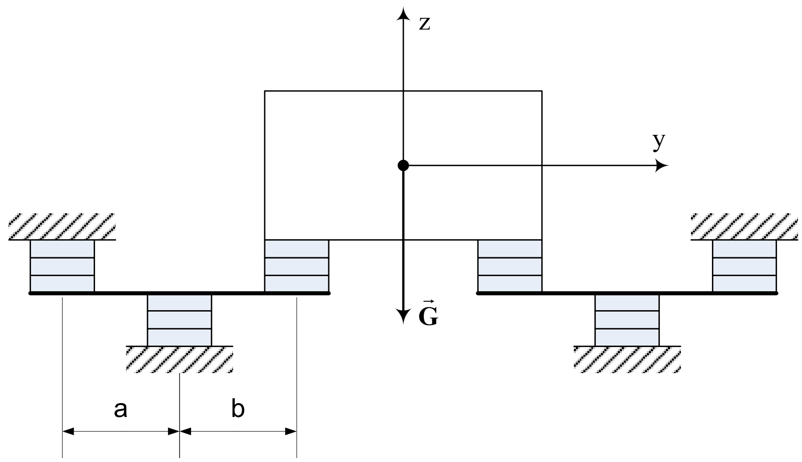

Figure 2: Simple elastic system.

Figure 3 represents the composed elastic system – CES basing on the cumulated effect of two levers with elastic links between them and the fixed base.

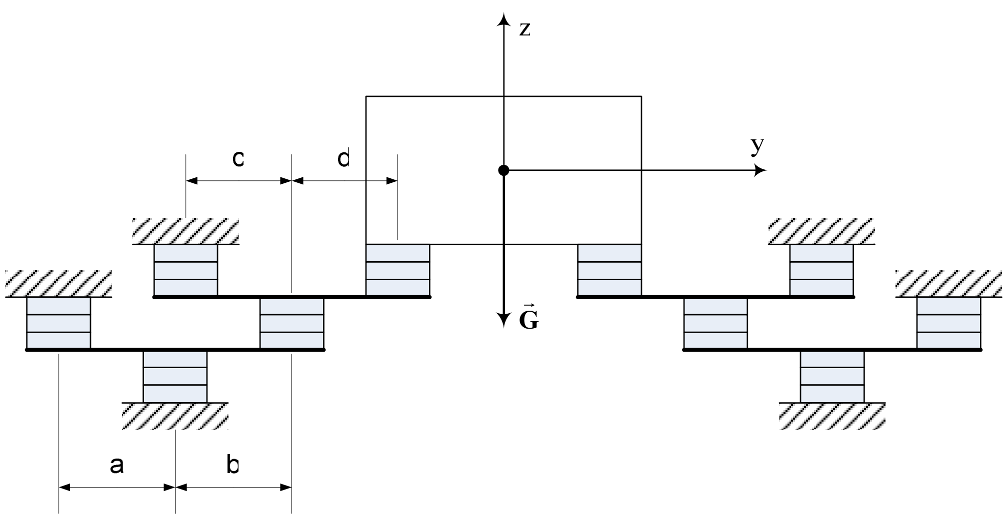

Figure 3: Composed elastic system

2. ANALYSIS OF THE OPTIMAL SOLUTIONS

All the passive elastic systems consist in identical antivibrating elements having cylindrical shape. The total number of the elements, depending on the passive isolation system configuration, differs as follows: $N_1 = 8\ elements$ for DSS; $N_2 = 12\ elements$ for SES; $N_3 = 20\ elements$ for CES. The total weight of the supported equipment is $G = 10^4\ N$ and the working exciting frequency is $f = 12{,}5\ Hz$ or $n = 750\ min^{-1}$.
The fundamental requirement for the optimal criterion is represented by the isolation degree, I ≥ 0,95, so that it results in the system configurations.

2.1. Directly supporting system - DSS

The static deformation Δ_{st} given by the equipment dead weight G for the directly supporting system consisting in N_1 rubber antivibrating elements with individual rigidity k_0, parallel connected can be expressed under the form:

$$\Delta_{st} = \frac{G}{k_0 N_1} \tag{1}$$

and the fundamental eigen frequency is given by:

$$f_0^{DSS} = \frac{1}{2\pi}\sqrt{\frac{g}{\Delta_{st}}} \tag{2}$$

where $g = 9{,}81\ m/s^2$ represents the gravity acceleration.

In case $G = 10^4\ N$, $k_0 = 1700\ N/cm$, $N_1 = 8\ elements$ it results in $f_0 = 5{,}81\ Hz$.

For the equipment working frequency $f = 12{,}5\ Hz$ in the diagram illustrated in figure 4 one can obtain the vibration isolation degree $I = 0{,}723 < 0{,}9$ showing that the passive elastic isolation is dissatisfactory for an imposed performance level $I_{nec} \geq 0{,}9$.

For the elastic model of the directly supporting system, the curve $I = I(\Omega)$ has been obtained basing on the analytic function of the isolation degree given by relation:

$$I = 1 - \left|\frac{1}{1-\Omega^2}\right| \tag{3}$$

where $\Omega = \dfrac{f}{f_0}$ represents frequency or relative pulsation.

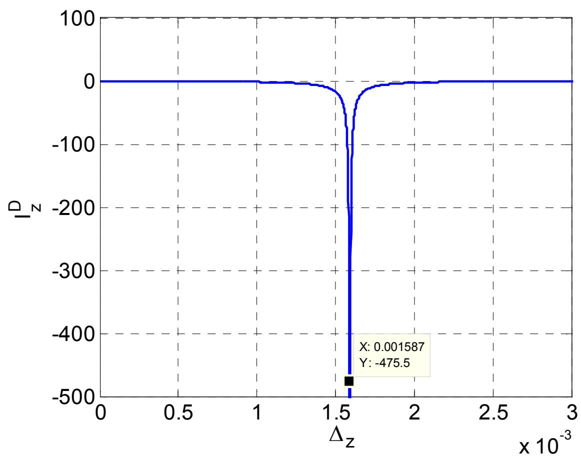

Figure 4: Variation of the isolation degree for DSS

2.2. Simple elastic system - SES

For the system illustrated in figure 2, the equivalent vertical rigidity in case of supporting on four levers with elastic links, we have:

$$k_{ech}^{SES} = \frac{k_0 N_2}{2(\lambda^2+\lambda+1)} \tag{4}$$

and the fundamental eigen frequency is

$$f_0^{SES} = f_0^{DSS}\sqrt{\frac{N_2}{2N_1(\lambda^2+\lambda+1)}}\,. \tag{5}$$

In case of SES with $N_1 = 8\ elements$, $N_2 = 12\ elements$, $f_0^{DSS} = 5{,}81\ Hz$ it results in

$$\nu = \frac{f_0^{SES}}{f_0^{DSS}} = \frac{0{,}56}{\sqrt{\lambda^2+\lambda+1}} \tag{6}$$

where $\lambda = \dfrac{b}{a}$ represents the coefficient controlling the static deformation multiplying effect for the first elastic stage of the simple elastic system.

$$\Omega_{SES} = \frac{f}{f_0^{SES}} = \frac{f}{f_0^{DSS}\nu} \tag{7}$$

where Ω_{SES} is the relative frequency of SES, f representing the perturbing frequency of the elastic supported equipment.

For the given situation $f = 12{,}5\ Hz$, it results in:

$$\Omega_{SES} = \frac{12{,}5}{5{,}81} \cdot \frac{1}{\nu} = \frac{2{,}15}{\nu} \tag{8}$$

where $\Omega_{SES}^2 = \dfrac{4{,}62}{\nu^2}$.

The vibration isolation degree for SES is given by relation:

$$I_{SES} = 1 - \left| \frac{1}{1 - \dfrac{4{,}62}{\nu^2}} \right| . \tag{9}$$

Figure 5 represents variation of the eigen frequencies ratio ν depending on λ coefficient and figure 6 represents the variation curve for the isolation degree I^{SES} as a function of the controlling coefficient λ.

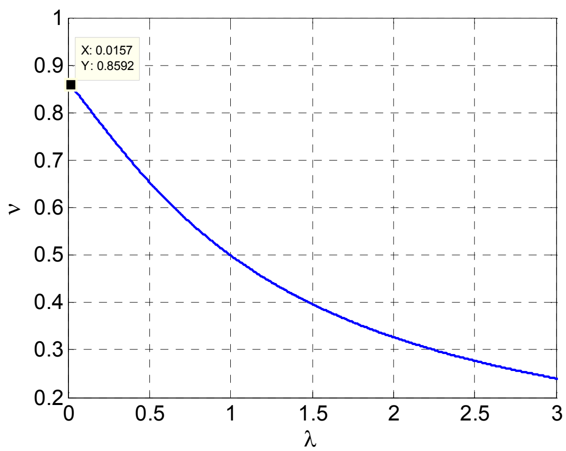

Figure 5: Variation of the relative eigen frequencies for SES

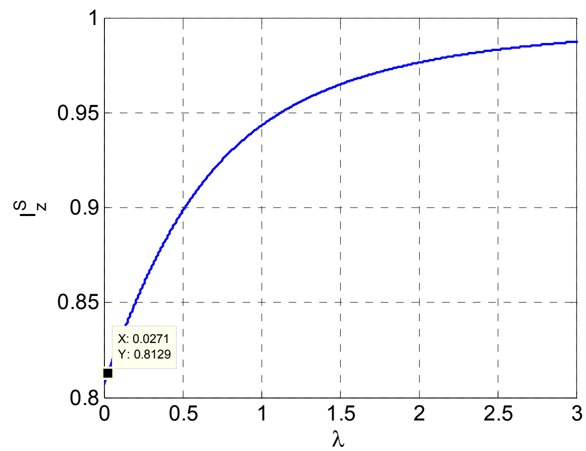

Figure 6: Variation of the vibration isolation degree for SES

2.3. Composed elastic system – SES

In case of the system consisting in two levers with elastic links, according to figure 3, the equivalent rigidity coefficient in the vertical direction is given by relation:

$$k_{ech}^{CES} = \frac{k_0 N_3}{2(\lambda^2 + \lambda + 1)(\mu^2 + 2\mu + 1) + \mu^2 + 1} \tag{10}$$

where $\mu = \dfrac{d}{c}$ represents the coefficient controlling the multiplying effect for the second elastic stage.

The relative eigen frequency $\varphi = \dfrac{f_0^{CES}}{f_0^{DSS}}$ can be written as:

$$\varphi = \frac{f_0^{CES}}{f_0^{DSS}} = \sqrt{\frac{N_3}{2N_1(\lambda^2 + \lambda + 1)(\mu^2 + 2\mu + 1) + \mu^2 + 1}} \tag{11}$$

In case of a two stages system with $N_1 = 8\ elements$, $N_3 = 20\ elements$, $f_0^{DSS} = 5{,}81\ Hz$, we have:

$$\varphi = \sqrt{\frac{1{,}58}{(\lambda^2 + \lambda + 1)(\mu^2 + 2\mu + 1) + \mu^2 + 1}} \tag{12}$$

The frequency ratio for the composed elastic system is:

$$\Omega_{SES} = \frac{f}{f_0^{CES}} = \frac{f}{f_0^{DSS}\nu} \tag{13}$$

where taking into consideration $f = 12{,}5\ Hz$ and $f_0^{DSS} = 5{,}81\ Hz$ it results in:

$$\Omega_{CES} = \frac{2{,}15}{\varphi} . \tag{14}$$

The isolation degree for the two elastic stages system is expressed by relation:

$$I_{CES} = 1 - \left| \frac{1}{1 - \frac{4{,}62}{\nu^2}} \right| \tag{15}$$

For the current variable λ and parametric discreet variation $\mu = 1, 2, 3, 4$ it results in the variation curves for the relative eigen frequencies $\varphi = \varphi(\lambda, \mu)$ and the isolation degree $I_{CES} = I_{CES}(\lambda, \mu)$ represented in figures 7 and 8.

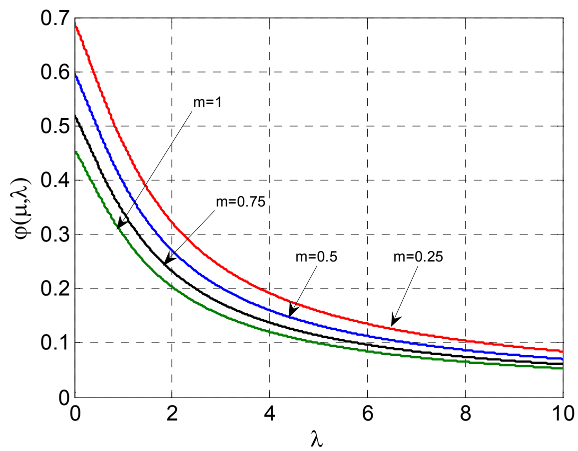

Figure 7: Variation of the relative eigen frequencies for CES

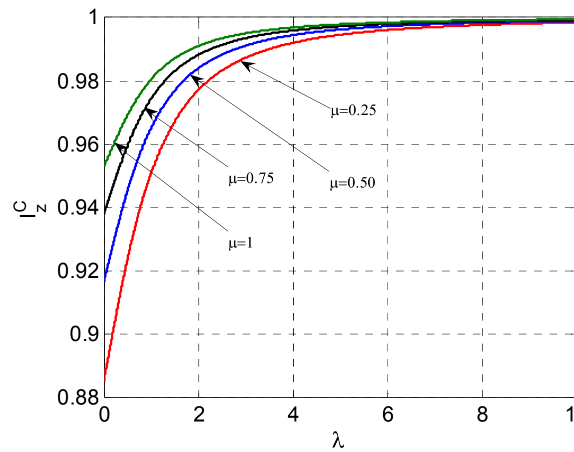

Figure 8: Variation of the isolation degree for CES

3. CONCLUSION

The performance parameters for the vibration passive isolation systems are the following:

- the equivalent rigidity in the dynamic freedom degree direction;
- the vertical static deformation under the total gravity load;
- the isolation degree in the dynamic freedom degree direction.

The innovative passive isolation systems, patented in Romania, conceived as elastic structures with one or two stages have the best performances for isolation degrees > 98 %.

While designing these systems, all the resistance and stability conditions must be taken into account in order to ensure the insulated dynamic equipment operation.

REFERENCES

[1]Bratu, P.: Sisteme elastice de rezemare pentru mașini și utilaje, Editura Tehnică, București, România, 1990.

[2]Rivin, E.:Passive Vibration Isolation, ASME Press, New York, 2003.

[3]Bratu, P,: Analysis of insulator rubber elements subjected to actual dynamic regime, Proceedings of the 9th International Congress on Sound and Vibration, University of Orlando, Florida, USA, 2002.

[4]Bratu, P.: Antivibrating systems having amplified deflection under loading, vibration control and transmissibility, Proceedings of the International Conference of RSA on Sound and Vibration, Bucharest, Romania, 14 - 17 October 2004.