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Passive reduction of vibrations of building structures by means of synthesis methods

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Abstract. The paper presents a method for determining the vibration-damping elements for the mechanical system under consideration, necessary to obtain the desired amplitude value. The structural and parametric identification was performed by using the method for synthesizing the dynamical distribution in the form of sluggishness and mobility. This paper presents the solution of the problem of passive vibration reduction of discrete mechanical systems to minimize the buildings structure response on the negative impact of external factors.

1. Introduction

Advanced construction methods and the desire to construct the impressive buildings make that the contemporary buildings are getting taller. Unfortunately, this increase of the height is accompanied with increasing probability of low eigenfrequencies, hence these buildings are susceptible to wind-induced vibrations which can be highly disturbing for inhabitants.

In order to ensure the effective functionality of flexible structures, various design modifications are possible, ranging from alternative structural systems up to utilization of passive and active devices for energy dissipation [1,2]. The paper concerns on the formulation and the solution making of the problem of passive vibration reduction of mechanical systems in view of desired dynamic properties [3,4,5].

In the paper is proposed the use of methods for the synthesis [6-11] of discrete mechanical systems in order to determine the value of the damping elements that ensure appropriate dynamic properties of the structure being tested. The necessary therefore is to specify the criterion for searching the structure and parameters of the model basing on knowledge on the dynamic properties of a real object. The synthesis, as the inverse task, enables the identification of parameters for the desired properties in the form of frequencies and amplitude values. For this purpose, it has been formulated the conditions of physical reliability of dynamic characteristics in the form of the slowness and mobility, in the case of passive vibration reduction [9,10,11].

The values of passive components are generated in systems subjected to vibration reduction, on the basis of information on the dynamic state of the system [12,13]. The reduction of vibration is related with determining parameters that allow controlling the vibration amplitude of a chosen point in a feedback loop, or utilizing in the system so-called energy dissipating components. The conditions, under which the structural and parametric identification of the system [14,15,16], subject to vibration reduction, is realized, should meet in advance the required dynamic properties.

2. Dynamic model of the system

The analysed system has four degrees of freedom in the horizontal direction. The system was deflected from the equilibrium position under the influence of the horizontal force simulating the impact of wind at the frequency close to the first harmonic of the investigated model [1]. The investigated model is a four-storey wall frame shown in figure 1.

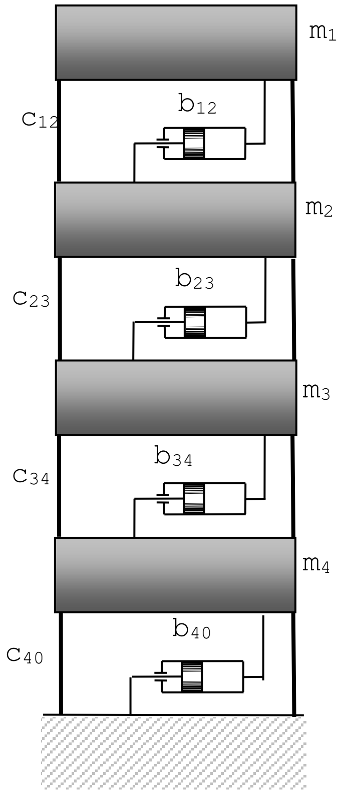

Figure 1. Four-storey wall frame.

The data of the analysed system was taken from [1], in which the case of damping vibrations of the structure, a four-storey wall frame, caused by the earthquake was analysed. The values of inertial elements and elastic elements of the analysed model were: m1= 450 000 kg, m2=m3=m4= 345 000 kg, c12= 18 050 000 N/m, c23= 326 000 000 N/m, c34= 285 000 000 N/m, c40= 250 000 000 N/m.

In the case of the problem under consideration, the dynamic characteristic in the form of stiffness of the undamped system (equation 1) was determined, assuming the force acting and system response on the inertial component marked with the symbol m_1 (figure 1). The result of which are properties of the system in the form of four resonance frequencies ω_i and three anti-resonance frequencies $\omega_{i,i+1}$.

$$U(s)=\frac{1}{V(s)}=\frac{F(s)}{X1(s)}\frac{(s^2+\omega_1{}^2)(s^2+\omega_2{}^2)(s^2+\omega_3{}^2)(s^2+\omega_4{}^2)}{(s^2+\omega_{12})(s^2+\omega_{23}{}^2)(s^2+\omega_{34}{}^2)}$$
$$=\frac{(s^2+5.6923^2)(s^2+13.6840^2)(s^2+36.4962^2)(s^2+52.9856^2)}{(s^2+13.4247^2)(s^2+36.4885^2)(s^2+52.9844^2)}. \tag{1}$$

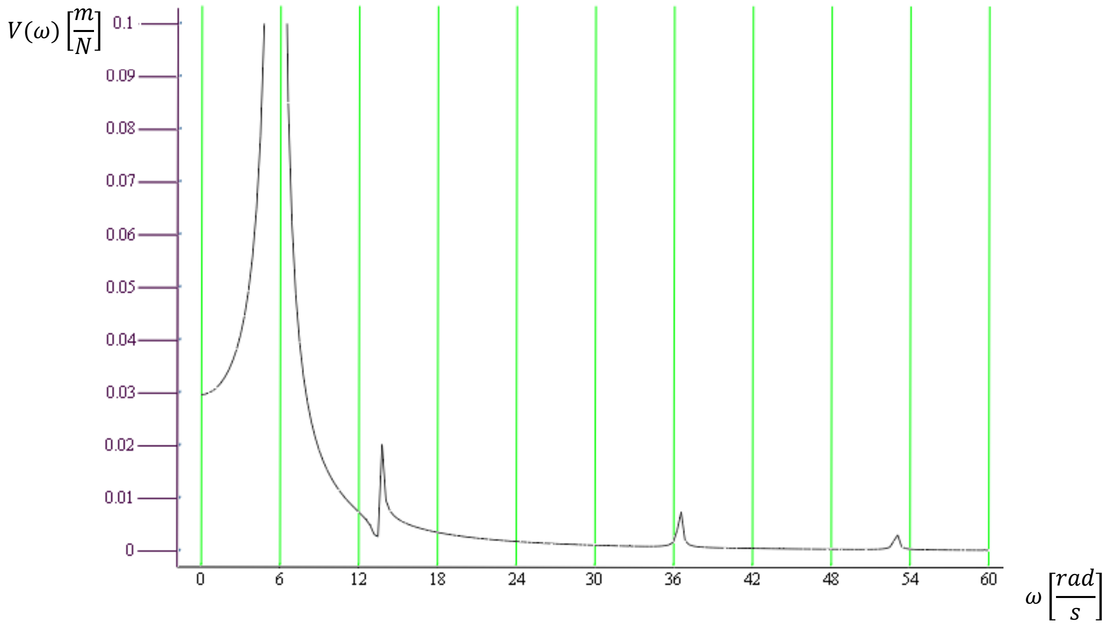

Figure 2. Characteristic function described with the equation (1).

Dynamic characteristic (figure 2) is the starting point for further considerations as a result of which will be determined the unknown parameters of the system $b_{12}, b_{23}, b_{34}, b_{40}$ meeting the desired vibration amplitude of an inertial component subjected to the horizontal force $F_{m1}(t) = \mathrm{A}\,sin(\omega_1 t)$.

3. Passive vibrations damping

In order to determine the values of damping components, the dynamic characteristic *U (s)* (1) should be expressed in the form of a stiffness function describing damped systems [4,5,11] in the following way:

$$U(s,h) = \frac{1}{V(s,h)}$$
$$= \frac{(s^2 + 2hs + 5.6923^2)(s^2 + 2hs + 13.6840^2)(s^2 + 2hs + 36.4962^2)(s^2 + 2hs + 52.9856^2)}{(s^2 + 2hs + 13.4247^2)(s^2 + 2hs + 36.4885^2)(s^2 + 2hs + 52.9844^2)}, \quad (2)$$

where: h - means the factor of free vibration decreasing expressed in $\left[\frac{\mathrm{rad}}{\mathrm{s}}\right]$.

The analytical form of the dynamic characteristic (2) is the function on the basis of which the coefficient of frequency decrease of free vibrations h is determined. This coefficient corresponds to the given amplitude value of the reduced resonance frequency. The desired amplitude is the maximum deflection of the inertial component, corresponding to the response of the system at the unit amplitude of the extortion force. Using the graphical interpretation showing the dynamic properties of the analysed system in the form of a spatial function $V(s,h)$ (2), the coefficient of dropping of free frequencies h (figure 3) is determined. Important and simultaneously dangerous for the structure, there may be a case when the frequency of extorting force $F_{m1}(t)$ is close to the first form of free vibrations. Therefore, the amplitude of the system's vibrations in such a situation should not be great. In the tests $A_{max} = 0.056$ [m] was assumed on the basis of which the value of the coefficient $h_1 = 1.5 \left[\frac{\mathrm{rad}}{\mathrm{s}}\right]$ was determined (figure 3).

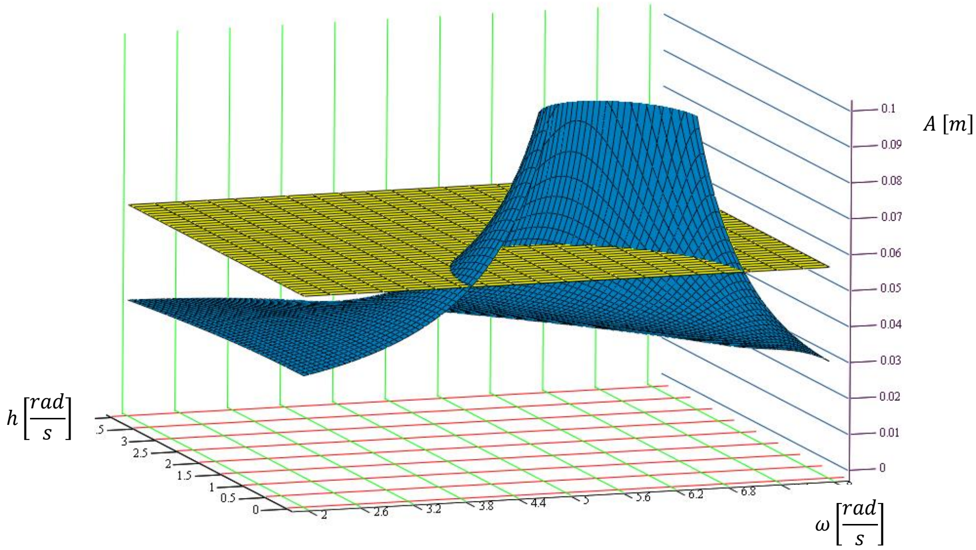

Figure 3. Defining the requirements for characteristics.
Selected frequency $\omega_1 = 5.6923 \left[\frac{\text{rad}}{\text{s}}\right]$ plane $A_{max} = 0.056$ [m].

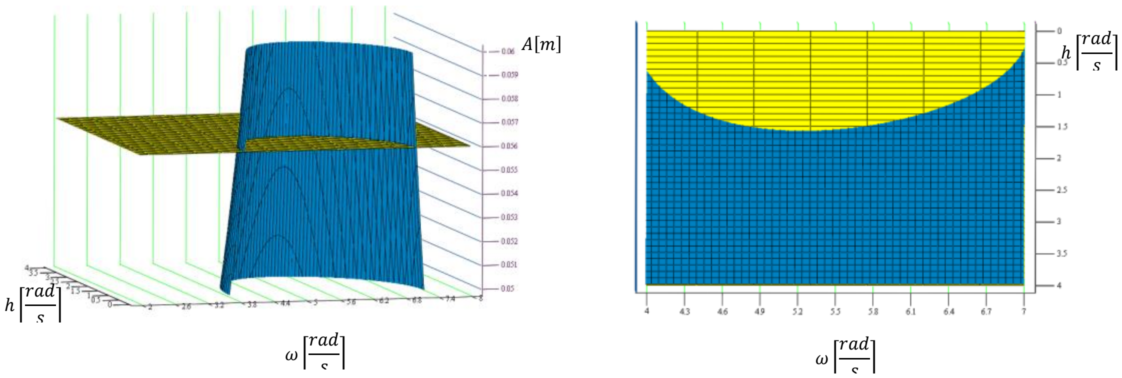

Figure 4. Graphical interpretation for determining the free vibration frequency drop coefficient $h_1 = 1.5 \left[\frac{\text{rad}}{\text{s}}\right]$ depending on the pre-set vibration amplitude.

Assuming that the passive component of vibration reduction is a two-node damping one with the value proportional to the value of elastic components, then the free vibration decreases coefficients, in the succeeding resonance $\mathrm{h_i}$ and antiresonance $\mathrm{h_{i,i+1}}$ zones, of the function (2) take the following value:

$$h_1 = \frac{\alpha(\omega_1)^2}{2} => \alpha = \frac{2h_1}{(\omega_1)^2} = 0.0926[\text{s}], \tag{3}$$

$$h_i = \frac{\alpha(\omega_i)^2}{2},\ h_{i,i+1} = \frac{\alpha\left(\omega_{i,i+1}\right)^2}{2} \tag{4}$$

where: $\alpha = 0.0926[\text{s}] = idem$ - the constant of proportionality.

Taking into account the character of the existing passive element and determined drop coefficients, the dynamical characteristics (2) takes the following analytic form (5):

$$U(s,h) = \frac{1}{V(s,h)}$$
$$= \frac{(s^2 + 2h_1 s + 5.6923^2)(s^2 + 2h_2 s + 13.6840^2)(s^2 + 2h_3 s + 36.4962^2)(s^2 + 2h_4 s + 52.9856^2)}{(s^2 + 2h_{12} s + 13.4247^2)(s^2 + 2h_{23} s + 36.4885^2)(s^2 + 2h_{34} s + 52.9844^2)} \tag{5}$$
$$= \frac{(s^2 + 3s + 5.6923^2)(s^2 + 17.34s + 13.6840^2)(s^2 + 123.34s + 36.4962^2)(s^2 + 259.98s + 52.9856^2)}{(s^2 + 16.69s + 13.4247^2)(s^2 + 123.29s + 36.4885^2)(s^2 + 259.96s + 52.9844^2)}.$$

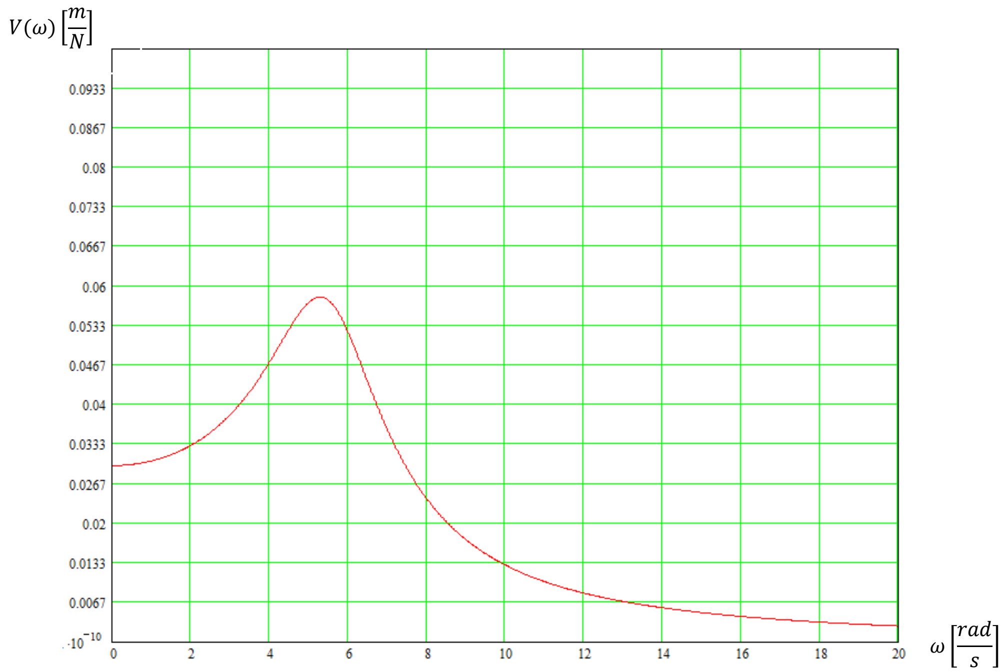

Figure 5. Dynamical characteristics (5) for the susceptibility of systems obtained as a result of passive vibration reduction performed.

The passive vibration damping elements can be determined using the following dependence:

$$\begin{cases} \alpha = 0.0926 \\ b_k = \alpha c_k \end{cases} \tag{6}$$

where: b_k- the value of two-node damping element, c_k- the value of two-node elastic element, α- the constant of proportionality.

By using the dependence (6), it is possible to determine the values of searched two-node damping elements in the case depending on the structures obtained:

$$b_{12} = \alpha c_{12} = 1\,671\,178.73 \left[\frac{\mathrm{Ns}}{\mathrm{m}}\right], b_{23} = \alpha c_{23} = 30\,187\,600 \left[\frac{\mathrm{Ns}}{\mathrm{m}}\right], b_{34} = \alpha c_{34} = 26\,391\,000 \left[\frac{\mathrm{Ns}}{\mathrm{m}}\right],\quad b_{40} = \alpha c_{40} = 23\,150\,000 \left[\frac{\mathrm{Ns}}{\mathrm{m}}\right]. \tag{7}$$

The figures show the results of the response of particular stories before and after vibration reduction of the construction model (figure 1).

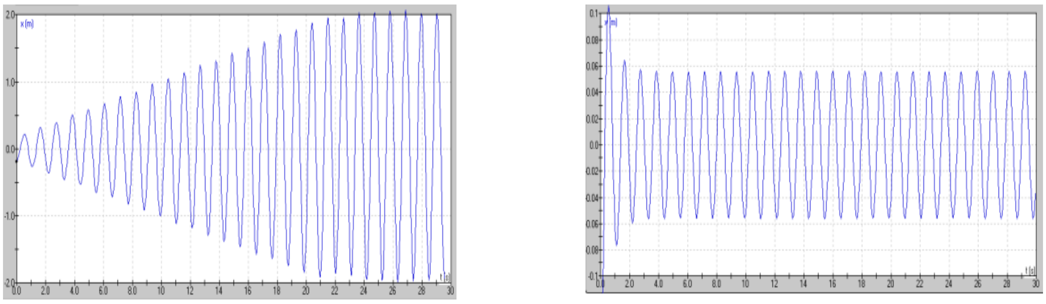

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Figure 6. Response of the component m1 on acting of the force $F_{m1} = 1\sin(5.6923t)$
a. Before vibration reduction; b. After vibration reduction.

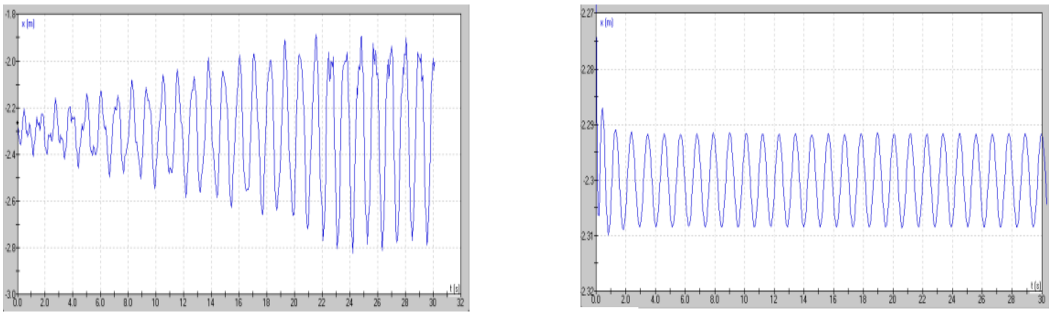

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Figure 7. Response of the component m2 on acting of the force $F_{m1} = 1\sin(5.6923t)$
a. Before vibration reduction; b. After vibration reduction.

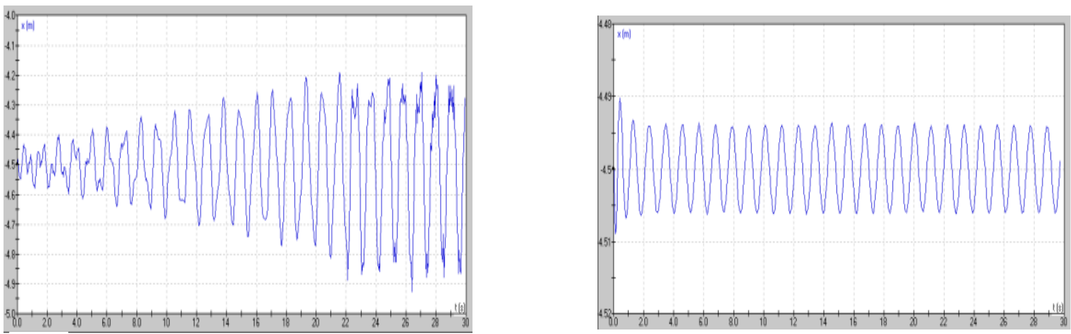

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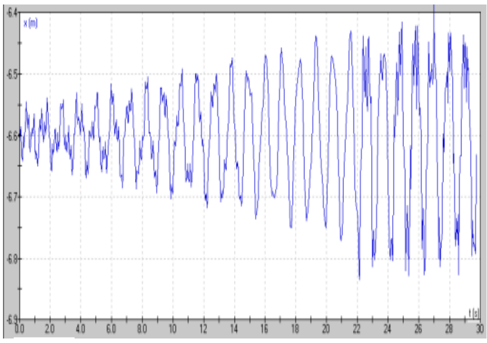
Figure 8. Response of the component m3 on acting of the force $F_{m1} = 1\sin(5.6923t)$
a. Before vibration reduction; b. After vibration reduction.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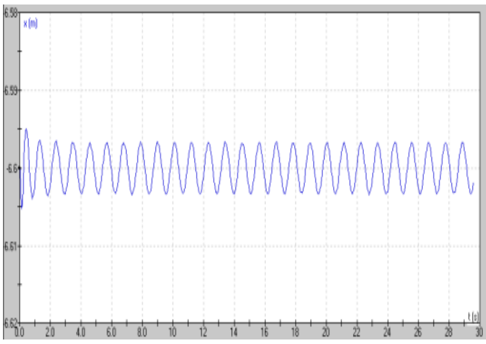
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Figure 9. Response of the component m4 on acting of the force $F_{m1} = 1\sin(5.6923t)$
a. Before vibration reduction; b. After vibration reduction.

4. Conclusions
The presented problem of passive vibration reduction of the identified structure and system parameters makes it possible to determine the values of passive elements compliant with the required amplitude value. This approach to the passive system vibration reduction allows designers to control the values of vibration amplitude of the system under consideration. By performing the passive vibration reduction in mechanical systems supported by synthesis methods, a sequence of structures and parameters applying to the dynamical characteristics in the form of resonance and anti-resonance frequencies is obtained. The set of models created that way constitutes a quality base of the identified object. One of those methods can be the structural or parametric sensitivity used to assess the impact of parameters on frequency changes.

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