Journal of Achievements in Materials and Manufacturing Engineering
VOLUME 47 ISSUE 2 August 2011

Application of kinematic excitation as implementation of active reduction of vibration

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Received 07.06.2011; published in revised form 01.08.2011

Analysis and modelling

ABSTRACT

Purpose: The work presents methods of reducing vibration with the use of passive and active elements as well as reveals the results of implementation of active reduction of vibration applying mechanical elements in the form of kinematic excitation.

Design/methodology/approach: The publication also includes structural and parametric synthesis understood as designing of mechanical systems with a desired frequency spectrum and presents analysis of obtained systems in order to verify if the latter meet related requirements.

Findings: The research reveals that the application of kinematic excitation as the implementation of active reduction of vibration does not satisfy expectations as obtained models are characterised by frequency spectrum different to the desired one.

Research limitations/implications: The research relates only to discrete vibratory systems, in which the reduction is carried out by means of mechanical elements. It is advisable to apply another type of elements e.g. electric ones.

Practical implications: The results represented this work extend the tasks of synthesis to other spheres of science e.g. electric systems. The practical realization of the reverse task of dynamics introduced in this work can find uses in designing of machines with active and passive elements with the required frequency spectrum.

Originality/value: Owing to the presented approach i.e. a non-classical synthetic method applied in designing mechanical systems, one (as early as at the design and construction stage) may verify future systems.

Keywords: Process systems design; Synthesis; Reduction of vibrations

Reference to this paper should be given in the following way:
K. Białas, Application of kinematic excitation as implementation of active reduction of vibration, Journal of Achievements in Materials and Manufacturing Engineering 47/2 (2011) 185-194.

1. Introduction

The phenomenon of vibration is very common in our environment. Vibration is understood as periodical changes of the state of a physical system (e.g. mechanical) proceeding around the state of equilibrium triggered by energy supplied to this system.

Mechanical vibration is often deliberately introduced to a system in order to perform expected tasks related to technological processes. Most common applications can be found in equipment and machinery used for comminution, densification, purification, milling, crushing, drilling, boring or grinding of materials.

In addition to vibration being the operational basis of some equipment and machinery there are also those having a harmful effect on the operation and reliability of mechanisms, components and operators. For this reason, many research centres investigate the possibilities of reducing or eliminating vibration. The tasks of designers and constructors of machines and objects contain the prevention of improper use of the latter or their adaptation so that they could meet specific requirements.

There are numerous ways to prevent excessive vibration affecting the elements and components of equipment and machinery, including passive, semi-active and active methods.

Passive vibration-reducing methods consist in the application of passive elements. In case of passive methods of vibration reduction it is possible to observe strong relation between the efficiency and vibration frequency as well as significant sensitivity to changes of parameters. The methods in question are characterised by simple structure and possibility to dissipate or store energy without the necessity of connecting additional energy sources. In case of such elements there is no possibility to change their parameters in time.

The most common applications of passive elements include the following dampers:

- viscoelastic,
- viscous,
- dynamic.

The operation principle of viscoelastic dampers consists in vibration causing steel plates to move. Between the plates there is one or two layers of viscoelastic material which undergo shearing as plates move and thus dissipate energy.

An example of viscous damper is Taylor damper. The basic components of the damper include a cylinder filled with high-viscosity liquid and a piston with a head containing openings enabling the flow of the liquid. The damping consists in dissipating energy during the flow of the liquid caused by the movement of the piston – mechanical energy is then transformed into thermal one.

In case of dynamic (mass) dampers, structural solutions are various and many.

Semi-active methods of vibration reduction utilise passive elements, except that, unlike in passive systems, their parameters can be modified in real time. For this reason these systems are referred to as the combination of active and passive methods Unlike passive systems, semi-active systems do not obtain energy from external sources.

The most popular semi-active dampers are:

- semi-active hydraulic dampers,
- semi-active viscous dampers,
- semi-active electro-rheological dampers,
- semi-active magneto-rheological dampers,
- semi-active friction dampers,
- semi-active dampers changing structural rigidity,
- semi-active fluid dampers.

The method of active reduction of vibration is characterised by the fact that vibration is compensated by vibration from additional external sources. Active execution systems may utilise mechanical, pneumatic, hydraulic, electromagnetic or electro-dynamic elements. Active systems may supply or absorb energy, in a specific manner, from any area of equipment. The application of active methods of reducing vibration enables the elimination of limitations present in passive systems [1-3].

2. Synthesis with selected method and analysis of obtained system

The principal objective of this work is to develop a method for determining the structure and parameters, i.e. structural and parametric synthesis, of the mechanical system model. Carrying out synthesis, viewed as modification of machinery components (as early as at the design stage) with respect to desired frequency spectrum of system vibration is aimed at obtaining a system meeting such requirements. The synthesis is double-staged – the first phase is devoted to the synthesis of the passive system without reducing vibration, whereas the second phase is the synthesis of the system with passive or active reduction (Figure 1) [4-10].

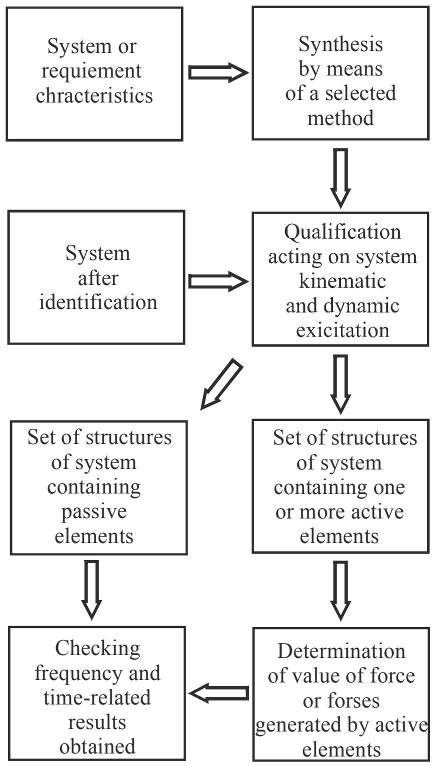

Fig. 1. Idea of synthesis of active and passive mechanical systems

2.1. Synthesis by means of a selected method

The required frequency spectrum:

$$\begin{cases} \omega_1 = 8\dfrac{rad}{s}, \quad \omega_3 = 24\dfrac{rad}{s}, \quad \omega_5 = 40\dfrac{rad}{s}, \\ \omega_0 = 0\dfrac{rad}{s}, \quad \omega_2 = 16\dfrac{rad}{s}, \quad \omega_4 = 32\dfrac{rad}{s}. \end{cases}$$

Due to the fact that the system obtained is to be branched and fixed on one side, the characteristic function is that of slowness (1). The synthesis was carried out through distribution of fractions into partial ones (2, 3) [4-11].

$$U(s)=H\left(\frac{\left(s^2+\omega_1^2\right)\left(s^2+\omega_3^2\right)\left(s^2+\omega_5^2\right)}{s\left(s^2+\omega_2^2\right)\left(s^2+\omega_4^2\right)}\right) \tag{1}$$

The synthesis made it possible to obtain the structure and values of individual parameters of the passive system without vibration.

$$U(s)\frac{1}{H}=\frac{c_1}{s}+s+\frac{1}{\frac{s}{c_2}+\frac{1}{m_2 s}}+\frac{1}{\frac{s}{c_3}+\frac{1}{m_3 s}} \tag{2}$$

$$U(s)\frac{1}{H}=\frac{225}{s}+s+\frac{1}{\frac{s}{420}+\frac{1}{1.64s}}+\frac{1}{\frac{s}{315}+\frac{1}{0.3s}} \tag{3}$$

Figure 2 presents the model of the system obtained by means of the synthesis. According to the scheme presented in Figure 1 it is necessary to specify the type and value (4) of external excitations affecting the system.

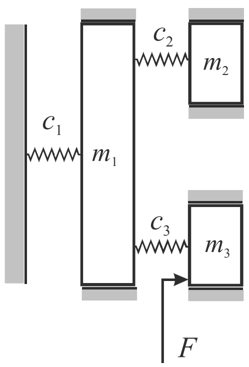

Fig. 2. Model of system with dynamic excitation

Value of external excitations:

$$F(t)=100\,\boldsymbol{sin}\,\omega t \tag{4}$$

2.2. Analysis of obtained system

Individual amplitudes are presented in Figures 3-5. The diagrams of those functions were developed on the basis of comparisons 5-7 [5, 7, 8, 12-16].

Symbols in Figs. 3-5:

$a(\omega)_1$, $a(\omega)_2$, $a(\omega)_3$ – amplitudes of system,

($a(\omega)_1=A_1$, $a(\omega)_2=A_2$, $a(\omega)_3=A_3$).

$$A_1=\frac{\left(-m_2c_3\omega^2+c_2c_3\right)(F)}{-m_1m_2m_3\omega^6+\omega^4\left(m_1m_2c_3+m_1m_3c_2+m_2m_3c_1+m_2m_3c_2+m_2m_3c_3\right)-\omega^2\left(m_1c_2c_3+m_2c_1c_3+m_3c_1c_2+m_2c_2c_3+m_3c_2c_3\right)+c_1c_2c_3} \tag{5}$$

$$A_2=\frac{\left(c_2c_3\right)(F)}{-m_1m_2m_3\omega^6+\omega^4\left(m_1m_2c_3+m_1m_3c_2+m_2m_3c_1+m_2m_3c_2+m_2m_3c_3\right)-\omega^2\left(m_1c_2c_3+m_2c_1c_3+m_3c_1c_2+m_2c_2c_3+m_3c_2c_3\right)+c_1c_2c_3} \tag{6}$$

$$A_3=\frac{\left(m_1m_2\omega^4-\omega^2\left(m_1c_2+m_2c_1+m_2c_2+m_2c_3\right)+c_1c_2+c_2c_3\right)(F)}{-m_1m_2m_3\omega^6+\omega^4\left(m_1m_2c_3+m_1m_3c_2+m_2m_3c_1+m_2m_3c_2+m_2m_3c_3\right)-\omega^2\left(m_1c_2c_3+m_2c_1c_3+m_3c_1c_2+m_2c_2c_3+m_3c_2c_3\right)+c_1c_2c_3} \tag{7}$$

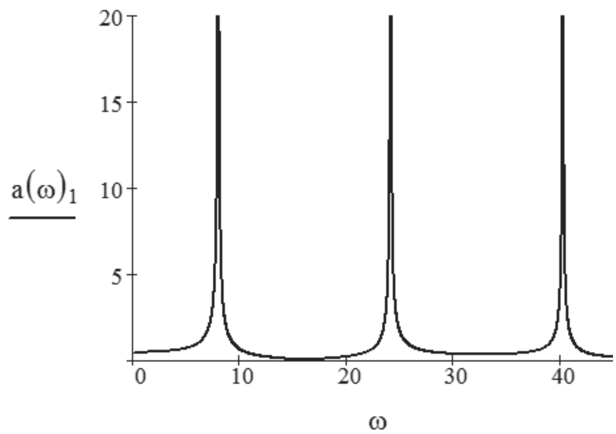

Fig. 3. Diagram of A_1 amplitude

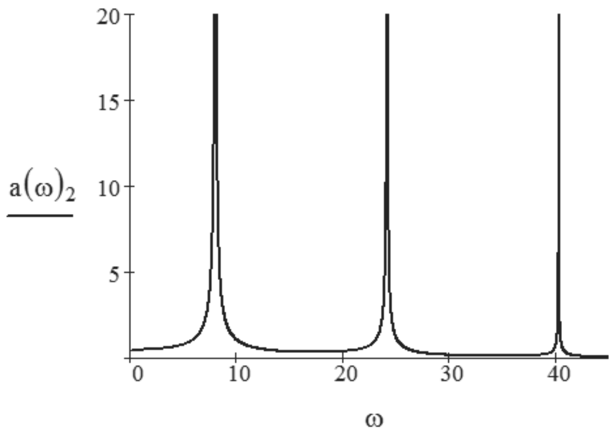

Fig. 4. Diagram of A_2 amplitude

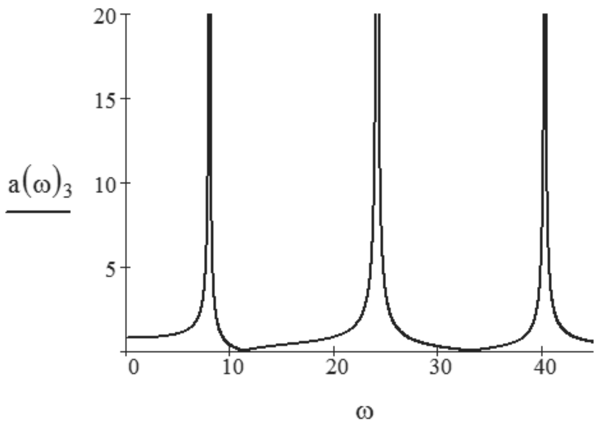

Fig. 5. Diagram of A_3 amplitude

2.3. The model of the research with passive reduction

In order to design a system characterised by required frequency values and containing passive or active elements reducing undesired vibration one should follow the instructions presented above (Figure 1). While selecting the passive reduction of vibration it is necessary to specify whether passive elements (viscous dampers) should be proportional to inertial or elastic elements. In case of the system under discussion, damping elements are proportional to elastic ones (Fig. 6). In order to determine the values of individual damping elements it is necessary to apply the following formula [5, 11]:

$$b_i = \lambda c_i \tag{8}$$

where:

b_i – damping elements

λ – modulus of proportionality $\left(0 < \lambda < \dfrac{2}{\omega_n}\right)$

ω_n – the largest value of frequency

c_i – elastic elements

λ =0.01s

$b_1 = 2.25 \dfrac{Ns}{m}$

$b_2 = 4.2 \dfrac{Ns}{m}$

$b_3 = 3.15 \dfrac{Ns}{m}$

Maximum displacements of system with passive elements are presented in formulas 9-11:

$$A_1 = \frac{\left(-m_2\omega^2 + b_2 i\omega + c_2\right)\left(b_2 i\omega + c_2\right)F}{\left(-m_1\omega^2 + \left(b_1 + b_2 + b_3\right)i\omega + c_1 + c_2 + c_3\right)\left(-m_2\omega^2 + b_2 i\omega + c_2\right)\left(-m_3\omega^2 + b_3 i\omega + c_3\right) - \left(-c_3 - b_3\omega i\right)^2\left(-m_2\omega^2 + b_2 i\omega + c_2\right) - \left(-b_2 i\omega - c_2\right)^2\left(-m_3\omega^2 + c_3 + b_3\omega i\right)} \tag{9}$$

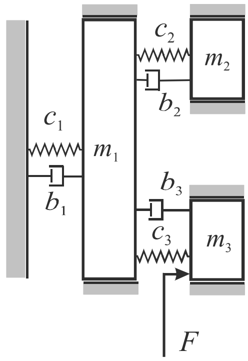

Fig. 6. The models of the system with passive elements

$$A_2 = \frac{\left(-b_2 i\omega - c_2\right)\left(-b_3 i\omega - c_3\right)F}{\left(-m_1\omega^2 + \left(b_1 + b_2 + b_3\right)i\omega + c_1 + c_2 + c_3\right)\left(-m_2\omega^2 + b_2 i\omega + c_2\right)\left(-m_3\omega^2 + b_3 i\omega + c_3\right) - \left(-c_3 - b_3\omega i\right)^2\left(-m_2\omega^2 + b_2 i\omega + c_2\right) - \left(-b_2 i\omega - c_2\right)^2\left(-m_3\omega^2 + c_3 + b_3\omega i\right)} \tag{10}$$

$$A_3 = \frac{\left((-m_1\omega^2 + (b_1 + b_2 + b_3)i\omega + c_1 + c_2 + c_3)(-m_2\omega^2 + b_2 i\omega + c_2) - (-b_2 i\omega - c_2)^2\right)F}{\left(-m_1\omega^2 + \left(b_1 + b_2 + b_3\right)i\omega + c_1 + c_2 + c_3\right)\left(-m_2\omega^2 + b_2 i\omega + c_2\right)\left(-m_3\omega^2 + b_3 i\omega + c_3\right) - \left(-c_3 - b_3 i\omega\right)^2\left(-m_2\omega^2 + b_2 i\omega + c_2\right) - \left(-b_2 i\omega - c_2\right)^2\left(-m_3\omega^2 + c_3 + b_3 i\omega\right)} \tag{11}$$

Maximum displacements of system were introduced in Figs. 7-9. Symbols in Figs. 7-9:

a(ω)$_1$, a(ω)$_2$, a(ω)$_3$ – amplitudes of system without reduction,

ap(ω)$_1$, ap(ω)$_2$, ap(ω)$_3$ – maximum displacements of system with passive reduction,

(ap(ω)$_1$=A_1, ap(ω)$_2$=A_2, ap(ω)$_3$=A_3).

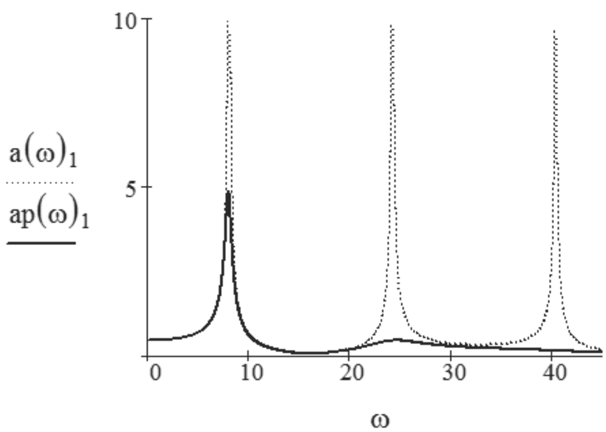

Fig. 7. Diagram of A_1 amplitude and maximum displacement

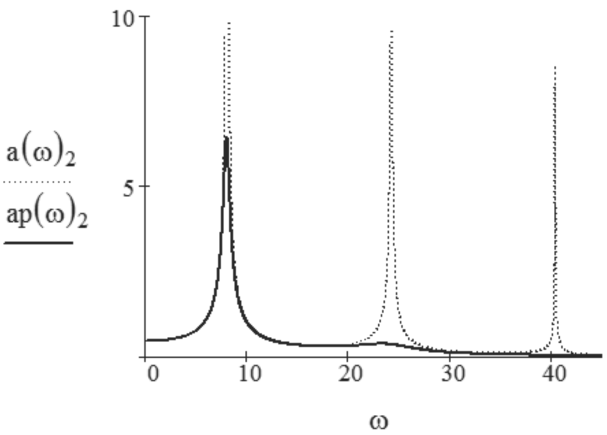

Fig. 8. Diagram of A_2 amplitude and maximum displacement

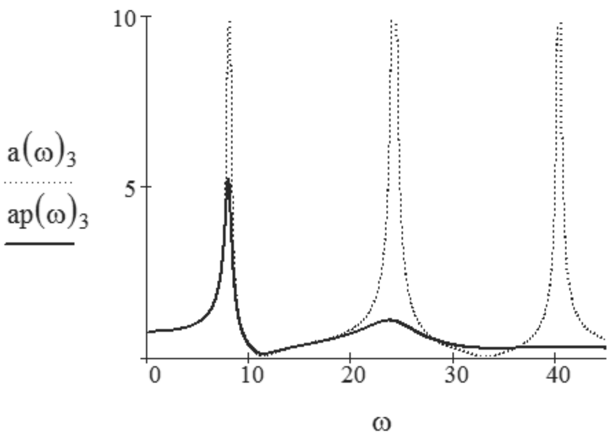

Fig. 9. Diagram of A_3 amplitude and maximum displacement

2.4. The model of the research with active reduction

In case when passive elements do not fully meet related requirements (of total reduction of vibration), designers and constructors can make use of active elements (Fig. 10). In order to determine the "location" and value of such elements, one should, in the first place, carry out synthesis or identification of the passive system without damping elements [4-6].

Solving the matrix set of equations (12), it is possible to obtain of values of individual amplitudes generated by active elements.

$$G = D \cdot A - F \quad (12)$$

where:
G – matrix of excitations generated by active elements,
D – matrix of dynamic stiffness,
A – matrix of amplitudes (approaching zero),
F – matrix of dynamic excitations.

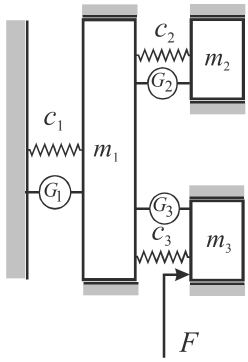

Fig. 10. The models of the system with active elements

at $\omega = 8\dfrac{rad}{s}$:

G_1= -199.85sinωt N; G_2= 0.32sinωt N; G_3= 100.06sinωt N

at $\omega = 24\dfrac{rad}{s}$:

G_1= 104.405sinωt N; G_2= 2.834sinωt N; G_3= 105.52sinωt N

at $\omega = 40\dfrac{rad}{s}$:

G_1= 113.44sinωt N; G_2= 7.87sinωt N; G_3= 101.440sinωt N

The comparison of amplitudes and displacements is introduced in Figs. 11-19. Symbols in Figs. 11-19:

a(ω)$_1$, a(ω)$_2$, a(ω)$_3$ – amplitudes of system without reduction,
aa(ω)$_1$, aa(ω)$_2$, aa(ω)$_3$ – displacements of system with active reduction,
(aa(ω)$_1$=A_1, aa(ω)$_2$=A_2, aa(ω)$_3$=A_3).

$$A_1 = \frac{\left(m_2m_3\omega^4 - \omega^2\left(m_2c_3 + m_3c_2\right) + c_2c_3\right)\left(G_1 + G_2 + G_3\right) + \left(-m_3c_2\omega^2 + c_2c_3\right)\left(-G_2\right) + \left(-m_2c_3\omega^2 + c_2c_3\right)\left(-G_3 + F\right)}{-m_1m_2m_3\omega^6 + \omega^4\left(m_1m_2c_3 + m_1m_3c_2 + m_2m_3c_1 + m_2m_3c_2 + m_2m_3c_3\right) - \omega^2\left(m_1c_2c_3 + m_2c_1c_3 + m_3c_1c_2 + m_2c_2c_3 + m_3c_2c_3\right) + c_1c_2c_3} \tag{13}$$

$$A_2 = \frac{\left(-m_3c_2\omega^2 + c_2c_3\right)\left(G_1 + G_2 + G_3\right) + \left(m_1m_3\omega^4 - \omega^2\left(m_1c_3 + m_3c_1 + m_3c_2 + m_3c_3\right) + c_1c_3 + c_2c_3\right)\left(-G_2\right) + \left(c_2c_3\right)\left(-G_3 + F\right)}{-m_1m_2m_3\omega^6 + \omega^4\left(m_1m_2c_3 + m_1m_3c_2 + m_2m_3c_1 + m_2m_3c_2 + m_2m_3c_3\right) - \omega^2\left(m_1c_2c_3 + m_2c_1c_3 + m_3c_1c_2 + m_2c_2c_3 + m_3c_2c_3\right) + c_1c_2c_3} \tag{14}$$

$$A_3 = \frac{\left(-m_2c_3\omega^2 + c_2c_3\right)\left(G_1 + G_2 + G_3\right) + \left(c_2c_3\right)\left(-G_2\right) + \left(m_1m_2\omega^4 - \omega^2\left(m_1c_2 + m_2c_1 + m_2c_2 + m_2c_3\right) + c_1c_2 + c_2c_3\right)\left(-G_3 + F\right)}{-m_1m_2m_3\omega^6 + \omega^4\left(m_1m_2c_3 + m_1m_3c_2 + m_2m_3c_1 + m_2m_3c_2 + m_2m_3c_3\right) - \omega^2\left(m_1c_2c_3 + m_2c_1c_3 + m_3c_1c_2 + m_2c_2c_3 + m_3c_2c_3\right) + c_1c_2c_3} \tag{15}$$

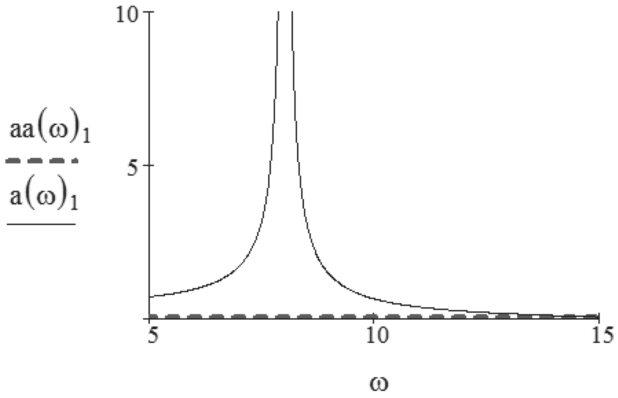

Fig. 11. Diagram of amplitude and displacement at $\omega = 8\frac{rad}{s}$

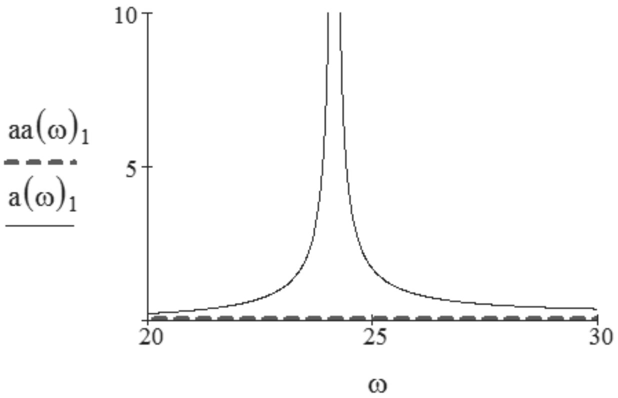

Fig. 12. Diagram of amplitude and displacement at $\omega = 24\frac{rad}{s}$

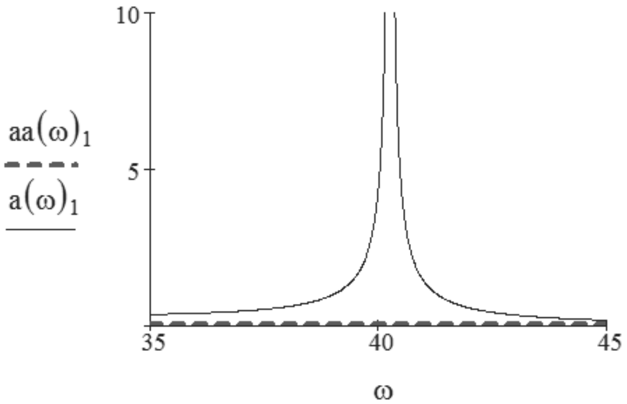

Fig. 13. Diagram of amplitude and displacement at $\omega = 40\frac{rad}{s}$

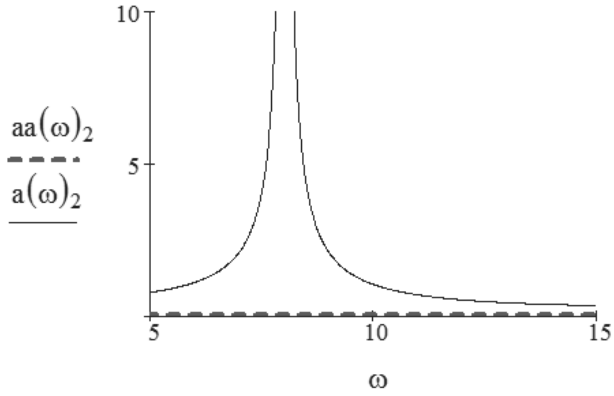

Fig. 14. Diagram of amplitude and displacement at $\omega = 8\frac{rad}{s}$

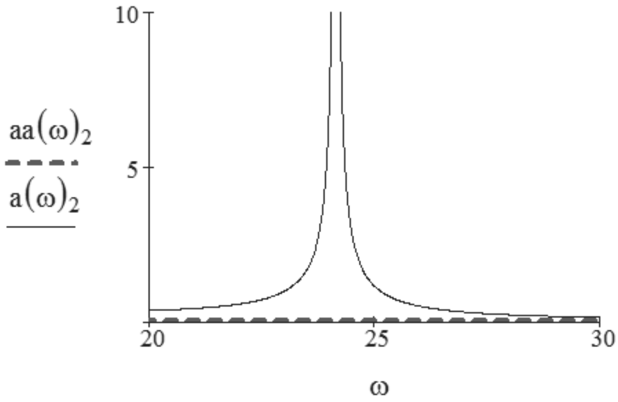

Fig. 15. Diagram of amplitude and displacement at $\omega = 24\frac{rad}{s}$

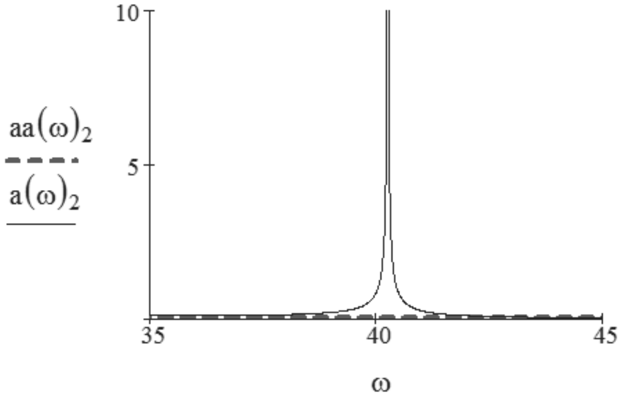

Fig. 16. Diagram of amplitude and displacement at $\omega = 40\frac{rad}{s}$

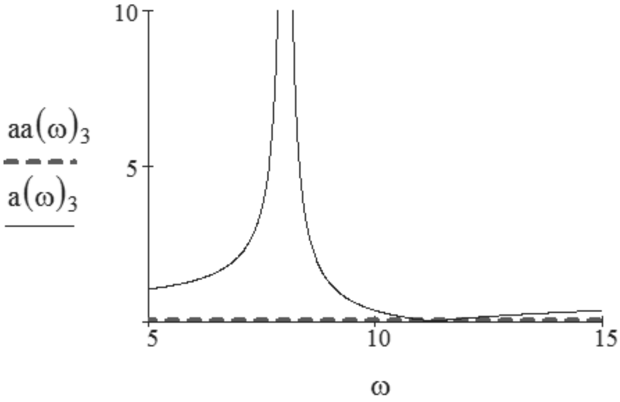

Fig. 17. Diagram of amplitude and displacement at $\omega = 8\frac{rad}{s}$

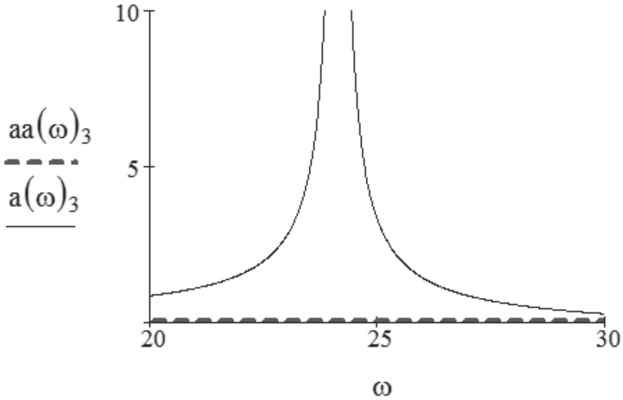

Fig. 18. Diagram of amplitude and displacement at $\omega = 24\frac{rad}{s}$

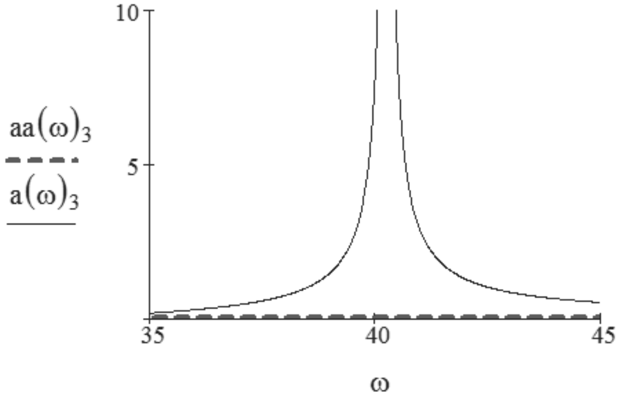

Fig. 19. Diagram of amplitude and displacement at $\omega = 40\frac{rad}{s}$

2.5. The model of the research with active elements executed by means of kinematic excitations

A very important issue is to determine the possibility of physical implementation of the active reduction of vibration. A characteristic feature of this method is the ability to modify the values of parameters of excitation generated by the elements of this subsystem in time. The elements composing these subsystems may be different i.e. mechanical, electrical etc [17]. One of the possibilities is the application of only mechanical elements in the form of kinematic excitations. In such case, the component which can change in time is displacement *"y"*, whereas the constant component is coefficient of elasticity *"c"*. In order to determine the value of displacement and that of the coefficient of elasticity, it is necessary to apply the dependence below (16) [18]:

$$y_i = \frac{K_i}{c_{ii}} \tag{16}$$

where:
K_i – values of kinematic excitations, equivalent to the values of G_i,
y_i – displacement that occurs in the specific kinematic equation,
c_{ii} – values of the elastic components that occurs in the specific kinematic equation.

Figure 20 presents a model of the system that incorporates kinematic excitations where the excitation values determined for active vibration damping are implemented.

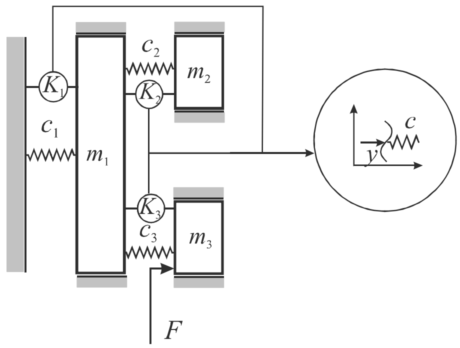

Fig. 20. A model of the system with kinematic excitations

Considering the fact that not all the results provided in Table 1 can be obtained in reality because of very high displacement values, our further analysis covered only the values from item 9. The diagrams of amplitudes and dislocations are presented in Figures 21-29.

Symbols in Figs. 21-29:
$a(\omega)_1$, $a(\omega)_2$, $a(\omega)_3$ – amplitudes of system without reduction,
$ak(\omega)_1$, $ak(\omega)_2$, $ak(\omega)_3$ – displacements of system with active reduction executed by means of kinematic excitations.

Table 1.
The values of elastic elements and dislocations in the case of system presented in Figure 20

No.	values of elastic elements	$\omega = \omega_1 = 8$ *rad/s* values of displacements	$\omega = \omega_2 = 24$ *rad/s* values of displacements	$\omega = \omega_3 = 40$ *rad/s* values of displacements
1.	$c_{11} = 0.05$ *rad/s*; $c_{22} = 0.05$ *rad/s*; $c_{33} = 0.05$ *rad/s*	$y_1 = 9.66$ *m*; $y_2 = 6.3$ *m*; $y_3 = 2001.14$ *m*	$y_1 = 21$ *m*; $y_2 = 56.6$ *m*; $y_3 = 2010.4$ *m*	$y_1 = 82.5$ *m*; $y_2 = 157.4$ *m*; $y_3 = 2028.8$ *m*
2.	$c_{11} = 0.5$ *rad/s*; $c_{22} = 0.5$ *rad/s*; $c_{33} = 0.5$ *rad/s*	$y_1 = 0.96$ *m*; $y_2 = 0.63$ *m*; $y_3 = 200.11$ *m*	$y_1 = 2.1$ *m*; $y_2 = 5.6$*m*; $y_3 = 201.04$ *m*	$y_1 = 8.2$ *m*; $y_2 = 15.7$*m*; $y_3 = 202.8$ *m*
3.	$c_{11} = 5$ *rad/s*; $c_{22} = 5$ *rad/s*; $c_{33} = 5$ *rad/s*	$y_1 = 0.09$ *m*; $y_2 = 0.06$ *m*; $y_3 = 20.01$ *m*	$y_1 = 0.21$*m*; $y_2 = 0.56$ *m*; $y_3 = 20.1$ *m*	$y_1 = 0.8$ *m*; $y_2 = 1.57$ *m*; $y_3 = 20.28$ *m*
4.	$c_{11} = 10$ *rad/s*; $c_{22} = 10$ *rad/s*; $c_{33} = 10$ *rad/s*	$y_1 = 0.05$ *m*; $y_2 = 0.03$ *m*; $y_3 = 10.01$ *m*	$y_1 = 0.1$ *m*; $y_2 = 0.28$ *m*; $y_3 = 10.05$ *m*	$y_1 = 0.4$ *m*; $y_2 = 0.78$ *m*; $y_3 = 10.14$ *m*
5.	$c_{11} = 30$ *rad/s*; $c_{22} = 30$ *rad/s*; $c_{33} = 30$ *rad/s*	$y_1 = 0.02$ *m*; $y_2 = 0.01$ *m*; $y_3 = 3.33$ *m*	$y_1 = 0.03$ *m*; $y_2 = 0.09$ *m*; $y_3 = 3.35$ *m*	$y_1 = 0.14$ *m*; $y_2 = 0.26$ *m*; $y_3 = 3.38$ *m*
6.	$c_{11} = 50$ *rad/s*; $c_{22} = 50$ *rad/s*; $c_{33} = 50$ *rad/s*	$y_1 = 0.01$ *m*; $y_2 = 0.006$ *m*; $y_3 = 2.001$ *m*	$y_1 = 0.02$ *m*; $y_2 = 0.05$ *m*; $y_3 = 2.01$ *m*	$y_1 = 0.08$ *m*; $y_2 = 0.16$ *m*; $y_3 = 2.03$ *m*
7.	$c_{11} = 100$ *rad/s*; $c_{22} = 100$ *rad/s*; $c_{33} = 100$ *rad/s*	$y_1 = 0.005$ *m*; $y_2 = 0.003$ *m*; $y_3 = 1$ *m*	$y_1 = 0.01$ *m*; $y_2 = 0.03$ *m*; $y_3 = 1.005$ *m*	$y_1 = 0.04$ *m*; $y_2 = 0.07$ *m*; $y_3 = 1.01$ *m*
8.	$c_{11} = 200$ *rad/s*; $c_{22} = 200$ *rad/s*; $c_{33} = 200$ *rad/s*	$y_1 = 0.002$ *m*; $y_2 = 0.001$ *m*; $y_3 = 0.5$ *m*	$y_1 = 0.005$ *m*; $y_2 = 0.014$ *m*; $y_3 = 0.5$ *m*	$y_1 = 0.02$ *m*; $y_2 = 0.04$ *m*; $y_3 = 0.5$ *m*
9.	$c_{11} = 300$ *rad/s*; $c_{22} = 300$ *rad/s*; $c_{33} = 300$ *rad/s*	$y_1 = 0.001$ *m*; $y_2 = 0.001$ *m*; $y_3 = 0.3$ *m*	$y_1 = 0.003$ *m*; $y_2 = 0.009$ *m*; $y_3 = 0.3$ *m*	$y_1 = 0.01$ *m*; $y_2 = 0.02$ *m*; $y_3 = 0.3$ *m*

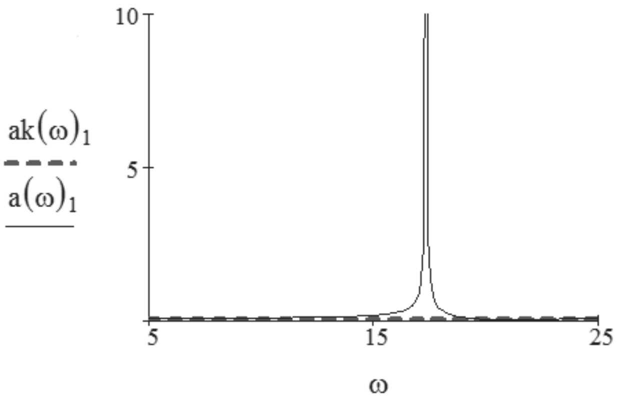

Fig. 21. Diagram of amplitude and displacement A_1

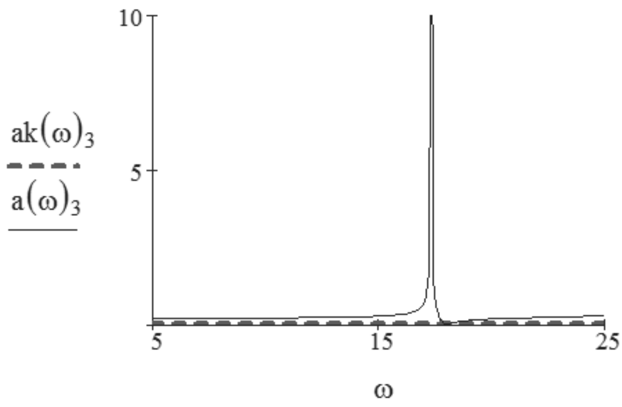

Fig. 23. Diagram of amplitude and displacement A_3

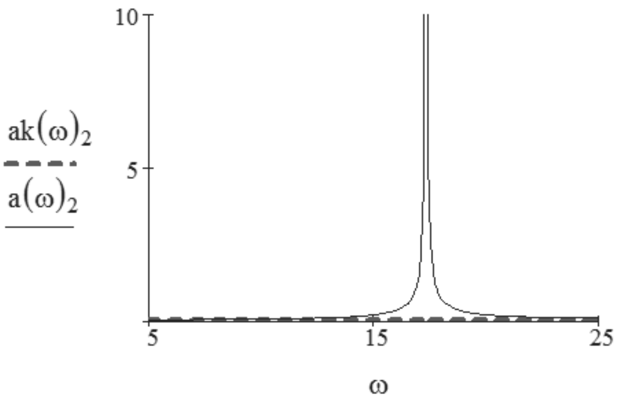

Fig. 22. Diagram of amplitude and displacement A_2

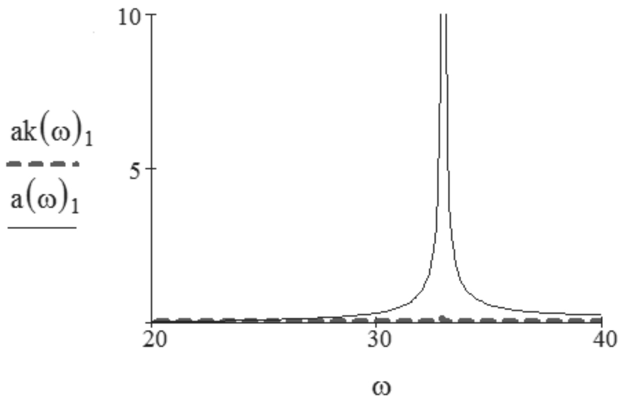

Fig. 24. Diagram of amplitude and displacement A_1

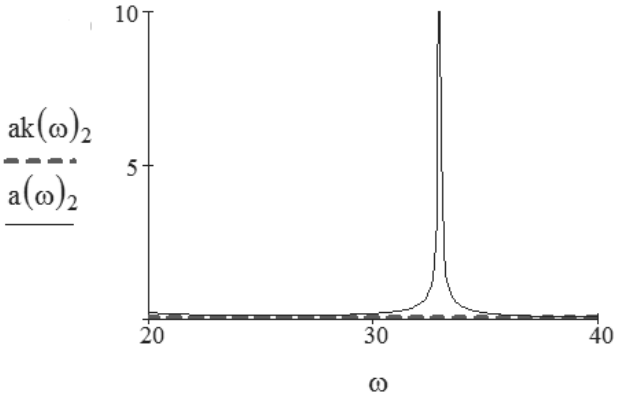

Fig. 25. Diagram of amplitude and displacement A_2

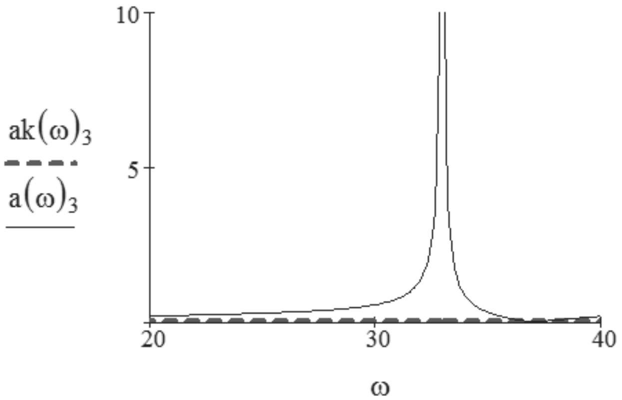

Fig. 26. Diagram of amplitude and displacement A_3

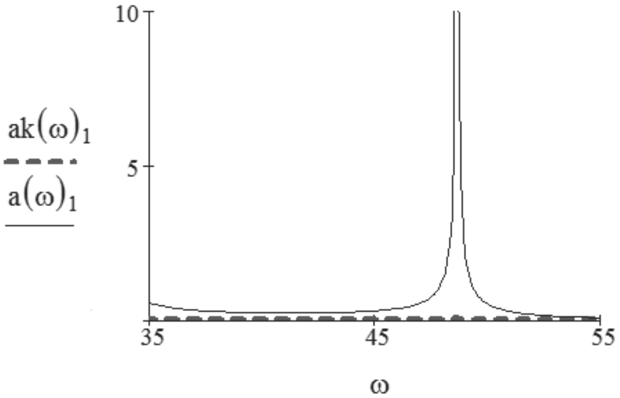

Fig. 27. Diagram of amplitude and displacement A_1

3. Conclusions

The work presents the comparison of the application of passive and active methods for reduction of vibration of discrete branched systems vibrating longitudinally. In order to compare these two methods it was necessary to illustrate the course of the

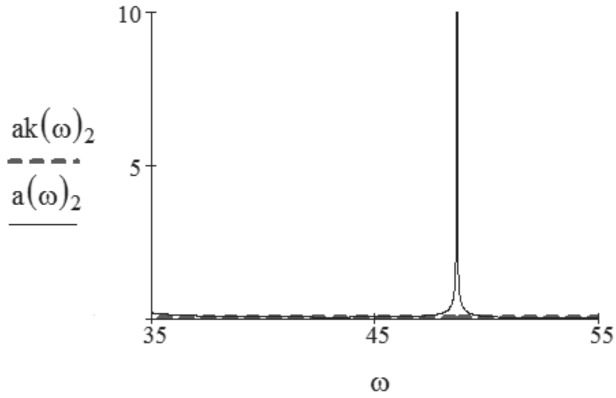

Fig. 28. Diagram of amplitude and displacement A_2

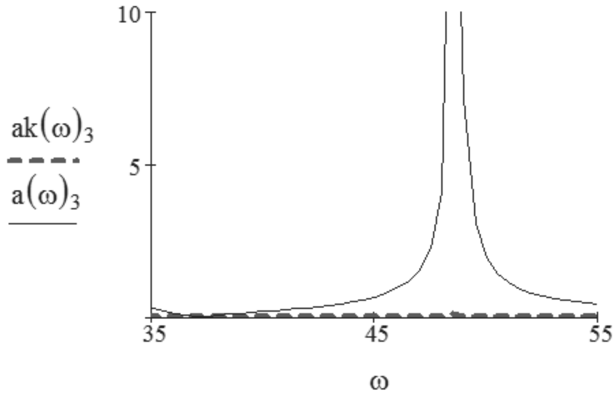

Fig. 29. Diagram of amplitude and displacement A_3

functions in the form of amplitudes and the maximum displacement of the system with and without the application of the elimination of vibration. The diagrams reveal that the passive method implemented with viscous dampers does not always bring desired results, in particularly in case of low frequency vibration. In turn, the application of the active methods enables the total reduction of vibration. The approach presented in this work makes it possible to apply measures aimed to eliminate undesired action of equipment and machinery as early as at the design stage.

Another issue under consideration was the physical implementation of the active subsystems and their effect on the basic system. The presented diagrams reveal that the application of only mechanical elements in the form of kinematic excitations does not fulfil its task. The action of the subsystem causes the entire change of the basic system.. The change is related to the modification of the frequency values of the basic system's internal vibration and for this reason this system cannot be regarded as meeting the initial requirements.

Acknowledgements

This work has been conducted as a part of the research project No. N N502 452139 supported by Polish Ministry of Scientific Research and Higher Education in 2010-2013.

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