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Reverse task of passive and active mechanical system in torsional vibrations

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Analysis and modelling

ABSTRACT

Purpose: The main aim of this paper is to develop a method for finding structure and parameters, i.e. a structural and parameter synthesis, of an active model of a viscous damper mechanical system in vibrations. The aim is to perfect the synthesis seen as modification at the sub-assembly design level in relation to the required spectrum of vibration frequency of the system.

Design/methodology/approach: With complex systems classic design is very time consuming and it does not always produce satisfactory results. Therefore, it is necessary to use other design methods, such as the inverse task, which is called synthesis. It is searching for a system structure, together with elements value, which realizes the required frequency characteristics.

Findings: Using the active elements allows complete elimination of the oscillations. The conducted analysis show that it is not necessary to use both the active and passive elements, as using only active elements produces the same results.

Research limitations/implications: The scope of discussion is reverse task of mechanical system in torsional vibrations including passive and active elements, but for this type of systems, such approach is sufficient.

Practical implications: The methods of reverse task and analysis can be base of design and construct for this type of mechanic systems.

Originality/value: Thank to the approach, introduced in this paper, can be conducted as early as during the designing of future functions of the system as well as during the construction of the system. Using method and obtained results can be value for designers of mechanical systems with elements reducing vibrations.

Keywords: Process systems design; Polar graphs; Structural numbers; Reduction of vibrations

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1. Introduction

Vibrations are very common in our surroundings. Their harmfulness is connected with their influence on incorrect operation and behaviour of equipment and machines. Negative influence on human body is also observed. This is why it is so important to reduce unwanted vibrations. Equipment constructors and designers face the task of both counteracting the incorrect operation or usage of the newly constructed and adapting already existing machines to their requirements.

There are many ways of preventing excessive vibrations effecting machine sub-assemblies and elements. The main vibration reducing unit types are: passive, semi-active and active. The characteristic feature of the passive units is that they only offer the possibility to periodically disperse or store energy and it is not possible to change their parameters later on. Semi-active units include passive elements but, unlike in passive units, they can be changed. In active units, vibrations are compensated with vibrations from other, external, sources [1,2].

The classic approach to mechanical systems design is looking for such element values that would meet the requirements. A system that does not meet the requirements is modified and analyzed anew. This is the trial and error method. Such design method is successful only in case of simple systems. With complex systems such design is very time consuming and it does not always produce satisfactory results. Therefore, it is necessary to use other design methods, such as the inverse task, which is called synthesis. It is searching for a system structure, together with elements value, which realizes the required frequency characteristics [3-11].

2. Design of mechanical system in torsional vibrations by using synthesis method

The main aim of this paper is to develop a method for finding structure and parameters, i.e. a structural and parameter synthesis, of an active model of a viscous damper mechanical system in torsional vibrations. The aim is to perfect the synthesis seen as modification at the sub-assembly design level in relation to the required spectrum of vibration frequency of the system. Such definition of the problem requires application of the synthesis methods, in categories appropriate for the class of active systems with damping, described with polar graphs and structural numbers. The synthesis will have two stages. The first stage will include a synthesis of the passive system, and then a synthesis of the active system (dashed line), system with damping or the active system with damping (continuous line) (Figure 1)[12-19].

2.1. The system of the research

The structures of systems after the synthesis (continued fraction expansion) was introduced in Figure 2.

The required frequency spectrum:

$$\begin{cases} \omega_0 = 0\dfrac{rad}{s}, \quad \omega_2 = 100\dfrac{rad}{s}, \quad \omega_4 = 200\dfrac{rad}{s}. \\ \omega_1 = 50\dfrac{rad}{s}, \quad \omega_3 = 150\dfrac{rad}{s}, \end{cases}$$

Synthesis by means of continued fraction expansion method**:**

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

$$U(s) = \frac{s^5+50000s^3+400000000s}{s^4+25000s^2+56250000} = \\ = s + \cfrac{1}{\cfrac{s}{25000} + \cfrac{1}{2.2s + \cfrac{1}{\cfrac{s}{19551} + \cfrac{1}{3.9s}}}} \tag{2}$$

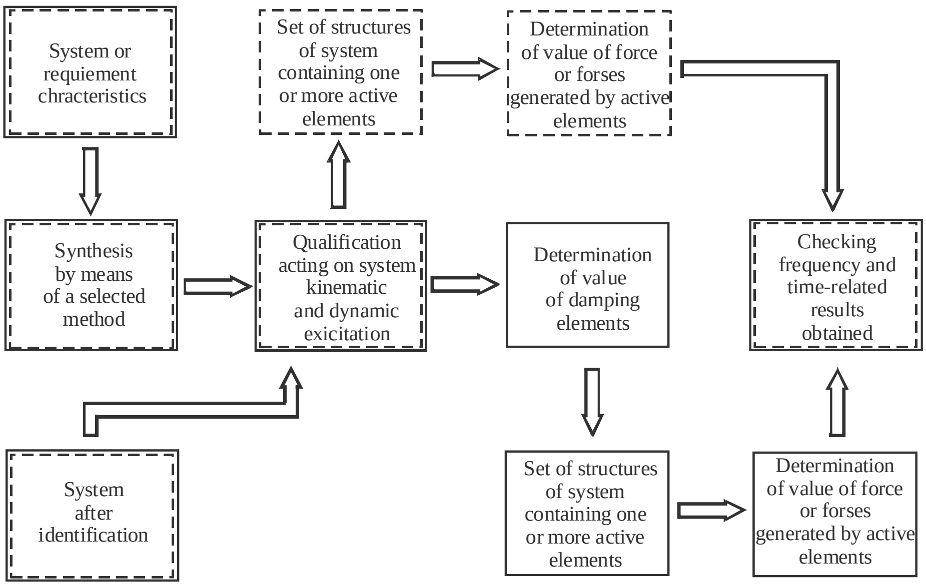

Fig. 1. Idea of synthesis of mechanical systems

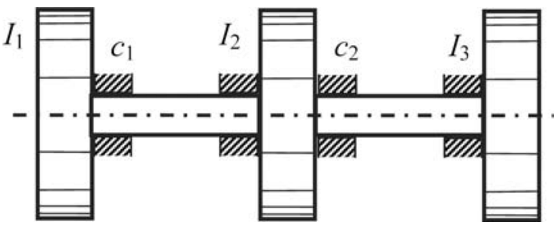

Fig. 2. Model of system after synthesis

This system was weighted dynamic excitation (Figure 3). Polar graph of this system was introduced in Figure 4.

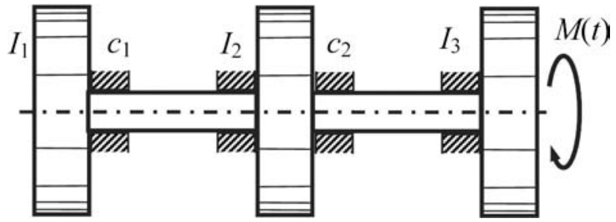

Fig. 3. Model of system with dynamic excitation

The above elements of polar graph are numbered according to the following standard:

$[1]-I_1p^2 \rightarrow I_1 = 1\,kgm^2 \Rightarrow$ inertial element,

$[2]-I_2p^2 \rightarrow I_2 = 2.2\,kgm^2 \Rightarrow$ inertial element,

$[3]-I_3p^2 \rightarrow I_3 = 3.9\,kgm^2 \Rightarrow$ inertial element,

$[4]-c_1 \rightarrow c_1 = 25000\,\frac{Nm}{rad} \Rightarrow$ elastic element,

$[5]-c_2 \rightarrow c_2 = 19551\,\frac{Nm}{rad} \Rightarrow$ elastic element,

$[6]-M \rightarrow M = 300\sin\omega t\ Nm \Rightarrow$ dynamic excitation.

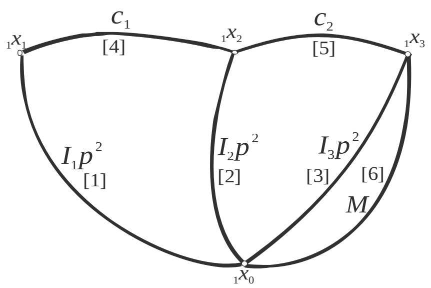

Fig. 4. Polar graph of the systems from Figure 3

The amplitudes of system (Fig.3) are introduced in Figures 5-7.

$$A_1 = \frac{c_1c_2M}{-I_1I_2I_3\omega^6 + \omega^4\left(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1\right) - \omega^2\left(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2\right)} \tag{3}$$

$$A_2 = \frac{\left(-I_1c_2\omega^2 + c_1c_2\right)M}{-I_1I_2I_3\omega^6 + \omega^4\left(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1\right) - \omega^2\left(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2\right)} \tag{4}$$

$$A_3 = \frac{\left(I_1I_2\omega^4 - \omega^2\left(I_1c_1 + I_1c_2 + I_2c_1\right) + c_1c_2\right)M}{-I_1I_2I_3\omega^6 + \omega^4\left(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1\right) - \omega^2\left(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2\right)} \tag{5}$$

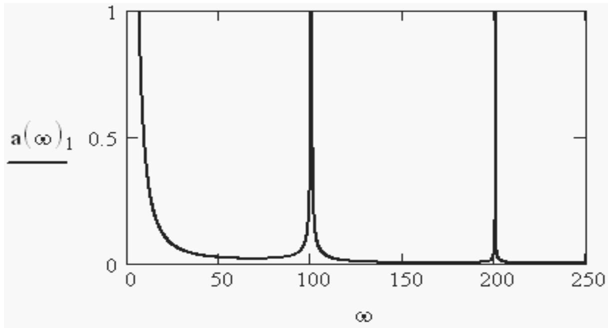

Fig. 5. Diagram of A_1 amplitude

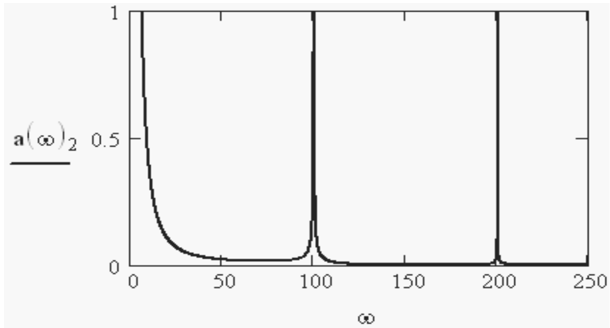

Fig. 6. Diagram of A_2 amplitude

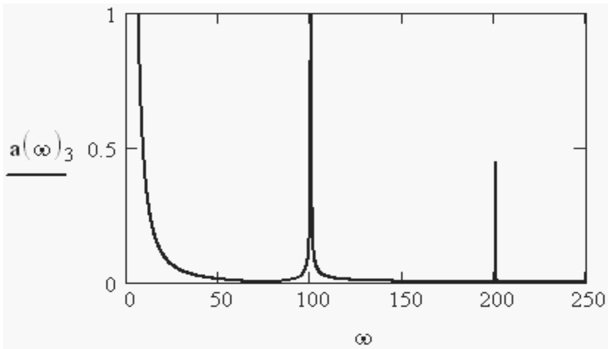

Fig. 7. Diagram of A_3 amplitude

2.2. The system with passive elements

System with passive elements reducing vibrations was introduced in Figure 8 (polar graph in Figure 9):

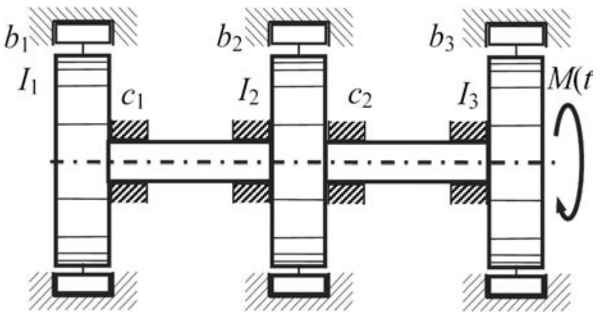

Fig. 8. Model of the system with passive elements

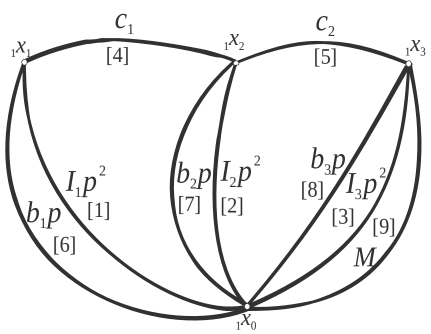

Fig. 9. Polar graph of the systems from Figure 8

A general formula for value of damping [8], when damping is proportional to inertial element, is as follows:

$$b_i = 2hI_i \tag{6}$$

where:
b_i - damping elements
h - parameter $(0 < h < \omega_1)$
I_i - inertial elements

Using formula (6) it is possible to mark the values of damping elements:

at $h = 0.5$; $b_1 = 1\frac{Nms}{rad}$ $b_2 = 2.2\frac{Nms}{rad}$ $b_3 = 3.9\frac{Nms}{rad}$

Maximum displacements of system were introduced in Figures 10-12. Symbols in Figures 10-12:

$a(\omega)_1$, $a(\omega)_2$, $a(\omega)_3$ – amplitudes of system without reduction
$ap(\omega)_1$, $ap(\omega)_2$, $ap(\omega)_3$ – maximum displacements of system with passive reduction

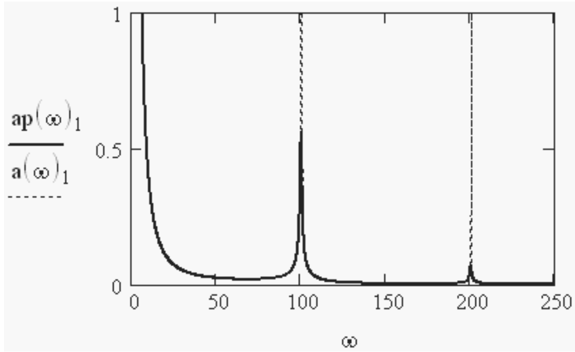

Fig. 10. Diagram of A_1 amplitude and maximum displacement

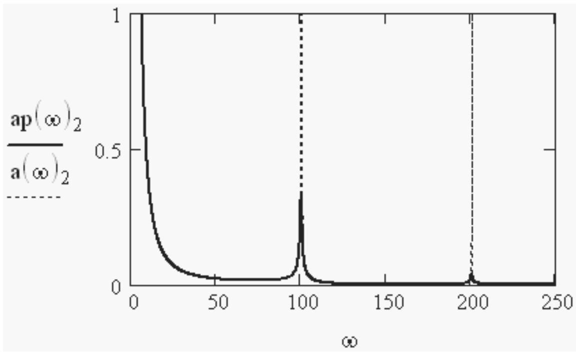

Fig. 11. Diagram of A_2 amplitude and maximum displacement

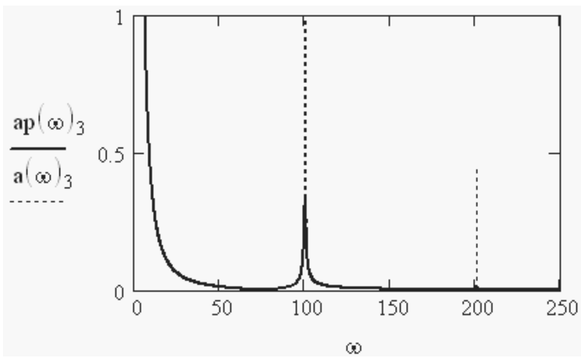

Fig. 12. Diagram of A_3 amplitude and maximum displacement

2.3. The system with active elements

System with active elements reducing vibrations was introduced in Figure 13 (polar graph in Figure 14):

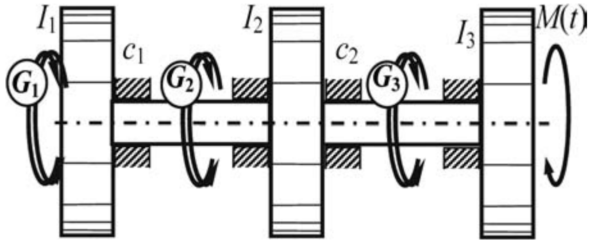

Fig. 13. Model of the system with active elements

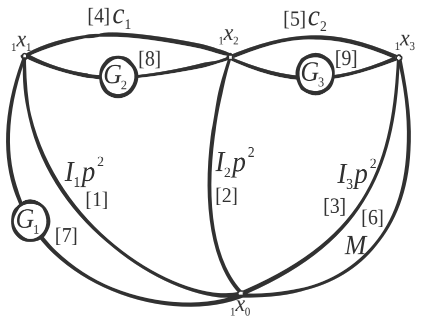

Fig. 14. Polar graph of the systems from Figure 13

Using the theory of polar graphs and their relation to structural numbers [20], it is possible to determine the values of amplitudes of forces generated by active elements.

$$A_1 = \frac{\left(\left(\frac{\partial D(\omega)}{\partial[1]}\right)(-[7]+[8])\right) + \left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[1]};\frac{\partial D(\omega)}{\partial[2]}\right)(-[8]+[9])\right) + \left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[1]};\frac{\partial D(\omega)}{\partial[3]}\right)(-[9]+[6])\right)}{D(\omega)} =$$
$$= \frac{\left(I_2I_3\omega^4 - \omega^2(I_2c_2 + I_3c_1 + I_3c_2) + c_1c_2\right)(-G_1+G_2) + \left(-I_3c_1\omega^2 + c_1c_2\right)(-G_2+G_3) + c_1c_2(-G_3+M)}{-I_1I_2I_3\omega^6 + \omega^4(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1) - \omega^2(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2)} \qquad (7)$$

$$A_2 = \frac{\left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[1]};\frac{\partial D(\omega)}{\partial[2]}\right)(-[7]+[8])\right) + \left(\left(\frac{\partial D(\omega)}{\partial[2]}\right)(-[8]+[9])\right) + \left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[2]};\frac{\partial D(\omega)}{\partial[3]}\right)(-[9]+[6])\right)}{D(\omega)} =$$
$$= \frac{\left(-I_3c_1\omega^2 + c_1c_2\right)(-G_1+G_2) + \left(-I_1c_2\omega^2 + c_1c_2\right)(-G_3+M) + \left(I_1I_3\omega^4 - \omega^2(I_1c_2 + I_3c_1) + c_1c_2\right)(-G_2+G_3)}{-I_1I_2I_3\omega^6 + \omega^4(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1) - \omega^2(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2)} \qquad (8)$$

$$A_3 = \frac{\left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[1]};\frac{\partial D(\omega)}{\partial[3]}\right)(-[7]+[8])\right) + \left(\underset{z}{Sim}\left(\frac{\partial D(\omega)}{\partial[2]};\frac{\partial D(\omega)}{\partial[3]}\right)(-[8]+[9])\right) + \left(\left(\frac{\partial D(\omega)}{\partial[3]}\right)(-[9]+[6])\right)}{D(\omega)} =$$
$$= \frac{c_1c_2(-G_1+G_2) + \left(-I_1c_2\omega^2 + c_1c_2\right)(-G_2+G_3) + \left(I_1I_2\omega^4 - \omega^2(I_1c_1 + I_1c_2 + I_2c_1) + c_1c_2\right)(-G_3+M)}{-I_1I_2I_3\omega^6 + \omega^4(I_1I_2c_2 + I_1I_3c_1 + I_1I_3c_2 + I_2I_3c_1) - \omega^2(I_1c_1c_2 + I_2c_1c_2 + I_3c_1c_2)} \qquad (9)$$

Solving the equations (7-9), it is possible to obtain of values of individual amplitudes generated by active elements (Table 1).

The comparison of amplitudes and maximum displacements is introduced in Figures 15-23. Symbols in Figures 15-23:

$a(\omega)_1$, $a(\omega)_2$, $a(\omega)_3$ – amplitudes of system without reduction
$ap(\omega)_1$, $ap(\omega)_2$, $ap(\omega)_3$ – maximum displacements of system with passive reduction
$aa(\omega)_1$, $aa(\omega)_2$, $aa(\omega)_3$ – amplitudes of system with active reduction

Table 1.
Values of individual amplitudes generated by active elements

at $\omega=\omega_0$	at $\omega=\omega_2$	at $\omega=\omega_4$
G_1= 300sinωt *Nm*	G_1= 513sinωt *Nm*	G_1= 1152sinωt *Nm*
G_2= 300sinωt *Nm*	G_2= 483sinωt *Nm*	G_2= 1032sinωt *Nm*
G_3= 300sinωt *Nm*	G_3= 417sinωt *Nm*	G_3= 768sinωt *Nm*

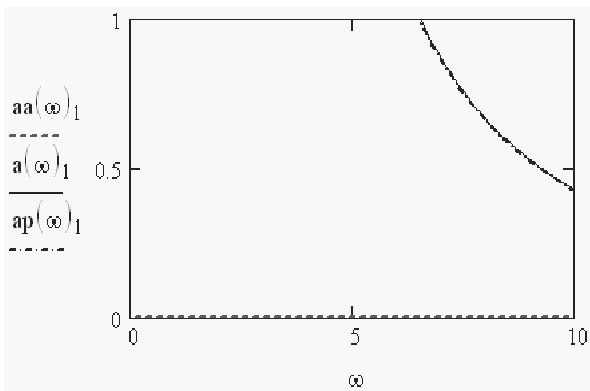

Fig. 15. Diagram of amplitude and maximum displacement $\omega=\omega_1$

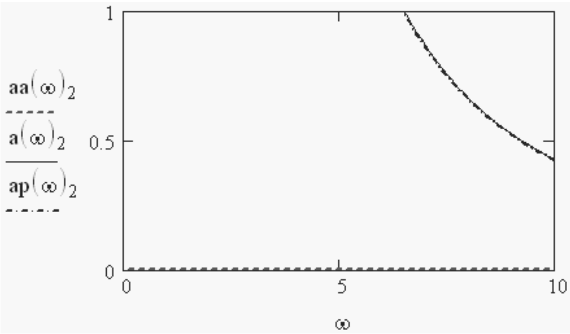

Fig. 16. Diagram of amplitude and maximum displacement $\omega=\omega_1$

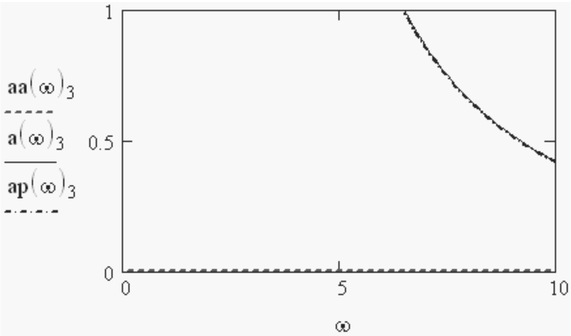

Fig. 17. Diagram of amplitude and maximum displacement $\omega=\omega_1$

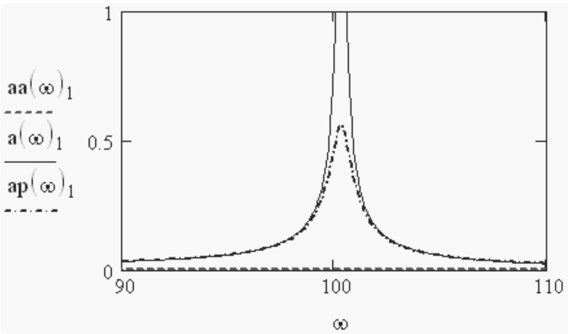

Fig. 18. Diagram of amplitude and maximum displacement $\omega=\omega_2$

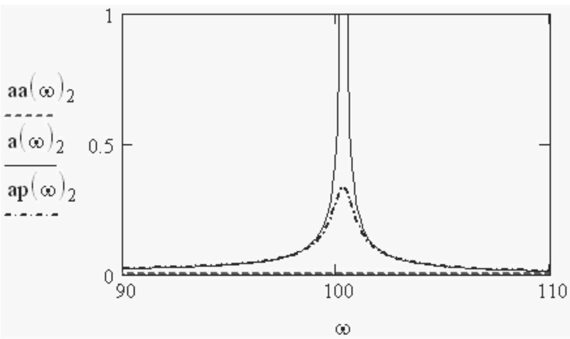

Fig. 19. Diagram of amplitude and maximum displacement $\omega=\omega_2$

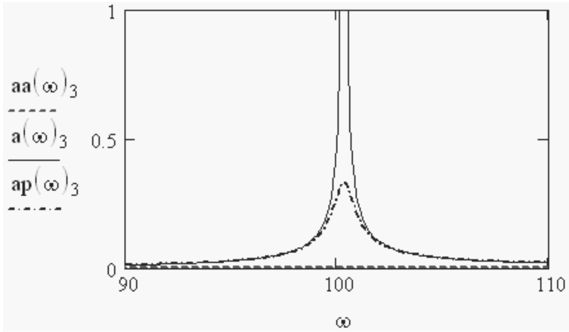

Fig. 20. Diagram of amplitude and maximum displacement $\omega=\omega_2$

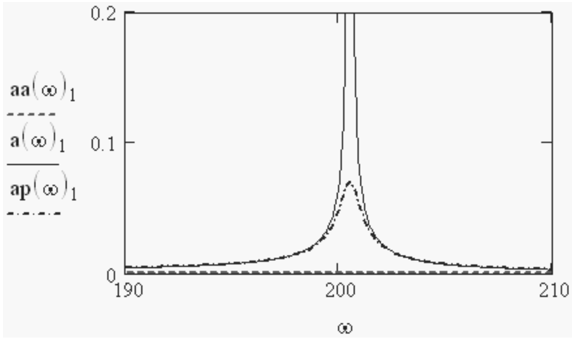

Fig. 21. Diagram of amplitude and maximum displacement $\omega=\omega_3$

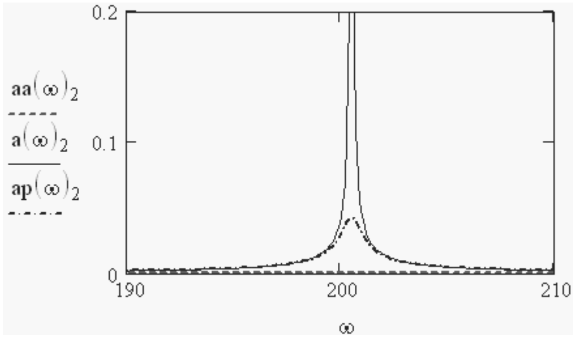

Fig. 22. Diagram of amplitude and maximum displacement $\omega=\omega_3$

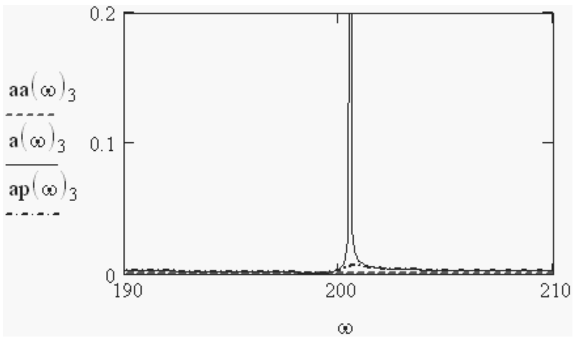

Fig. 23. Diagram of amplitude and maximum displacement $\omega=\omega_3$

2.4. The system with passive and active elements

System with passive and active elements reducing vibrations was introduced in Figure 24 (polar graph in Figure 25):

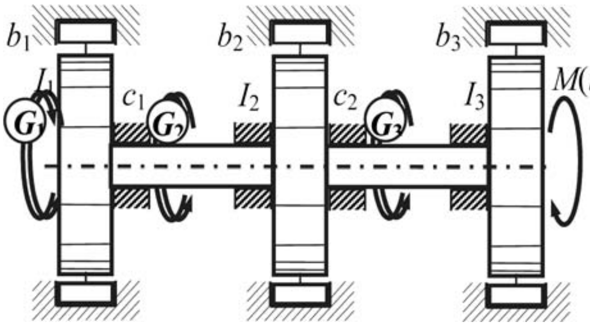

Fig. 24. Model of the system with passive and active elements

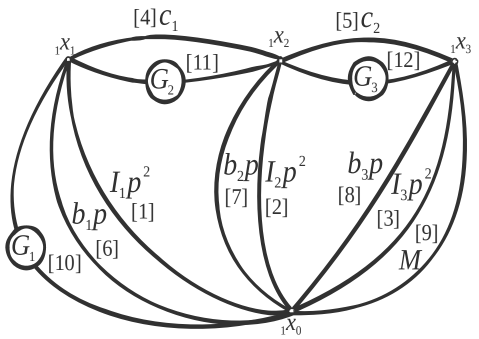

Fig. 25. Polar graph of the systems from Fig. 24

The comparison of amplitudes is introduced in Figures 26-34. Symbols in Figures 26-34:

$a(\omega)_1$, $a(\omega)_2$, $a(\omega)_3$ – amplitudes of system without reduction
$aap(\omega)_1$, $aap(\omega)_2$, $aap(\omega)_3$ – amplitudes of system with active and passive reduction

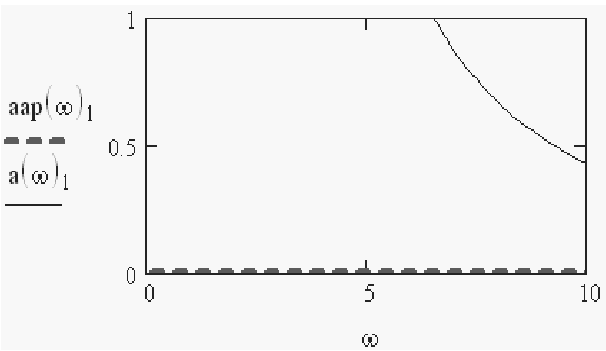

Fig. 26. Diagram of amplitude and maximum displacement $\omega=\omega_1$

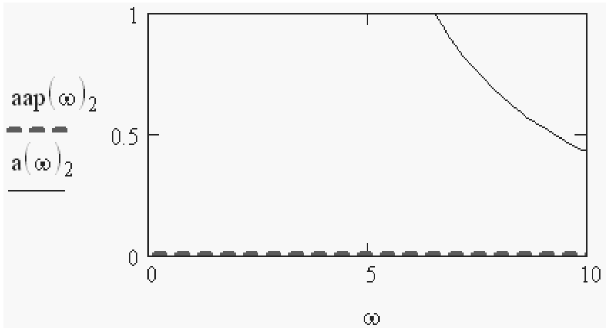

Fig. 27. Diagram of amplitude and maximum displacement $\omega=\omega_1$

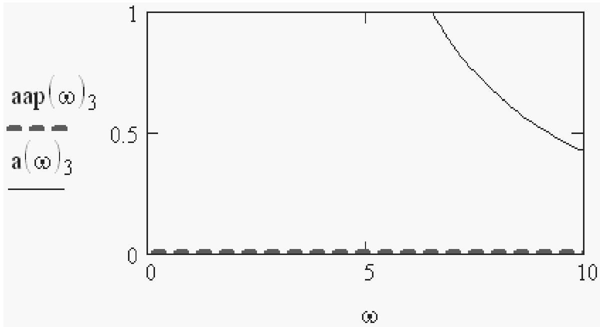

Fig. 28. Diagram of amplitude and maximum displacement $\omega=\omega_1$

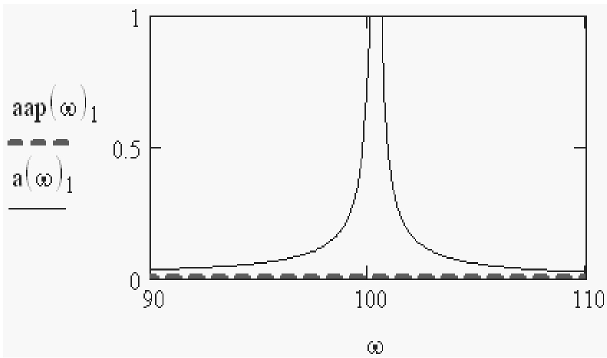

Fig. 29. Diagram of amplitude and maximum displacement $\omega=\omega_2$

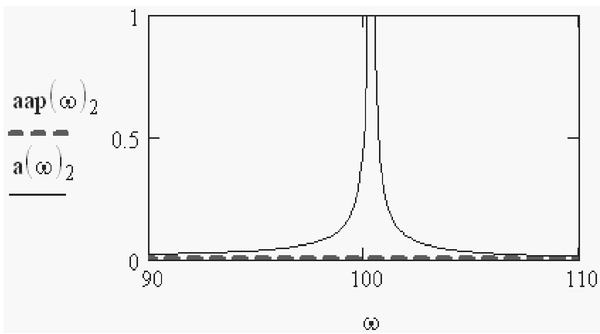

Fig. 30. Diagram of amplitude and maximum displacement $\omega=\omega_2$

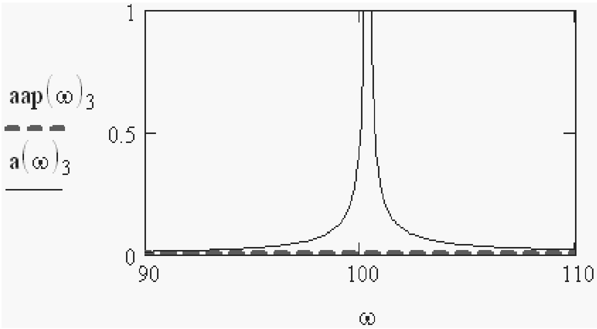

Fig. 31. Diagram of amplitude and maximum displacement $\omega=\omega_2$

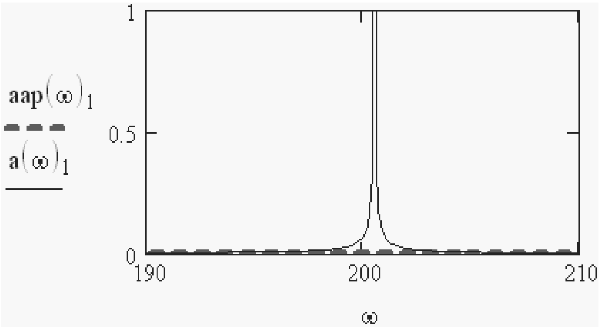

Fig. 32. Diagram of amplitude and maximum displacement $\omega=\omega_3$

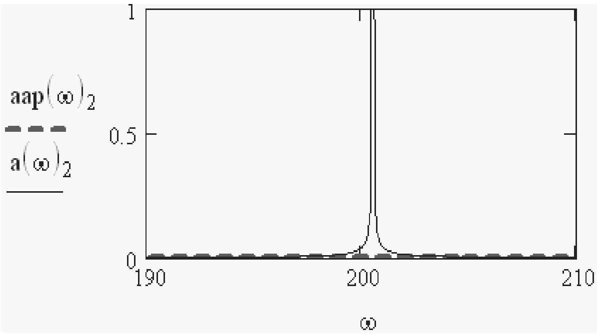

Fig. 33. Diagram of amplitude and maximum displacement $\omega=\omega_3$

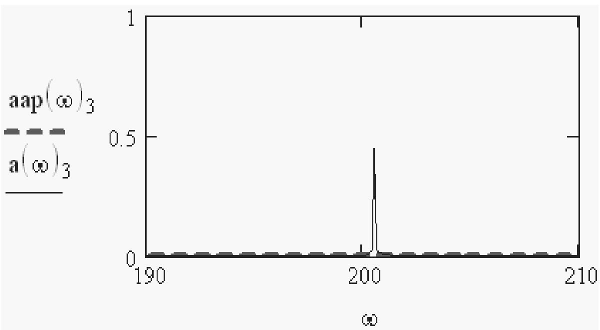

Fig.34. Diagram of amplitude and maximum displacement $\omega=\omega_3$

3. Conclusions

Using active ways of vibration reduction allows overcoming the passive systems limitations. As the resented diagrams show, systems containing passive elements do not fully reduce the vibration of the inertial elements of the system. Using the active elements allows complete elimination of the vibrations. The conducted analysis show that it is not necessary to use both the active and passive elements, as using only active elements produces the same results.

Acknowledgements

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