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IMPACT AND PARTICLE BUFFERED VIBRATION ABSORBERS OPTIMIZATION AND DESIGN

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Abstract. Passive, broadband targeted energy transfer refers to the one-way directed transfer of energy from a primary subsystem to a nonlinear attachment; this phenomenon is realized in damped, coupled, essentially nonlinear impact or particle dynamic vibration absorber (DVA). An impact damper is a passive control device which takes the form of a freely moving mass, constrained by stops attached to the structure under control, i.e. the primary structure. The damping results from the exchange of momentum during impacts between the mass and the stops as the structure vibrates. A particle-based damping system can overcome some limitations of ordinary DVA by using particles as the damping medium and inter- particle interaction as the damping mechanism. Large damping at such family constructions of DVA's does not bring to destruction an elastic DVA element over in critical cases, when working frequency approaches own frequency of DVA, or when the transitional process of acceleration of rotating machines is slow enough and DVA's has time to collect large amplitudes of vibrations.

The primary structure is modelled as a spring-mass system. In this paper, an efficient numerical approach based on the theoretical-experimental method is proposed to maximize the minimal damping of modes in a prescribed frequency range for general viscous tuned-mass systems. Methods of decomposition and numerical synthesis are considered on the basis of the adaptive schemes. The influence of dynamic vibration absorbers and basic design elastic and damping properties is under discussion. A technique is developed to give the optimal DVA's for the elimination of excessive vibration in sinusoidal forced rotating system. It is found that the buffered impact damper not only significantly reduces the accelerations, contact force and the associated noise generated by a collision but also enhances the level of vibration control.

The interaction of DVA's and basic design elastic and damping properties is under discussion. One task of this work is to analyze parameters identification of the dynamic vibration absorber and the basic structure. The discrete-continue models of machines dynamics of such rotating machines as water pump with the attachment of particle DVA's and elongated element with multi mass impact DVA's are offered. A technique is developed to give the optimal DVA's for the elimination of excessive vibration in harmonic stochastic and impact loaded systems.

Introduction

The impact damping method has two limitations. First, damping does not occur at low frequencies where the acceleration of the container is lower than that of gravity because the particles lack sufficient energy to begin colliding with each other and instead they move as one lumped mass. Second, the damping capabilities are dependent on the quantity and mass of free particles available for collisions. The tendency of particles to self-assemble into a packed configuration under vibration can reduce the availability of free particles and therefore decrease damping effectiveness.

A tuned mass damper (TMD), or dynamic vibration absorber (DVA), is found to be an efficient, reliable and low-cost suppression device for vibrations caused by harmonic or narrow-band excitations. In DVA design the stiffness and the damping ratio can be determined by balancing the two fixed points in the frequency response [1], in the case of harmonic excitation, or by minimizing the mean-square response under the random excitation, or by balancing the poles of system. Most leading text books on mechanical vibrations discuss the basic equations of DVA's to some extend, e.g. [1–3]. Among the pioneering publications providing an in-depth theoretical treatment are those by Ormondroyd and Den Hartog [4] and Den Hartog [5]. For linear DVA's a closed theory is available, but due to the large number of system parameters and varying technical applications with different requirements no unique solution exists. Generally, a significant influence of damping on the vibration reduction performance is observed.

The problem of attaching DVA to a discrete multi-degree-of-freedom or continuous structure has been outlined in many papers and monographs by Bishop and Welbourn [6], Warburton [7], Hunt [8], Snowdon [9], Korenev and Rabinovic [10] and Aida et al. [11] to name but a few. Nonlinear DVA have been investigated by Kolovsky [12], Kauderer [13], Pipes [14], Roberson [15]. The article [16] of Ibrahim presents a comprehensive assessment of nonlinear DVA's in the absence of active control means.

In [17] an improved scheme is proposed for identifying the time of contact and calculation of the state variables after impact. This scheme avoids false detection of collisions and embodies collisions or contacts with infinitesimally small differences in velocities. Detailed experiments with a horizontal impact damper explain in [17] the general performance and the resonance vibration of the integrated system, which occurs at a frequency, which is different from the original resonance frequency.

An impact damping system can overcome some limitations by impact as the damping medium and impact mass interaction as the damping mechanism. The paper contemplates the provision of DVA or any number of such absorbers. Such originally designed absorbers reduce vibration selectively in maximum vibration mode without introducing vibration in other modes. For example, the final result is achieved by DVA at far less expense compared to the cost needed to replace the machine foundation with a new, sufficiently massive one. In [18–20] the particle DVA's are presented.

In order to determine the optimal parameters of an absorber the need for complete modeling is obvious. Present research has developed a modern prediction and control methodology, based on a complex continuum theory and the application of special frequency characteristics of structures. The numerical schemes (NS) row for the complex vibroexcitated construction and methods of decomposition and the NS synthesis are considered in our paper on the basis of new methods of modal synthesis [21–24]. The DVA designed in accordance with our proposals also has the advantage that it can be constructed such that it has a wide-range vibration absorption property. Such originally designed absorbers reduce vibration selectively in maximum mode of vibration without introducing vibration in other modes.

Similar in a mathematical plan tasks are examined in [25]. Here basic task: maximally effectively to pass energy to the container with details which are processed.

Dynamic equations

Let us consider condensed model of DVA - main system. In Fig. 1 the impact mass type DVA is presented: an additional impact mass in container with elastic barrier elements.

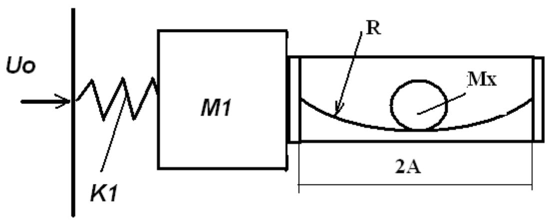


Fig. 1. *Pendulum type DVA with the additional elements*

The system of equations in the condensed rangy is obtained:

$$
\begin{gathered}
m_1 \frac{d^2 u_1}{dt^2} + k_1(u_1 - u_0) + k_A(u_1 - u_A) - {}^{m_{X1}}\!/_{R_{X1}} (u_{X1} - u_A) + k_{X1}F_1(u_1 - u_{X1}) \\
- \ldots - {}^{m_{XN}}\!/_{R_{XN}} (u_{XN} - u_A) + k_{XN} F_N (u_1 - u_{XN}) = F(t), \\
m_{X1} \frac{d^2 u_{X1}}{dt^2} + {}^{m_{X1}}\!/_{R_{X1}} (u_{X1} - u_A) - k_{X1}F_1(u_1 - u_{X1}) = 0, \\
\ldots \\
m_{XN} \frac{d^2 u_{XN}}{dt^2} + {}^{m_{XN}}\!/_{R_{XN}} (u_{XN} - u_A) - k_{XN} F_N (u_1 - u_{XN}) = 0 .
\end{gathered} \tag{1}
$$

Here an arbitrary number N of DVA's is considered. Parameters m_1, k_1 of the prime system may be found by means of FEM or experimentally. The nonlinear functions are:

$$F_i = -K_{vi}(x_i - A_i) \quad |x_x| > A_i, \qquad F_i = 0 \quad |x_i| < A_i \quad ; \; F(t) = a\sin(\omega t), \tag{2}$$

where A are clearans and K_{vi} is boundary elements rigidity.

Numerical results, optimization

DVA's are appropriately optimized by genetic algorithms near the beam first eigen-frequency f_R. The evaluation function is:

$$CiL = Max(u_1(f)), \alpha f_R < \mathrm{f} < \beta f_R. \tag{3}$$

The process and results of optimization for the DVA (Fig. 1) is presented in Fig. 2 for different DVA's masses.

N = 2121

Dx .263E-01 Dx2 .265E-01 DG .544E-02 Ax .150E+02 CiL .404E-01
fx .996E+00 fx2 .879E+00 Ekx .959E-05 Mx .750E+00

N = 5585

Dx .173E-01 Dx2 .746E-02 DG .855E-01 Ax .150E+02 CiL .273E-01
fx .892E+00 fx2 .100E+01 Ekx .193E-03 Mx .100E+01

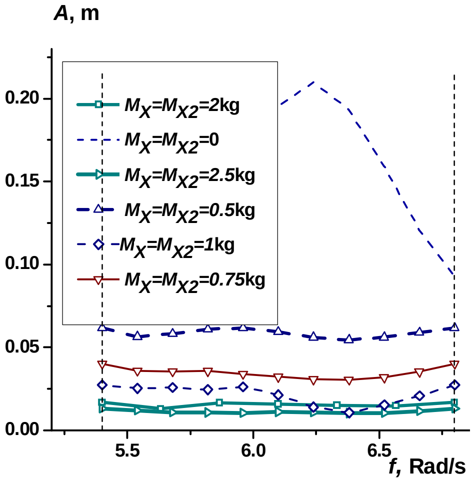
N = 1602

Dx .275E-01 Dx2 .167E-01 DG .664E-01 Ax .150E+02 CiL .168E-01
fx .100E+01 fx2 .885E+00 Ekx .601E-04 Mx .200E+01

N = 5844

Dx .151E-01 Dx2 .208E-01 DG .577E-01 Ax .150E+02 CiL .132E-01
fx .911E+00 fx2 .100E+01 Ekx .494E-02 Mx .250E+01

a)



b)

***Fig. 2.** The process (a) and results (b) of optimization for the two DVA's*

Here 4 parameters of optimization are used: f_x, f_{x2} are DVA's eigenfrequencies; D_x, D_{x2} are proportional viscous damping (added to all equations terms $k_{Xi}D_{Xi}\,{}^{du_i}\!/_{dt}$). In the all numerical examples the prime system mass is m_1 = 10 kg, the prime system eigenfrequency f_R = 1 Hz = 6.28 Rad/s, the proportional damping – D_1 = 0.03. For system with two dangerous frequency intervals the grate number of DVA's may be used (Fig. 3). For $N_A = 4$ the better result may be seen.

Let's consider now the DVA with 3 different impact masses in one container (Fig. 4). The system of equations is now:

$$\begin{gathered}
m_1\frac{d^2u_1}{dt^2} + k_1(u_1-u_0) + k_A(u_1-u_A) - {}^{m_{X1}}\!/_{R_{X1}}(u_{X1}-u_A) + k_{X1}F_1(u_1-u_{X1}) - \\
-\ldots - {}^{m_{XN}}\!/_{R_{XN}}(u_{XN}-u_A) + k_{XN}F_N(u_1-u_{XN}) = F(t), \\
m_{X1}\frac{d^2u_{X1}}{dt^2} + {}^{m_{X1}}\!/_{R_{X1}}(u_{X1}-u_A) - k_{X1}F_1(u_1-u_{X1}) + F_{12}(u_{X1},u_{X2}) + F_{13}(u_{X1},u_{X3}) = 0, \\
m_{X2}\frac{d^2u_{X2}}{dt^2} + {}^{m_{X2}}\!/_{R_X}(u_{XN}-u_A) - k_X F_N(u_1-u_{X2}) - F_{12}(u_{X1},u_{X2}) + F_{23}(u_{X2},u_{X3}) = 0, \\
m_{X3}\frac{d^2u_{X3}}{dt^2} + {}^{m_{X2}}\!/_{R_X}(u_{XN}-u_A) - F_{13}(u_{X1},u_{X3}) - F_{23}(u_{X2},u_{X3}) = 0.
\end{gathered} \tag{4}$$

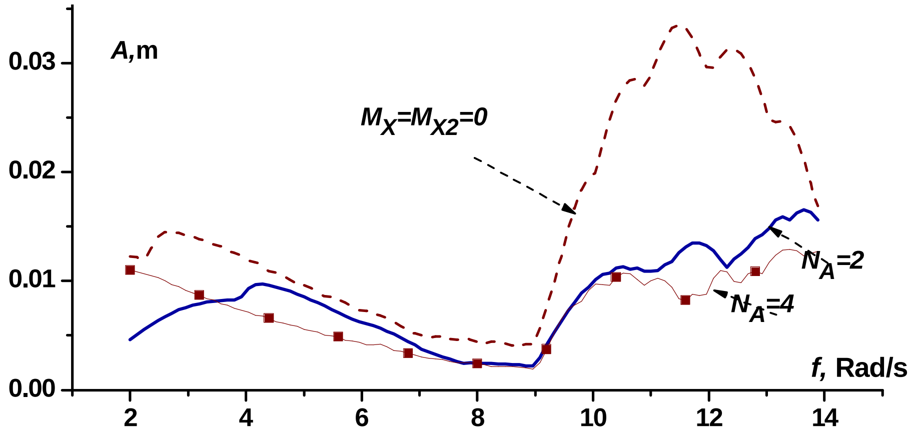


Fig. 3. *The results of optimization for system with two frequency intervals by number of DVA's* $N_A = 2.4$

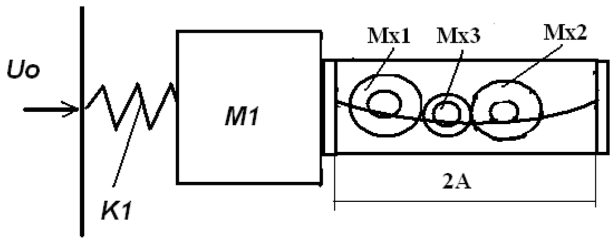


Fig. 4. *DVA with 3 different impact masses*

Here two DVA's are considered. Parameters m_1, k_1 of the prime system may be found by means of FEM or experimentally. The nonlinear functions are:

$$F_i = -K_{vi}(x_i - A_i) \quad |x_x| > A_i, \qquad F_i = 0 \quad |x_i| < A_i; \qquad F(t) = a\sin(\omega t). \tag{5}$$

Coordinates x_1, x_2, x_3 of the impact masses and the differences between this coordinates x_1, x_3 and x_2, x_3 are presented in Fig. 5.

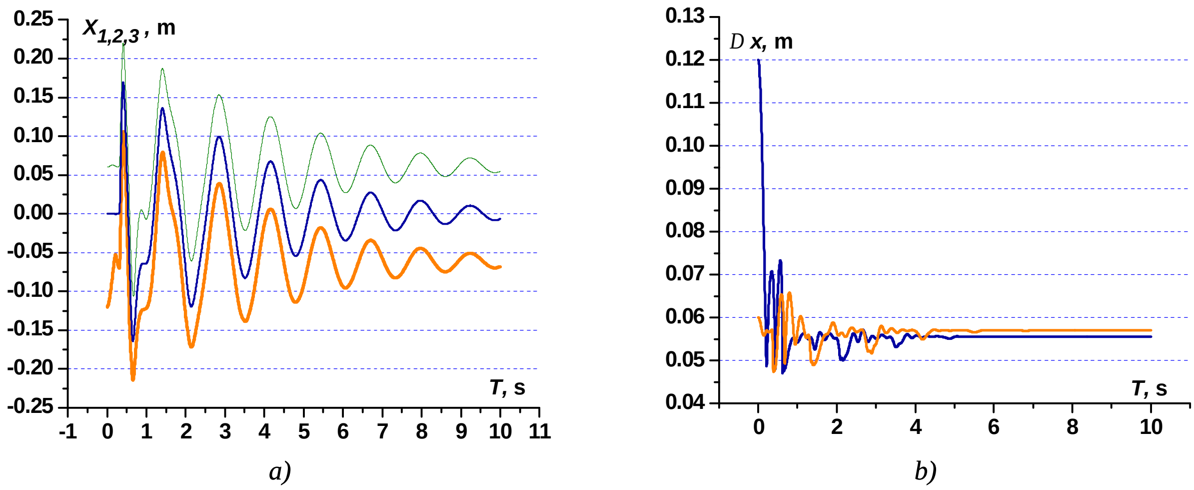


Fig. 5. *Coordinates* x_1, x_2, x_3 *of the impact masses (a);*
the differences between this coordinates x_1, x_3 *and* x_2, x_3 *(b)*

Here A – clearans and K_{vi} – boundary elements rigidity. The nonlinear functions $F_{13}(u_{X1}, u_{X3})$, $F_{23}(u_{X2}, u_{X3})$ of DVA's masses interaction may be defined analogously:

$$F_{13} = F_{13}(x_1 - x_3) \quad |x_1 - x_3| < R_1 + R_3, \qquad F_{13} = 0 \quad |x_1 - x_3| > R_1 + R_3,$$

$$F_{23} = F_{13}(x_2 - x_3) \quad |x_2 - x_3| < R_2 + R_3, \qquad F_{23} = 0 \quad |x_2 - x_3| > R_2 + R_3 .$$

In Fig. 6 the results of DVA's application is shown.

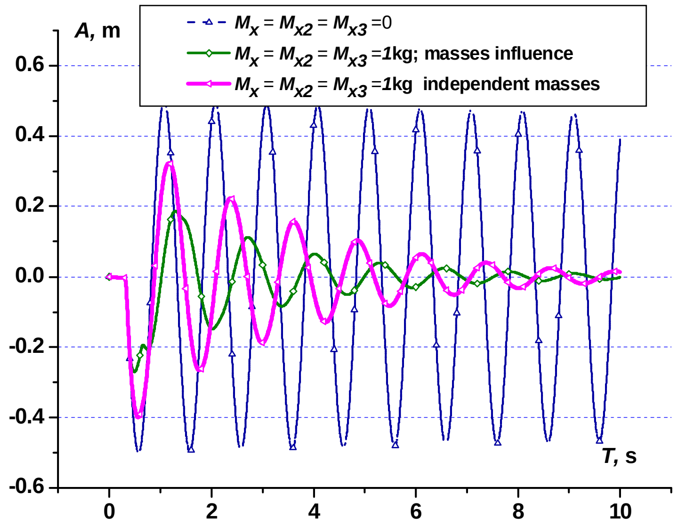


Fig. 6. *Results of DVA's application*

The 3 mass impact DVA seems to be better than independent 3 DVA's with the same masses.

Here the optimization in the real time is done. Let us consider the optimization of this DVA's by criterion:

$$CiL = Max(x_1(t)), t > t_P . \tag{6}$$

The process of geometrical DVA's parameters evolution for different stage of impulse loading and different base system damping is shown in Fig. 7.

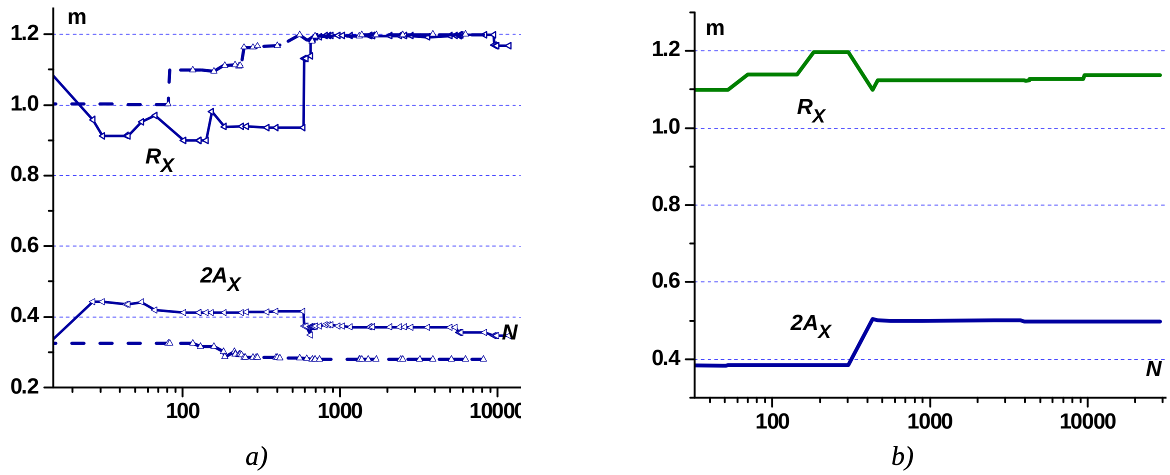


Fig. 7. *Process of geometrical DVA's parameters evolution for different stage of impulse loading: (a) – D_1 = 0.03; (b) – D_1 = 0.003*

In Fig. 8 results of one-mass DVA and 3 mass DVA optimization. The one-mass DVA is worse than 3-mass. The upper results are achieved with the Boltzman contact forces approximation (Fig. 9)

$$y = 1. + \frac{A_1 - A_2}{1 + \exp\left({x - x_0}/{dx} \right)} .$$

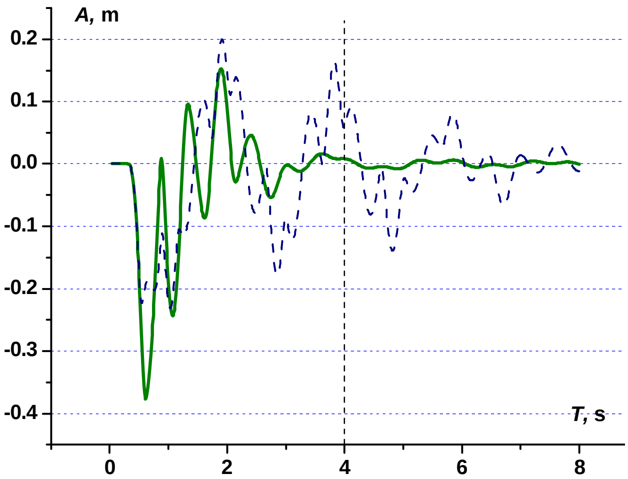


Fig. 8. *Results of one-mass DVA (dash line) and 3 mass DVA optimization*

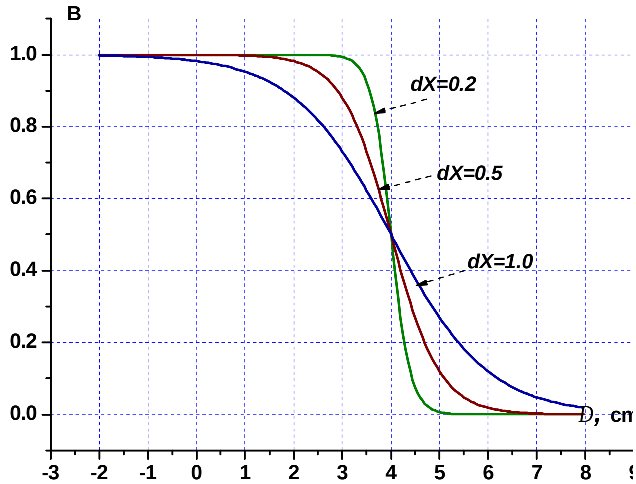


Fig. 9. *Boltzman contact forces approximation*

Here $A_1=1$, $A_2=2$, $X_0=4$. $x = 2X_0 - \Delta$. Δ - distance between centers of rolling masses, X_0 – width of contact zone.

Let us consider an another impact DVA with all impact masses I one container (Fig. 10).

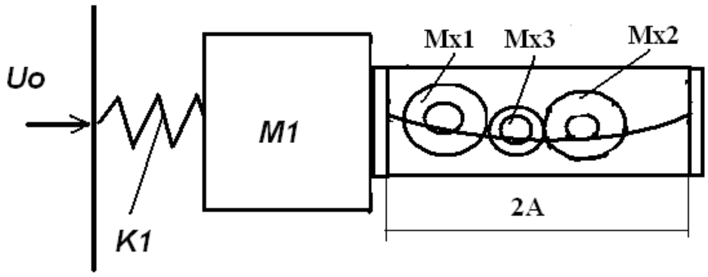


Fig. 10. *DVA with 3 different impact masses*

The system of equations is now:

$$
\begin{gathered}
m_1 \frac{d^2 u_1}{dt^2} + k_1(u_1 - u_0) + k_A(u_1 - u_A) - \frac{m_{X1}}{R_{X1}}(u_{X1} - u_A) + k_{X1}F_1(u_1 - u_{X1}) - \\
- \ldots - \frac{m_{XN}}{R_{XN}}(u_{XN} - u_A) + k_{XN}F_N(u_1 - u_{XN}) = F(t), \\
m_{X1}\frac{d^2 u_{X1}}{dt^2} + \frac{m_{X1}}{R_{X1}}(u_{X1} - u_A) - k_{X1}F_1(u_1 - u_{X1}) + F_{12}(u_{X1}, u_{X2}) + F_{13}(u_{X1}, u_{X3}) = 0, \\
m_{X2}\frac{d^2 u_{X2}}{dt^2} + \frac{m_{X2}}{R_X}(u_{XN} - u_A) - k_X F_N(u_1 - u_{X2}) - F_{12}(u_{X1}, u_{X2}) + F_{23}(u_{X2}, u_{X3}) = 0, \\
m_{X3}\frac{d^2 u_{X3}}{dt^2} + \frac{m_{X2}}{R_X}(u_{XN} - u_A) - F_{13}(u_{X1}, u_{X3}) - F_{23}(u_{X2}, u_{X3}) = 0.
\end{gathered} \tag{7}
$$

Here 3 DVA's are considered. Parameters m_1, k_1 of the prime system may be found by means of FEM or experimentally. The nonlinear functions are:

$$F_i = -K_{vi}(x_i - A_i) \quad |x_x| > A_i, \qquad F_i = 0 \quad |x_i| < A_i; \qquad F(t) = a\sin(\omega t). \tag{8}$$

Here A – clearans and K_{vi} – boundary elements rigidity. The nonlinear functions $F_{13}(u_{X1}, u_{X3})$, $F_{23}(u_{X2}, u_{X3})$ of DVA's masses interaction may be defined analogously.

$$F_{13} = F_{13}(x_1 - x_3) \quad |x_1 - x_3| < R_1 + R_3, \qquad F_{13} = 0 \quad |x_1 - x_3| > R_1 + R_3,$$
$$F_{23} = F_{13}(x_2 - x_3) \quad |x_2 - x_3| < R_2 + R_3, \qquad F_{23} = 0 \quad |x_2 - x_3| > R_2 + R_3.$$

In Fig. 11 coordinates x_1, x_2, x_3 of the impact masses are shown.

In Fig. 12 the results of DVA's application are shown.

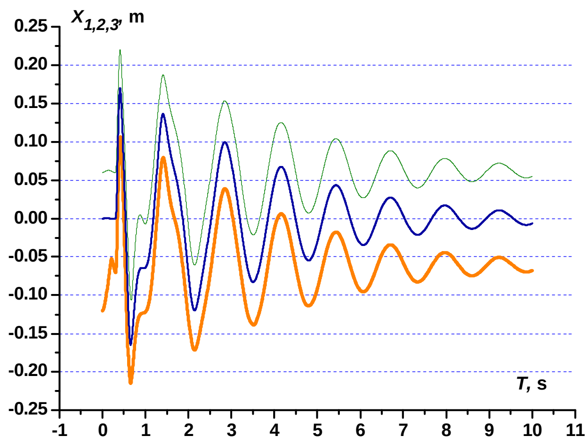


Fig. 11. *Coordinates x_1, x_2, x_3 of the impact masses*

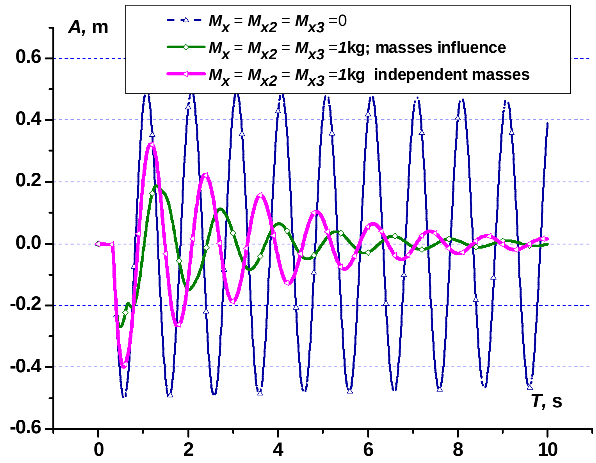
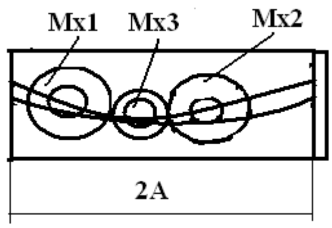


Fig. 12. *Results of 3-mass DVA's application*

The 3 mass impact DVA seems to be better than independent 3 DVA's with the same masses.

Let us consider new 3-mass DVA with the impact masses on the plates of different radius of curvature (Fig. 13).

Fig. 13. *DVA with 3 different impact masses on the plates of different radius*



Here the curvatures of DVA's plates are different. That prevents them to move synchronous. Consider 3 cases of optimization: 1) the simultaneous optimization by impact and harmonic loading; 2) optimization by harmonic loading; 3) optimization by impact loading. Results of optimization are presented in Fig. 14.

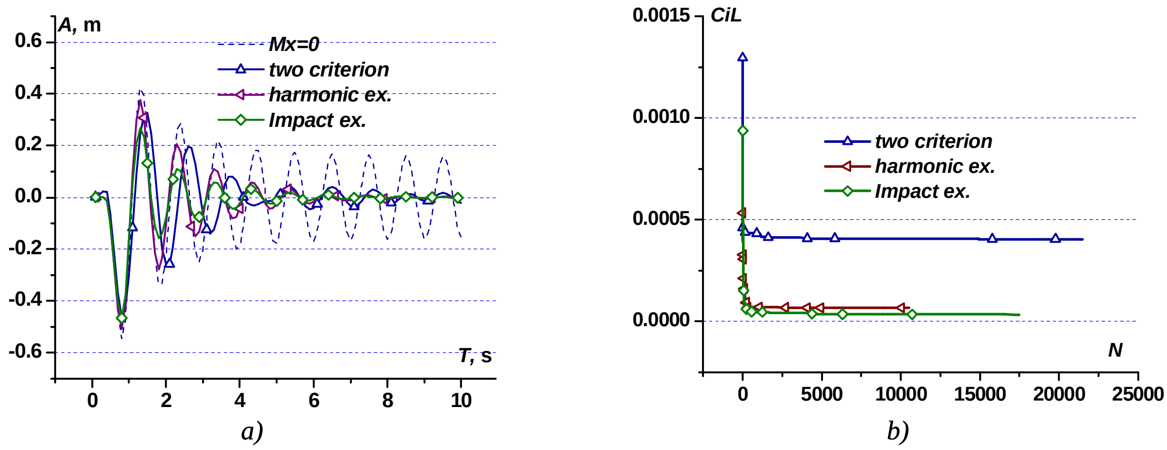
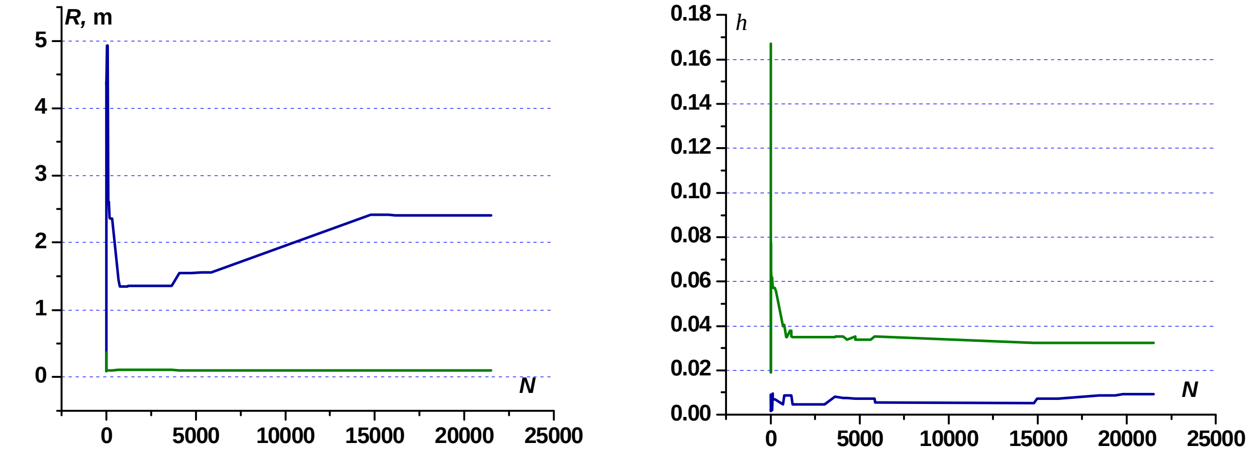


Fig. 14. *Results of 3-mass DVA's on the plates of different radius application: (a) – the main mass in time vibration; (b) – process of DVA's evaluation function evolution*

Process of geometrical DVA's parameters evolution by complex optimization is presented in Fig. 15.



a) *b)*

Fig. 15. *Process of geometrical DVA's parameters evolution by complex optimization: (a) – radiuses of curvature; (b) – damping*

Simultaneous optimization

Let's consider now simultaneous optimization by impulse and harmonic loading (the sum of evaluation functions (3) and (6)). In Fig. 16, 17 results of optimization for various initial time are presented.

The process of optimization is:

______________________________N = 1______________________________

Dx .333E-01 DG .179E+00 Mx3 .996E+00 fx .164E+01 fEkx .675E+00 CiL .136E+00 …

______________________________N = 5947______________________________

Dx .407E-01 DG .899E+00 Mx3 .998E+00 fx .100E+01 fEkx .800E+00 CiL .578E-01

The process of optimization is:

______________________________N = 1______________________________

Dx .333E-01 DG .179E+00 Mx3 .996E+00 fx .164E+01 fEkx .675E+00 CiL .393E+00 …

______________________________N = 5082______________________________

Dx .569E-01 DG .892E+00 Mx3 .999E+00 fx .775E+00 fEkx .369E+00 CiL .716E-01

Parameters of basic part are the same, the clearans is Ax = 0.15m.

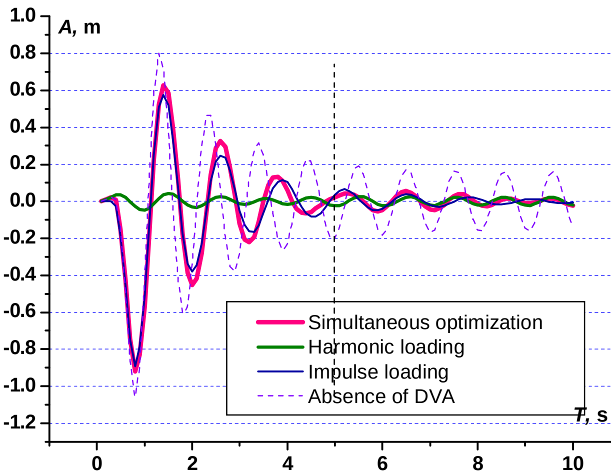


Fig. 16. *Results of optimization for initial time 5s*

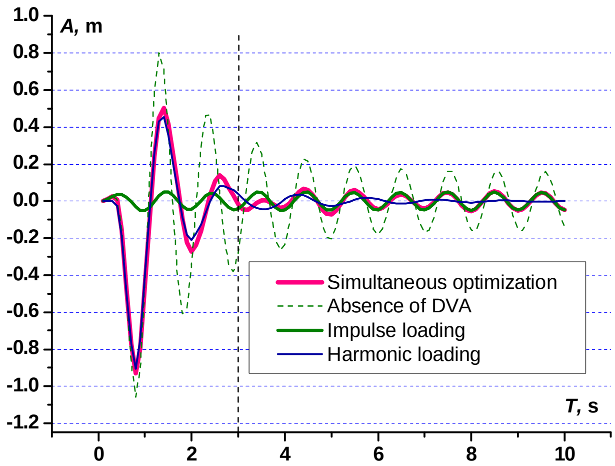


Fig. 17. *Results of optimization for initial time 3s*

Equations for the pump with the particle DVA's

In Fig. 18 the scheme of pump structure *P* with 2 particle absorbers attachment is presented.

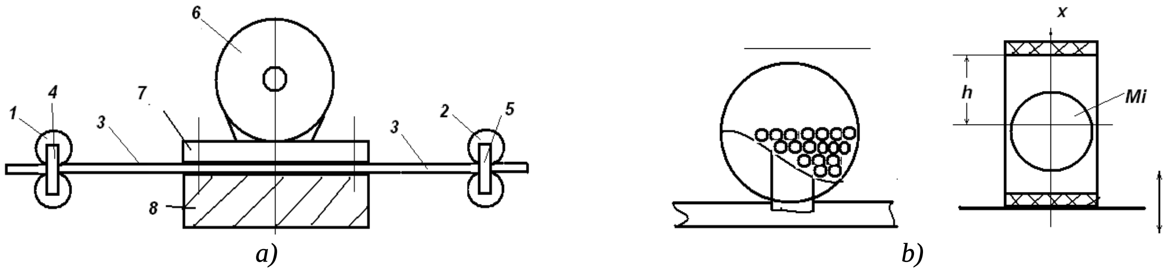


Fig. 18. *Pump – DVA scheme (a); DVA filled container(b); container model (c)*

Here (1) is pump base; (1,2,3,4,5) – DVA's; (6,7) – pump and pump base; (8) is pump foundation.

In this paper the condensed numerical model is proposed. The problem is solved on the basis of modified method of modal synthesis. The basis of these methods is in deriving solving set of equations in a normal form at minimum application of matrix operations [21–24]. The system of equations in the condensed rangy is obtained:

$$\begin{gathered}
m_1 \frac{d^2 w_0}{dt^2} + (k_1 D_K + k_A D_A + k_{A2} D_{A2}) \frac{dw_0}{dt} + (k_1 + k_A + k_{A2}) w_0 - k_A D_A \frac{dw_A}{dt} \\
- k_{A2} D_{A2} \frac{dw_{A2}}{dt} - k_A w_A - k_{A2} w_{A2} = F \\
m_A \frac{d^2 w_A}{dt^2} + k_A D_A \frac{dw_A}{dt} + k_A w_A - k_A D_A \frac{dw_0}{dt} - k_A w_0 = 0, \\
m_{A2} \frac{d^2 w_{A2}}{dt^2} + k_{A2} D_{A2} \frac{dw_{A2}}{dt} + k_{A2} w_{A2} - k_{A2} D_{A2} \frac{dw_0}{dt} - k_{A2} w_0 = 0.
\end{gathered} \tag{7}$$

Here: m, m_A, m_{A2} are masses of base and DVA's; k_1, k_{A1} k_{A2} – appropriate rigidities; D_K, D_A. D_{A2} – viscoelastic damping coefficients; w_0, w_A, w_{A2} – appropriate displacement, F – harmonic excitation. For the particle dynamic modeling the condensed impact mass damper was applied (Fig. 1).

The equations for the impact mass are:

$$\begin{aligned}
& m_i \frac{d^2 w_i}{dt^2} + C_i \frac{dw_i}{dt} + k_G(x)(w_i - w_0) + C_G(x)\left(\frac{dw_i}{dt} - \frac{dw_0}{dt}\right) = 0 \quad |w_i - w_0| > |h - R|; \\
& m_i \frac{d^2 w_i}{dt^2} + C_i \frac{dw_i}{dt} = 0, \qquad |w_i - w_0| \le |h - R|.
\end{aligned} \tag{8}$$

Here: m_i – particle mass, C_i – damping viscoelastic coefficient, modeling particle traction in container, K_G – rigid coefficient and C_G – viscoelastic coefficient for particle elastic impact modeling, w_i – impact mass displacement.

Experimental setup. Dynamic model parameters identification

There were applied two experimental schemes. First – DVA kinematic excitation (Fig. 19, a), second – base impact (Fig. 19, *b*). Here 1 – sensor, 2 – beam element, 3 – base, 4 – impact hammer. In Fig. 20 experimental setting for determination of dynamic properties of the DVA engine-pump system is presented.

Although some parameters of DVA and pump can be determined by experiments, but some, such as basic system mass m_1 remains unknown in equation 7. For a more precise definition of the model parameters was conducted several additional experiments (for the definition of parameters m_1 , k_1 - mass and stiffness of the primary system). At the same time DVA parameters m_A , k_A. require refinement

Although they can be calculated more accurately than the basic parameters of the system, yet it takes a lot of effort both in determining of the elastic properties of DVA and DVA clamping plate . Although you can conduct a detailed theoretical analysis [26; 27], but based on a series of simple experiments can be quite accurately determine these parameters as integrated value included in the system of equations (4). As the device is designed to test we are using our DVA. Perform for this series of experiments: kinematic perturbation DVA for its different masses To determine all the parameters k_1 , m_1 , m_A , k_A we should apply a genetic method to minimize the objective function $F_c = \sum_i |f_T(M_i) - f_e(M_i)|$, where $f_T(M_i) = f_T(M_i, k_1, m_A, k_A)$ theoretically obtained values of natural frequencies (first eigen-frequencies), $f_e(M_i)$ – experimental values. Detailed identification schemes are presented below.

Five stages are considered: adaptation of theory to various conditions of fixing and deformation; research of sensitiveness in relation to the DVA's and base design parameters; numerical experiments on identification of undefined parameters, practical parameters identification by exploring different schemes of experimental setup and, finally, posterior analysis of identification quality.

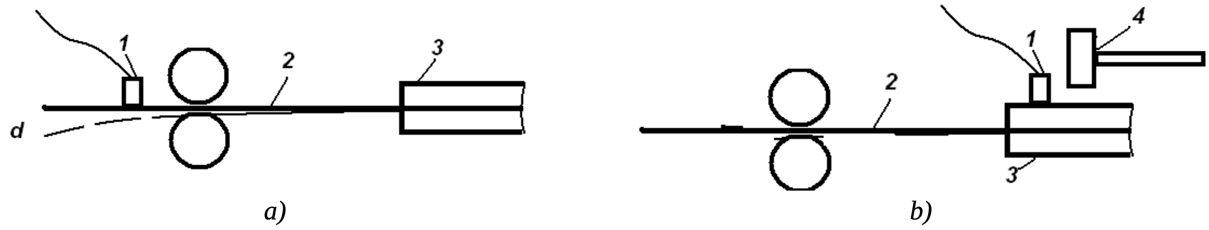


Fig. 19. Experimental schemes: (a) – DVA kinematic exitation; (b) – base impact

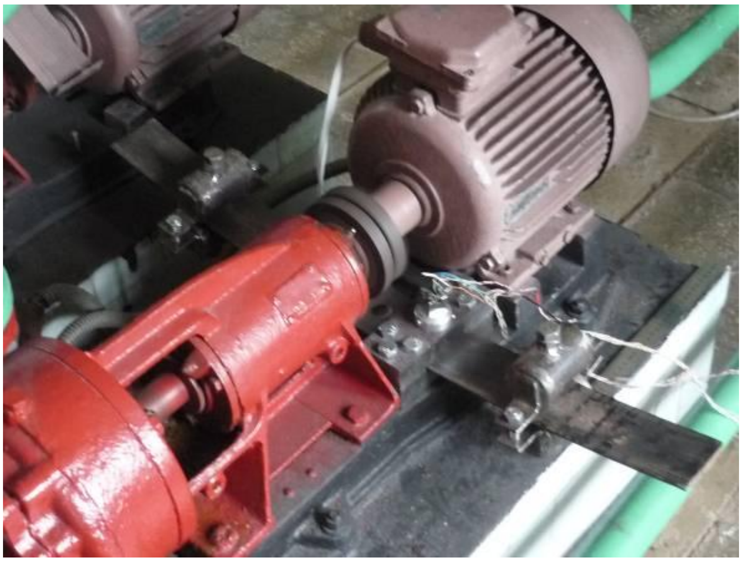


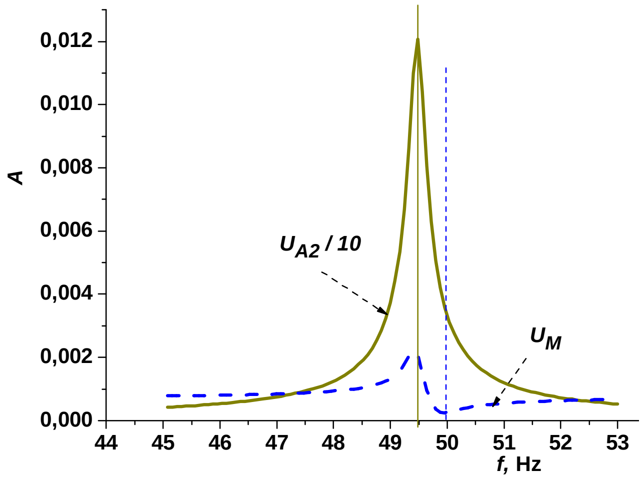
Fig. 20. Experimental setting for determination of dynamic properties of the DVA engine-pump system

Optimization for non-resonance DVA

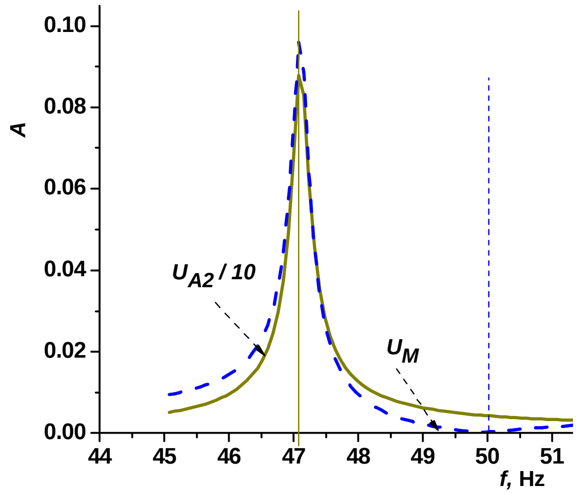
Optimization for non-resonance DVA has some not widely known specific. Let's consider DVA for the rigid basic system $f_M > f_A$ (basic mass eigen-frequency is greater than DVA's eigen-frequency). Frequency response functions (FRF's) for the base structure are presented in Fig. 21 for various f_M .

Parameters are: m_1 = 20 kg, m_A = 2.3 kg, k_1 = 2000-8000 kN/m , k_2 = 2000-8000 kN/m, $D_1 = D_A$ = = 0.00001. Only one DVA is considered. The parameters k_{A1} , D_A of DVA are optimized in frequency band $49Hz < f < 51Hz$ (see below).

Now let us consider DVA for the soft basic system $f_M < f_A$. FRF's are presented in Fig. 22 for various f_M. The large DVA's shift may be seen from the DVA's action zones.



a)



b)

Fig. 21. *FRF for basic system (dot line); FRF for DVA (solid line): (a) -* $f_M = 140\,\text{Hz}$ *; (b) -* $f_M = 70$ Hz

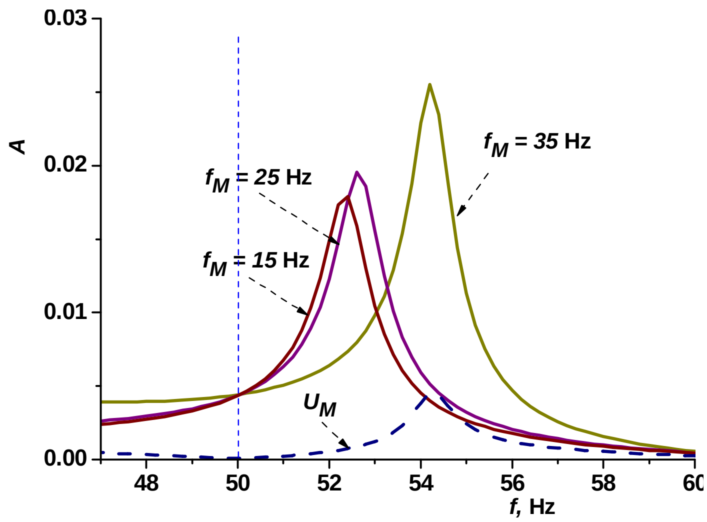


Fig. 22. *FRF for basic system (dot line); FRF for DVA's (solid lines)*

Particle DVA optimization

The complexity and high dimensionality of some models lead to the use of a heuristic search method. In this matter, Genetic Algorithms (GA) has proven to be a suitable optimization tool for a wide selection of problems. The optimization function is:

$$Fcil = \max_{f_1<\omega_1<f_2} \left(\int_{f_1}^{f_2} \left| u_1(f) \right| P(f)\,df \right), \tag{9}$$

where u_1 – vibration level of base, f_1, f_2 – boundaries of observed frequency domain, P– weight function, ω_1. – first eigen-frequency. Parameters of optimization are m_A, m_{A2}, k_{A1}, k_{A2}, D_A, D_{A2}. Sum of DVA's masses is constant $m_A + m_{A2} = 3.8kg$. In Fig. 23 results of optimization are presented.

On the basis of theoretical and experimental studies optimum parameters of DVA's was found. In Fig. 24 the acceleration of main structure at the operating frequency are presented. The measured deviation from the operating frequency were within 0.1-0.15 %. The following algorithm was applied: DVA mass was moved on a beam with some fixed pitch (1 cm). Based on the kinematic perturbation scheme (Fig. 17, *a*), DVA natural frequency was measured. Then, based on measurements carried out with the included pump, optimization was carried out. DVA mass – 1.881 kg. As you can see in Fig. 24, at a frequency close to the theoretical optimum, the amplitude of oscillation of the main structure is reduced by an order.

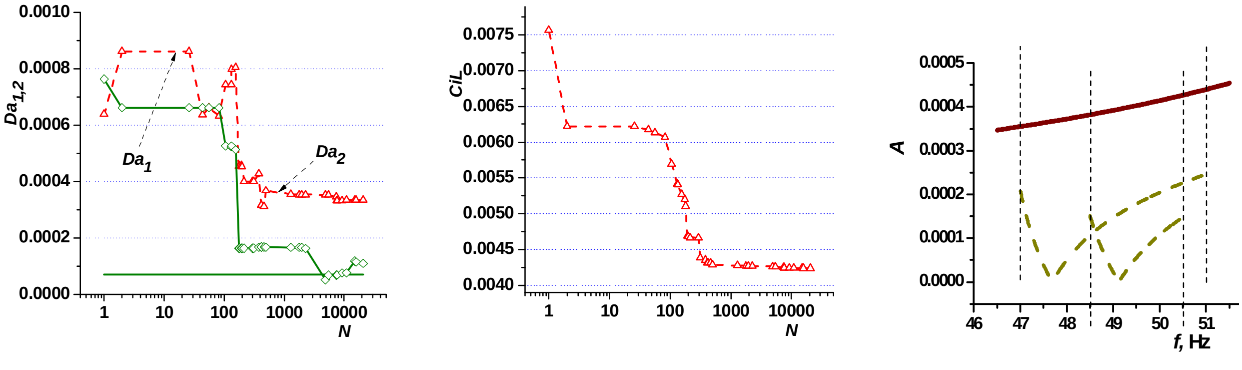


Fig. 23. Result of optimization: (a) – DVA's damping coefficients evolution; (b) – F_{cil} evolution; (c) – optimal FRF of base (for different frequency band), solid line – system without DVA

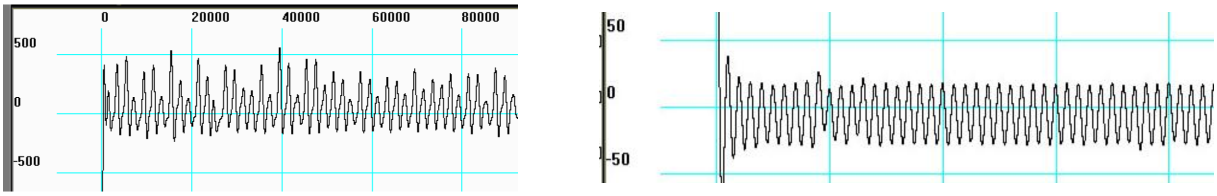


Fig. 24. Main structure acceleration at different DVA frequencies : (a) – $f_A = 46\text{Hz}$; (b) – $f_A = 47\text{Hz}$

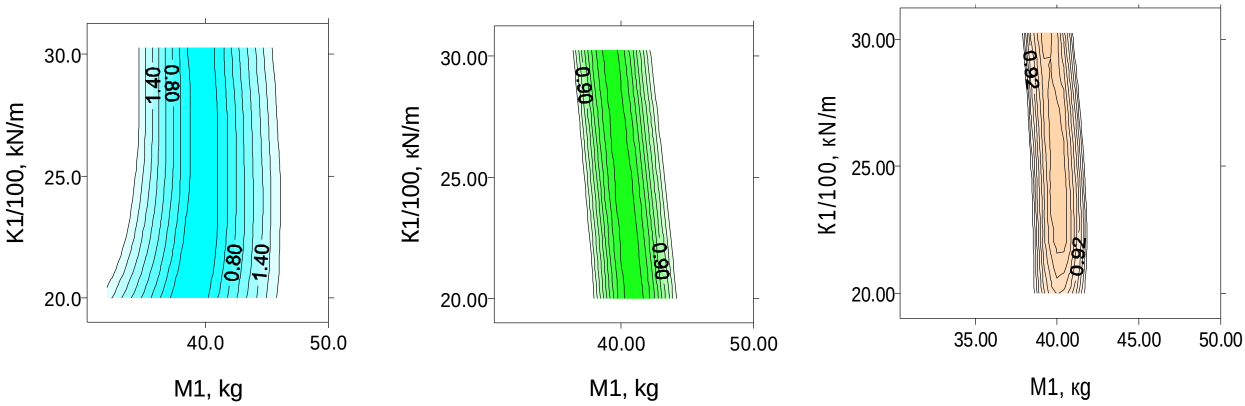
Identification scheme verification

Although some parameters of DVA and pump can be determined by experiments, but some, such as basic system mass m_1 remains unknown in equation (1.1). For a more precise definition of the model parameters several additional experiments were conducted (for the definition of parameters m_1 , k_1 - mass and stiffness of the primary system). At the same time DVA parameters m_A , k_A . require refinement. Although they can be calculated more accurately than the basic parameters of the system, yet it takes a lot of effort both in determining of the elastic properties of DVA and DVA clamping plate. Although you can conduct a detailed theoretical analysis [23–31], a series of simple experiments can quite accurately determine these parameters as integrated value included in the system of equations (1). Originally let us show the correctness of these schemes. Fig. 25 shows the frequency deviation map centered on values, depending on the changes of parameters m_1 , k_1 :

$$m_{1i} = m_{10}(i - N/2), i = 1,...N,$$
$$k_{1i} = k_{10}(i - N/2), i = 1,...N. \tag{10}$$

As the device is designed to test, we are using our DVA. Performing for this series of experiments: kinematical perturbation DVA for its different masses (see diagram of the numerical experiment in Fig. 25).

Values of first eigen-frequencies were obtained for different masses m_A located at the edge of the DVA plate. We see that the basic system parameters are determined uniquely by combined map (a) and (b) to (c). To determine all the parameters k_1 , m_1 , m_A , k_A we should apply a genetic method to minimize the objective function $F_c = \sum_i \left| f_T(M_i) - f_e(M_i) \right|$, where - $f_T(M_i) = f_T(M_i, k_1, m_A, k_A)$ theoretically obtained values of natural frequencies (first eigenfrequencies), $f_e(M_i)$ - experimental values. The next values of first eigenfrequencies were obtained for the masses, located on verge of DVA's plate (Table 1).



a) b) c)

Fig. 25. *The amount of frequencies deviation maps (3) of DVA centered values depending on the change of parameters k_1 , m_1 : (a) – for DVA weight m_1 = 1.5 kg; (b) – m_1 = 3 kg ; (c) – combined map (sum of maps (a) and (b))*

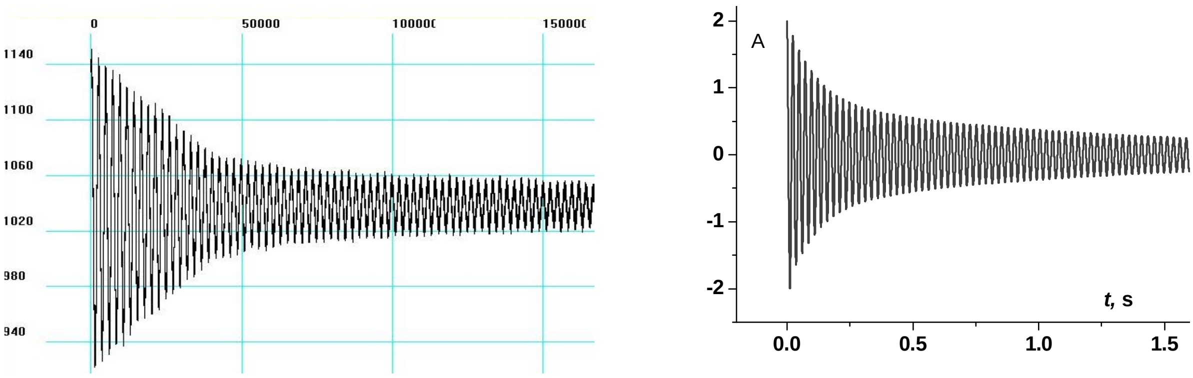
Table 1

Eigen-frequencies for different DVA's masses

M, kg	0	0,669	1,100	1,521	1,881	3,115
F, Hzц	69	48	36	32,3	29,2	24,4

We get the following values for the main components - pump in place joining DVA : f_{Km} = 65.5 Hz , m_1 = 34.4 kg. If the effect of the mass is difficult to track because of the complexity of the design of the pump, the oscillation frequency can be seen for the shock disturbance. We see that it is in the vicinity of 65 Hz (as defined in theory). That is, the natural frequency of the main structure above the operating frequency of 50 Hz. It gives information on what neighborhood eigen-frequencies DVA to seek optimum vibroabsorption at the operating frequency.

In Fig. 26 the experimental and theoretical vibration decay is presented for particle filled container.



a) b)

Fig. 26. *Experimental (a) and theoretical (b) vibration decay*

By means of such scheme the damping in filled container may be appreciated $D_A \approx 0.0001$.

Optimization

Now we can optimize DVA's system. DVA Optimization with particle filling container. The complexity and high dimensionality of some models lead to the use of a heuristic search method. In this matter, Genetic Algorithms (GA) has proven to be a suitable optimization tool for a wide selection of problems. The optimization function is (3). Parameters of optimization are m_A , m_{A2} , k_{A1} , k_{A2} , D_A . D_{A2} . Sum of DVA's masses is constant $m_A + m_{A2} = 3.8kg$

In Fig. 25 results of DVA’ s with particle filled container optimization are presented.

N = 1________________________

fKm .788E+02 DempK .101E-04

Ma .230E+01 Ma2 .150E+01 LA -.321E+02 La2 .215E+02

CiL .152E-02

Da .100E-04 Da2 .302E-05 fk .296E+02 fk2 .596E+02

. . .

N = 5459____________________

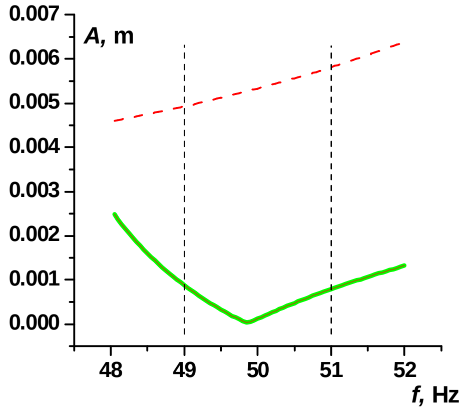
fKm .788E+02 DempK .991E-05

Ma .230E+01 Ma2 .150E+01 LA -.215E+02 La2 .258E+02

CiL .360E-03

Da .993E-05 Da2 .298E-05 fk .507E+02 fk2 .475E+02

a)



b)

***Fig. 27.** The process (a) and results (b) of DVA’ s with particle filled container optimization*

System modelling by program packages based on FEM

The most popular computational methods used in structural dynamics are: the finite element method (FEM) While investigating higher frequency ranges for acoustic applications and using finite elements, structures are decomposed into smaller and smaller elements. The mesh size is chosen so that its largest dimension does not exceed the wavelength of the vibration. Going in this direction, when dealing with complex and large structures, the number of elements often becomes prohibitive. The calculation of eigenvalues in the range of medium frequency becomes cumbersome and time consuming. For many cases the amplitude excitation maximum is low. This may be used in main structure modelling, taken into account only it thirst eigenvalue.

As example, let us consider the boom frame of the boom sprayer. The frequency characteristics of boom are defined by program APM WinMachin. In Fig. 28 the boom deformable part model and its eigenfrequencies are presented.

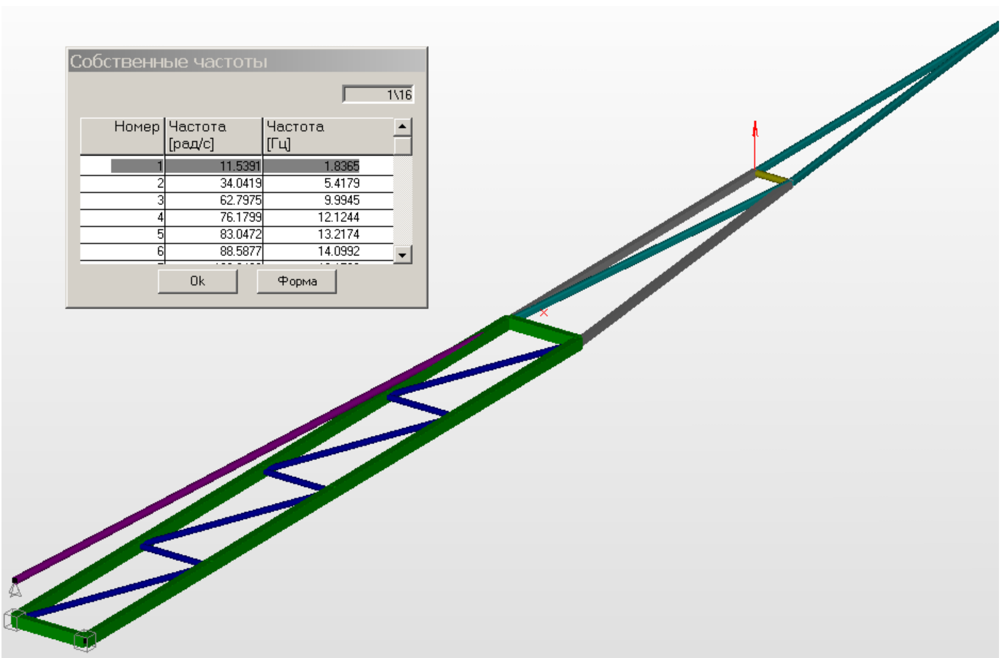


***Fig. 28.** Boom deformable part model and its eigenfrequencies*

First eigenfrequency is 1.8 Hz. Let us define the rigidity in the point marked by pointer (Fig. 18). By this date we can define the parameters of boom frame and put its into it equation:

$$m\ddot{w}+kD\dot{w}+kw=F. \qquad (11)$$

Here rigidity is:

$$k = F/\Delta \approx 1000/0.295 \approx 4556 N/\text{m}. \tag{12}$$

The mass is now:

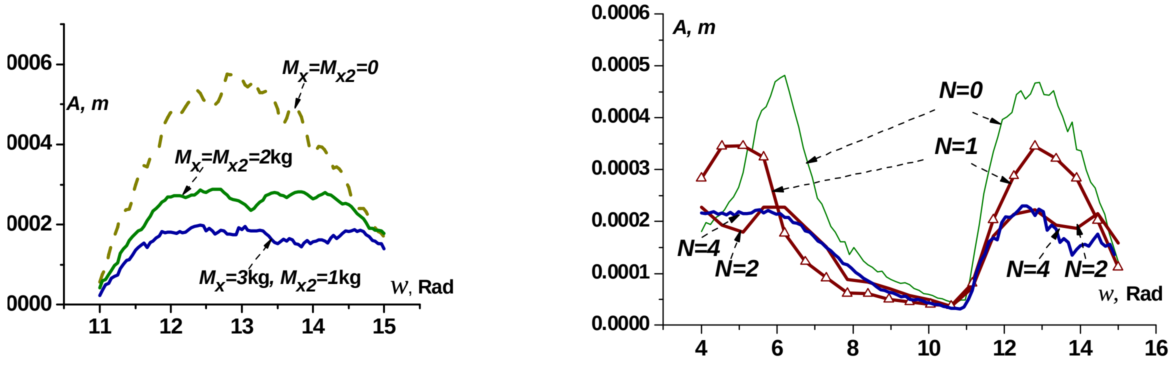
$$m = k/\omega^2 \approx 4556/11.54^2 \approx 34.2 kg. \tag{13}$$

DVA optimization for boom

Optimal DVA parameters are defined for the function (14):

$$CiL = Max(u_1(f)),\ \alpha f_R < \text{f} < \beta f_R. \tag{14}$$

It is maximum boom deflection in the frequency range. The optimization of boom in the frequency range is shown below (Fig. 29). The optimization may be done for various frequency range simultaneously (Fig. 29, *b*).



a) *b)*

Fig. 29. *Optimization of boom in the frequency range: (a) – one frequency range; (b) – two frequency range*

Substantial part is acted by the division of the masses between DVA (Fig. 20, *b*). It is possible to find (Fig. 18, *b*) that only for two DVA's substantial effect takes place in two ranges. Four DVA improve a situation in a higher frequency range. Here total mass of all DVA's is identical and 4 kg is evened. Character of action of different DVA (different half-length of containers) is resulted, on Fig. 30. Loop of hysteresis is shown in Fig. 31 for mass of DVA (the dotted line is a container without limitations).

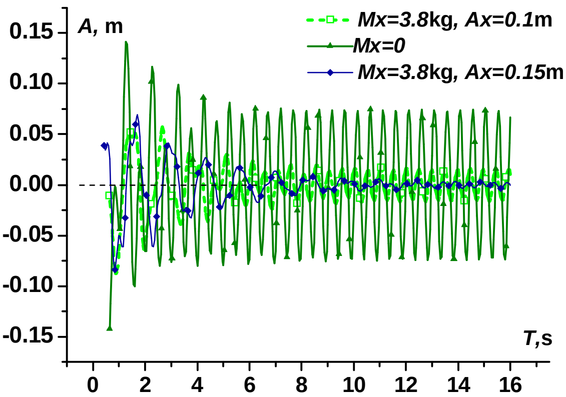


Fig. 30. *Character of action of different DVA (different half-length of containers)*

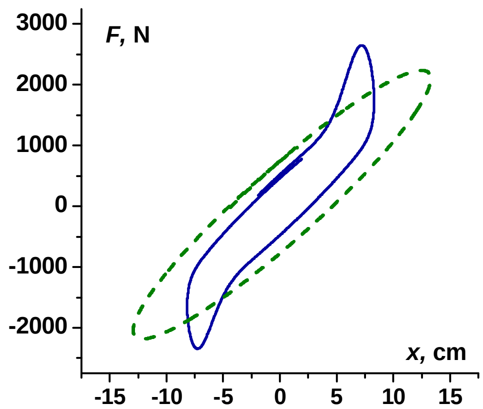


Fig. 31. *Loop of hysteresis for mass of DVA*

Conclusions

In order to determine the optimal parameters of DVA the complete modeling of dynamics of devices should be made. Paper deals with the new methods for the explicit determination of the frequency characteristics of dynamic vibration absorbers by narrow frequency and impulse excitation. Few parameters numerical schemes of vibration analysis are under discussion. The influence of elastic and damping properties of the basic construction and dynamic vibration absorbers are considered. Optimization for non-resonance DVA is done with its specific. The discrete-continue models of machines dynamics of or such machines as water pump with the attachment of dynamic vibration absorbers are offered. The algorithms for vibration decreasing are received. The new vibroabsorbing elements are proposed with more than one impact mass in container. The first eigenfrequencies are calculated and obtained experimentally for different masses attached to elastic elements of the dynamic vibration absorbers. The one-digit values are established not only for the dynamic vibration absorber parameters, but also for mechanical parameter of base structure – pump in connection points of the dynamic vibration absorbers. For the elongated base structures modeling in the low frequency range the FEA was used. Finally, present research develops the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling.

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