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Design and optimization of the impact and particle vibration absorbers

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ABSTRACT

Purpose: The main aim of this paper is improved dynamic vibration impact and particle aabsorbers design with taking into account complex machines dynamic.

Design/methodology/approach: The numerical schemes row is considered for the complex vibro-loaded constructions. Methods of decomposition and the numerical schemes synthesis on the basis of discrete-continuum modelling are considered.

Findings: Development of mathematical models of complicated machines in view of their interaction with system of dynamic vibration absorbers. Dynamic vibration absorbers – complicated machines system design optimized on vibro- absorption properties.

Research limitations/implications: The research must be done for non-linear machine dynamics.

Practical implications: The absorbers designed in accordance with this paper can be applied not only to electric machines or aeronautic structures, but to any other type of vibro-loaded structure, such as turbo machines, pumps, cars suspensions, chisel installation, optical, magneto-optical disks, washing machine, refrigerator, vacuum cleaner, etc.

Originality/value: The paper has novelty both in theoretical, and in practical aspect, In order that optimal parameters of DVA be determinate the complete modelling of dynamics of rotating machine should be made. Traditional design methodology, based on discontinuous models of structures and machines are not effective for high frequency vibration. The one-digit values are established not only for the dynamic vibration absorber parameters, but also for mechanical of base structure in connection points of the dynamic vibration absorbers.

Keywords: Complex machines dynamic; Impact absorber; Particle absorber; Base structure parameters; Optimal

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ANALYSIS AND MODELLING

1. Introduction

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 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.

Machines will typically introduce both acoustic and vibration energy into any fluids or structures surrounding the machinery. This is dangerous for both for its construction strength and human health. From two general classes of tools used to assess and optimize machines acoustic performance: test based methods and Computer Aided Engineering based methods, the second should be discussed in this paper. Large rotating elements, particularly such elements as exhaust fan rotors used in electric power generating plants or in gas compression plants, are unbalanced during operation due to their exposure to various factors. It is often impossible to balance the rotating elements properly to reduce the vibration to an acceptable level.

The paper contemplates the provision of dynamic vibration absorbers (DVA) or any number of such absorbers [1-3]. 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 concrete and steel foundation with a new, sufficiently massive one. In order to determine the optimal parameters of an absorber the need for complete modeling of rotating machine dynamics 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 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 [4], Warburton [5], Hunt [6], Snowdon [7], Korenev and Reznikovc [8] and Aida [9] et al.

A particle-based damping system can overcome some limitations by using particles as the damping medium and inter- particle interaction as the damping mechanism. The dual solid- and liquid-like properties of a particle medium provide the system with two unique advantages: (i) the solid-like properties can enable temperature independence, and (ii) the liquid- like properties can allow for flow and reorganization of particles to facilitate fatigue-free performance [10-12].

Although some parameters of DVA can be determined by experiments, but some, such as basic system mass m_1 remains unknown. For a more precise definition of the model parameters several additional experiments must be conducted (for the definition of the primary system parameters). At the same time DVA parameters 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. The numerical schemes (NS) row for the complex vibro-loaded construction and methods of decomposition and the NS synthesis are considered in our paper on the basis of new methods of modal synthesis [13-15]. The problem of DVA design may be divided into such steps (Fig. 1).

It is not a comprehensive list of criterions. This list may be completed, for example, by such criterions: damage control, aesthetic design etc. Not at the last place must be DVA design simplicity, especially for the theoretical purposes. Most important from them is criterion "vibration absorbing properties optimization". This important criterion should be discussed later. It is not enough attention paid to such an important criterion as DVA resource optimization. In order optimal parameters of DVA to be determinate the complete modeling of dynamics of machine is obvious. The two degrees of freedom model is totally inadequate to calculate the vibration frequencies of the construction with accuracy and therefore, for a sufficiently accurate determination of its dimensional characteristics so as to determine such frequencies. It is therefore necessary in practice to dimension the construction through more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination.

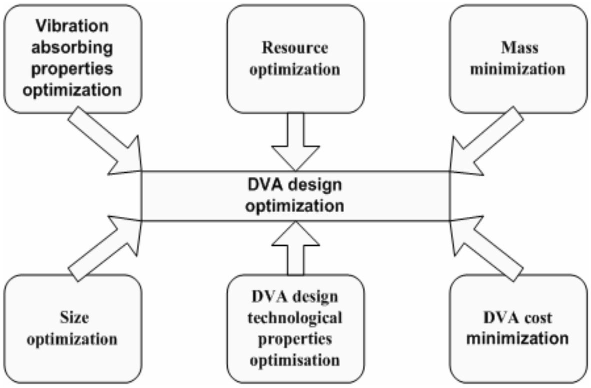


Fig.1. DVA design optimization

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 [16, 17].

2. One-mass system with the impact DVA

Let us consider condensed model DVA-one-mass system. In Fig. 2 the impact mass type DVA is presented: an additional impact mass in container with elastic barrier elements.

The system of equations is obtained:

$$m_1 \frac{d^2u_1}{dt^2} + k_1(u_1 - u_0) + k_A(u_1 - u_A) - {m_{X1}}/{R_{X1}}\,(u_{X1} - u_A)$$
$$+ k_{X1}F_1(u_1 - u_{X1}) - {m_{XN}}/{R_{XN}}\,(u_{XN} - u_A) \qquad (1)$$
$$+ k_{XN}F_N(u_1 - u_{XN}) = F(t),$$
$$m_{X1}\frac{d^2u_{X1}}{dt^2} + {m_{X1}}/{R_{X1}}\,(u_{X1} - u_A) - k_{X1}F_1(u_1 - u_{X1}) = 0,$$
$$\cdots$$
$$m_{XN}\frac{d^2u_{XN}}{dt^2} + {m_{XN}}/{R_{XN}}\,(u_{XN} - u_A) - k_{XN}F_N(u_1 - u_{XN}) = 0.$$

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:

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

Were A – clearance and K_{vi} – boundary elements rigidity.

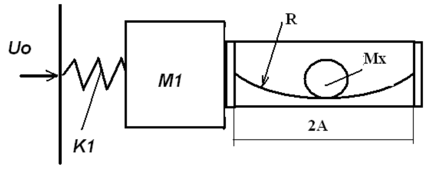


Fig. 2. Pendulum type DVA with the additional elements

2.1. Design of drop forging formed on TSFP

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

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

The results of the process of optimization for the DVA (Fig. 2) are presented below for different DVA's masses. Here 4 parameters of optimization are used: fx, fx2 DVA's eigenfrequencies; Dx, Dx2 – proportional viscous damping (added to all equations terms $k_{Xi}D_{Xi}\,{du_i}/{dt}$). The prime system mass is m_1 *=10*kg, the prime system eigenfrequency – f_R *=1*Hz=*6.28* Rad/s, the proportional damping η*=0.03*.

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

For system with two dangerous frequency intervals the grater amount of DVA's may be used (Fig. 3).

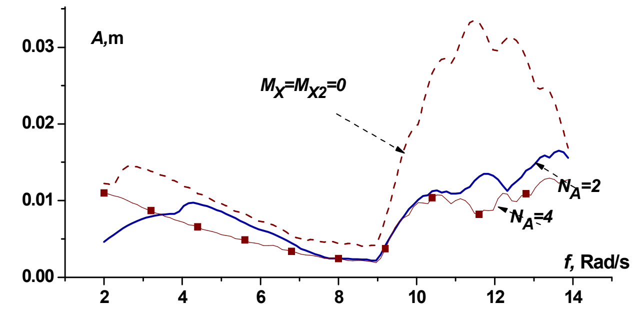


Fig. 3. The results of optimization for system with two frequency intervals by number of DVA's $N_A = 2{,}4$

Total DVA's mass is even 4kg. For $N_A = 4$ the better result may be seen.

3. DVA with the particle damper

In Fig.4 the scheme of pump structure P with 2 particle absorbers attachment is presented.

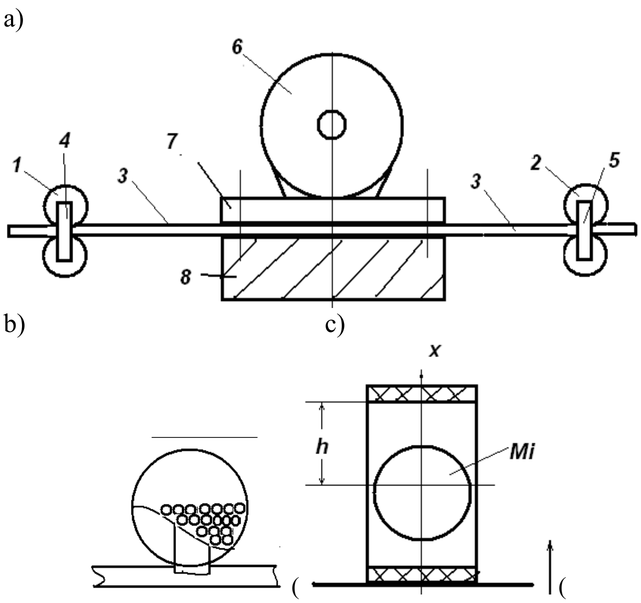


Fig. 4. Pump – DVA scheme a); DVA filled container b); container model c)
Here (1) pump base; (1,2,3,4,5) – DVA's; (6,7)– pump and pump base; (8) pump foundation.

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 [13-15]. The system of equations in the condensed rangy is obtained:

$$m_1 \frac{d^2 w_0}{dt^2} + \left(k_1 D_K + k_A D_A + k_{A2} D_{A2}\right)\frac{dw_0}{dt}$$

$$+\left(k_1 + k_A + k_{A2}\right)w_0 - k_A D_A \frac{dw_A}{dt} - \quad , \tag{4}$$

$$-k_{A2} D_{A2}\frac{dw_{A2}}{dt} - k_A w_A - k_{A2} w_{A2} = F,$$

$$m_A \frac{d^2 w_A}{dt^2} + k_A D_A \frac{dw_A}{dt} + k_A w_A - k_A D_A \frac{dw_0}{dt} - k_A w_0 = 0,$$

(4)

$$m_{A2}\frac{d^2 w_{A2}}{dt^2} + k_{A2} D_{A2}\frac{dw_{A2}}{dt} + k_{A2} w_{A2} - k_{A2} D_{A2}\frac{dw_0}{dt}\ .$$

$$-k_{A2} w_0 = 0.$$

Here: m, m_A, m_{A2} 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. 1c). The equations for the impact mass are

$$m_i \frac{d^2 w_i}{dt^2} + C_i \frac{dw_i}{dt} + k_G(x)\left(w_i - w_0\right)$$

$$+ C_G(x)\left(\frac{dw_i}{dt} - \frac{dw_0}{dt}\right) = 0 \qquad \left|w_i - w_0\right| > \left|h - R\right|, \tag{5}$$

$$m_i \frac{d^2 w_i}{dt^2} + C_i \frac{dw_i}{dt} = 0 \qquad\qquad \left|w_i - w_0\right| \le \left|h - R\right|\ .$$

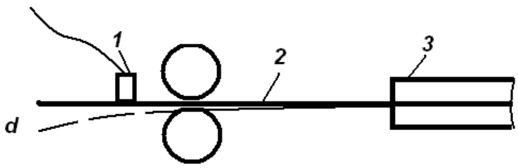
Here: m_i – particle mass, C_i - damping viscoelastic coefficient, modeling particle traction in container, K_G - rigid coefficient and C_G - viscoelastic coefficient for particle elastic impact modeling, w_i - impact mass displacement.

4. Experimental setup

There were two experimental schemes applied. First – DVA kinematic excitation (Fig. 5a), second – base impact (Fig. 5b).

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 4.1. 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.

a)



b)

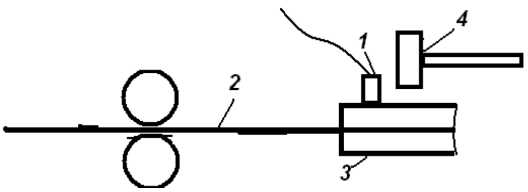


Fig. 5. Experimental schemes: a) - DVA kinematic exitation; b) - base impact: Here 1 – sensor, 2 – beam element, 3 – base, 4 – impact hammer.
Here 1 – sensor, 2 – beam element, 3 – base, 4 – impact hammer.

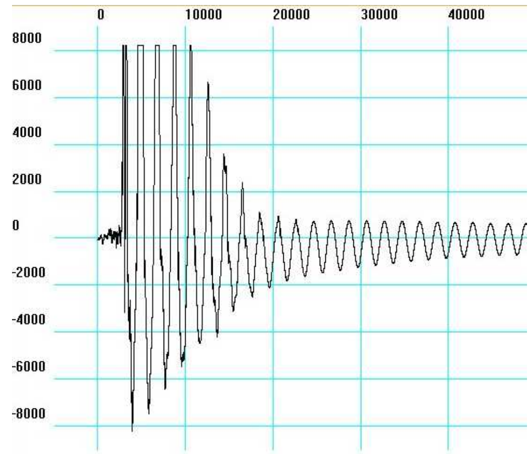
Although you can conduct a detailed theoretical analysis [16,17], but based on a series of simple experiments can be quite accurately determine these parameters as integrated value included in the system of equations (4). As the device is designed to test we are using our DVA. Perform for this series 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.

The next values of first eigen-frequencies were got for the masses, located on verge of DVA's plate (Tabl. 1).

Table 1.
Eigen-frequencies for different DVA's masses

M, kg	0	0.669	1.100	1.521	1.881	3.115
f, Hz	69	48	36	32.3	29.2	24.4

a)



b)

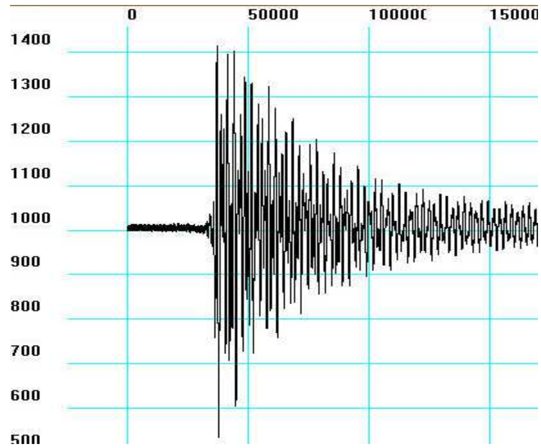
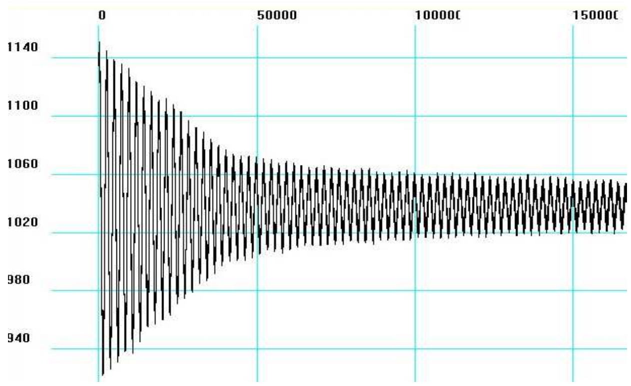


Fig. 6. Experimental signals: a) - DVA kinematic exitation; b) - base impact

We get the following values for the main components - pump in place joining DVA : $fKm = 65{,}5$ 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 vibro-absorption at the operating frequency. In Fig. 7 the experimental and theoretical vibration decay is presented for particle filled container.

By means of such scheme the damping in filled container may be appreciated $D_A \approx 0.0001$. Now we can optimize DVA's system.

a)



b)

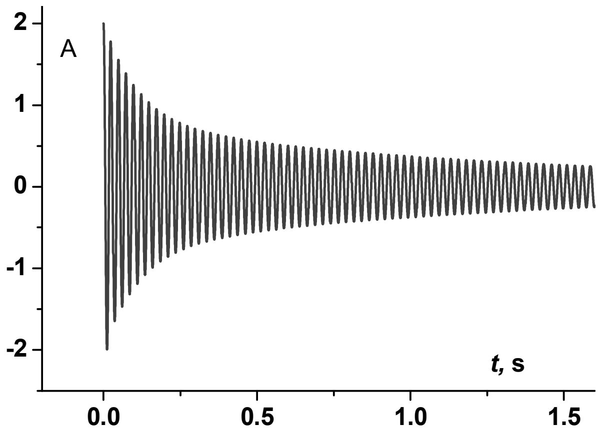


Fig. 7. Experimental (a) and theoretical (b) vibration decay

5. DVA optimization

5.1. DVA preliminary optimization

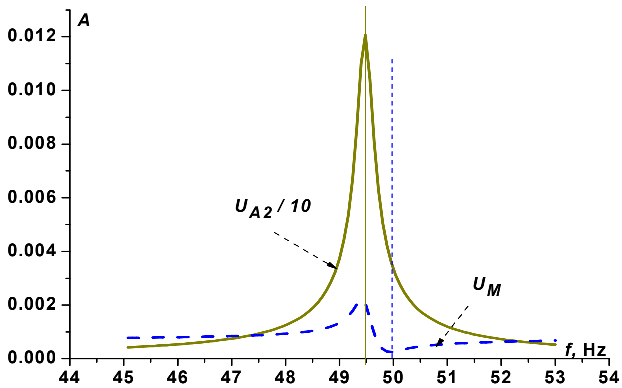
Genetic Algorithms (GA) has proven to be a suitable optimization tool for a wide selection of problems. The optimization function is

$$Fcil = \max_{f_1 < \omega_1 < f_2} \left(\int_{f_1}^{f_2} |u_1(f)| P(f) df \right) \tag{6}$$

U_1 – vibration level of base, f_1, f_2 – boundaries of observed frequency domain, P– weight function, ω_1. –first eigen-frequency.

Consider DVA for the rigid basic system $f_M > f_A$ (basic mass eigen-frequency is grater then DVA's eigen-frequency). Frequency response functions (FRF's) for the base structue and DVA are presented in Fig. 8 for various f_M. Here one-mass system with the single DVA is under discussion.

a)



b)

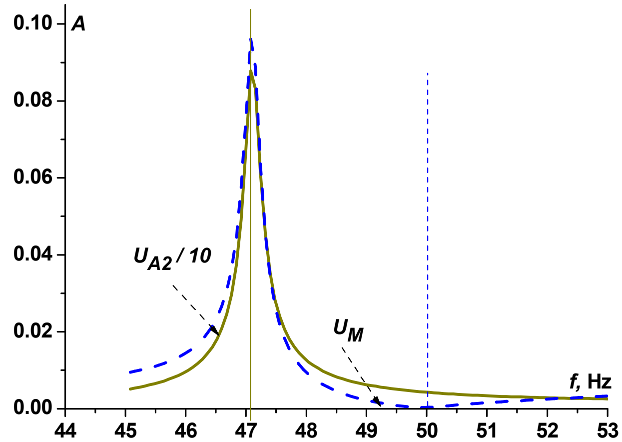


Fig. 8. FRF for basic system (dot line); FRF for DVA (solid line): (a) – $f_M = 140 Гц$; (b) – $f_M = 70 Гц$

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

Now let us consider DVA for the soft basic system $f_M < f_A$. FRF's are presented in Fig. 9 for various f_M.

The large optimal DVA's eigen-frequency shift may be seen from the DVA's action zones. In spite of the conventional resonance DVA, the maximum DVA amplitude is moved from the working frequency (50Hz).

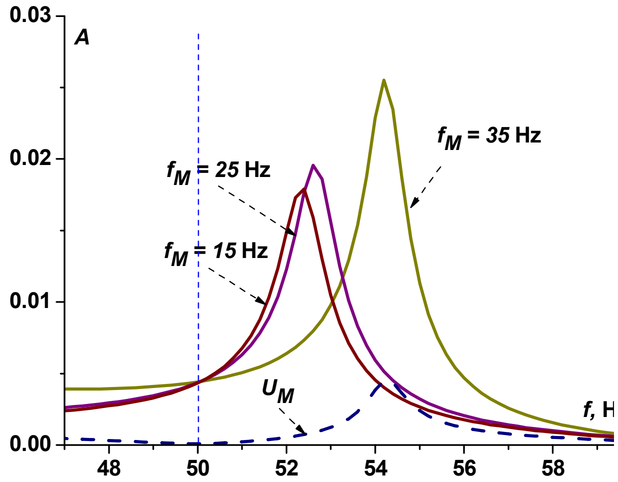


Fig. 9. FRF for basic system (dot line); FRF for DVA's (solid lines)

5.2. DVA multyparameter 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 (3). Parameters of optimization are m_A, m_{A2}, k_{A1}, k_{A2}, D_A. D_{A2}. Sum of DVA's masses is constant $m_A + m_{A2} = 3.8kg$

Results of DVA' s with particle filled container optimization in Fig. 10. are presented.

On the basis of theoretical and experimental studies optimum parameters of DVA's was found. The measured deviation from the operating frequency were within 0.1 - 0.15 %. They used the following algorithm: DVA mass was moved on a beam with some fixed pitch (1 cm). Based on the kinematic perturbation scheme (Fig. 2a), DVA natural frequency was measured. Then, based on measurements carried out with the included pump, optimization was carried. DHA mass - 1,881 kg. As you can see in Fig. 11, at a frequency close to the theoretical optimum, the amplitude of oscillation of the main structure is reduced by an order.

By the appropriate weight function $P(f)$ in (10) more uniform as in Fig. 7c absorbing may be reached. For example, such weight function

$$P(f) = Max(u_1) + (u_1(f) - 0.0001)**3./10.$$

$$f_1 + {(f_2 - f_1)}/{2} < f < f_2$$

$$P(f) = Max(u_1). \quad f_1 < f < +{(f_2 - f_1)}/{2} \tag{7}$$

Result of optimization for such weight function is presented in Fig. 12.

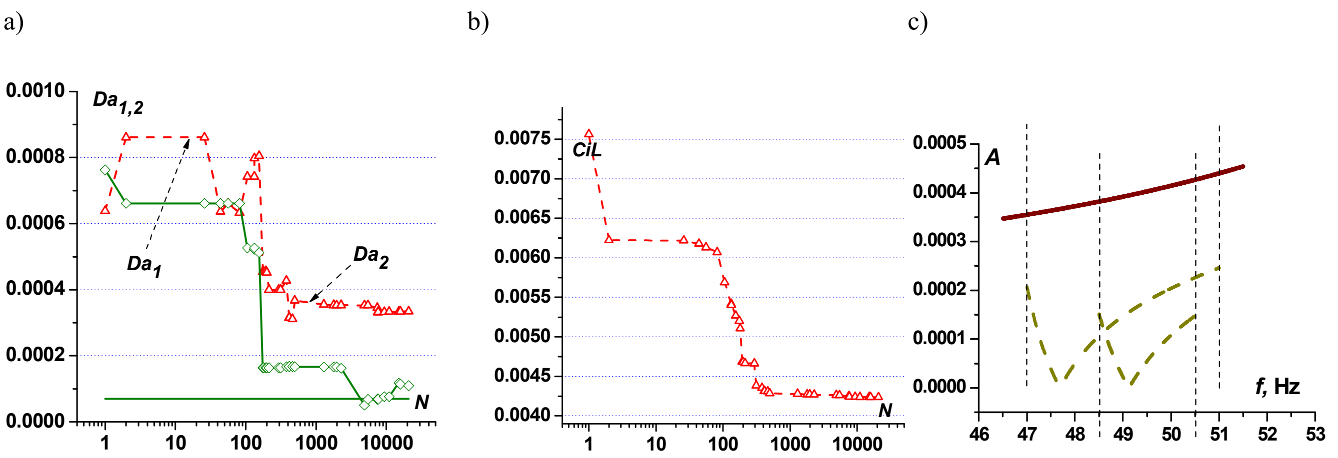


Fig. 10. Result of optimization: a) - DVA's damping coefficients evolution; b) - Fcil evolution; c) - optimal FRF of base (for different frequency band), solid line - system without DVA;s)

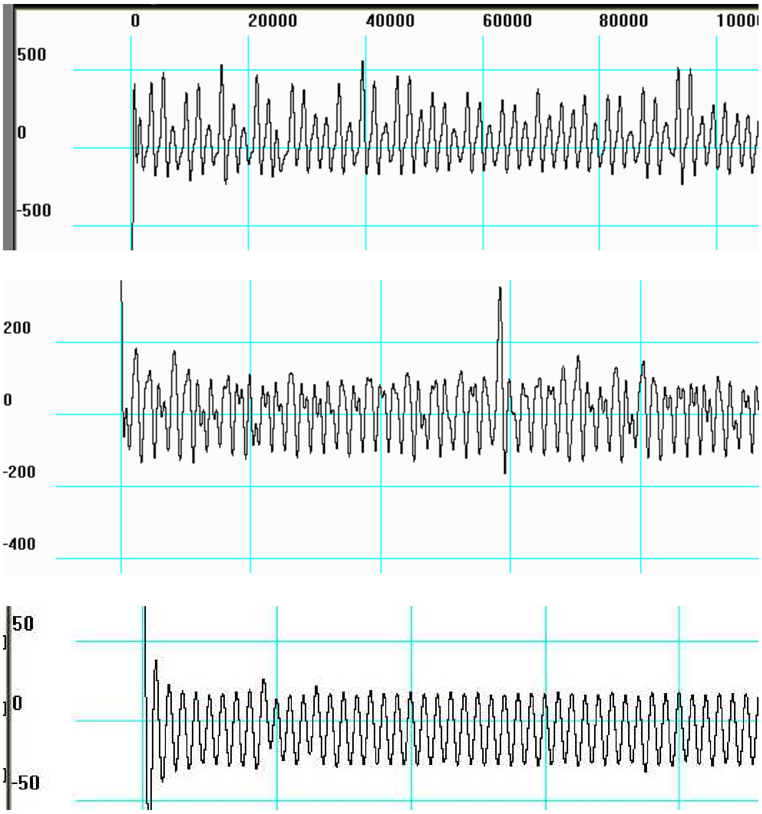


Fig. 11. Main structure acceleration at different natural frequencies DHA: a) - $f_A = 46Hz$; b) - $f_A = 46.5Hz$; c)- $f_A = 47Hz$

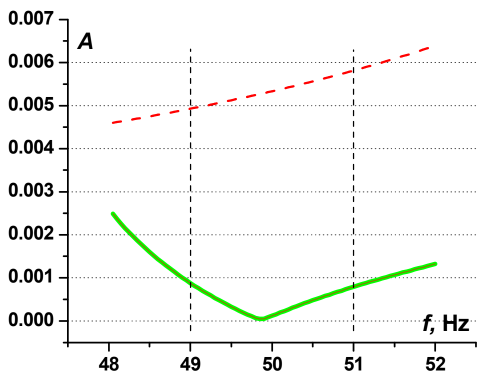


Fig. 12. Result of optimization for weight function (11)

5.3. DVA resourses

Finally, let us discuss the question of DVA strength. Influence of different DVA parameters was before explored on his efficiency. If rigidity parameters influence very substantially, damping parameters considerably less influential. At the same time it follows to expect the considerable influencing of DVA's damping properties on maximal amplitude of their vibrations, and the same on their durability. Maximal stress in the elastic DVA's element will be

$$\sigma_{MAX} = M/W = \omega^2 A M_A L_A z_{MAX}/EI = \omega^2 A M_A L_A / 6bh^2 E. \quad (8)$$

Here is M a moment, W is a moment of resistance of that cuts, A is amplitude of vibrations, M_A is DVA's mass, L_A is distance of mass from a clamp, z_{MAX} is maximal deviation of that cuts of plate from a middle line (in our case half- thickness of plate). All geometrical parameters of DVA's spring are regulated both his frequency descriptions and structural requirements. Amplitude of vibrations comes forward the unique independent managed parameter. In Fig.13 a are shown displacements of base construction at small and some greater damping. In Fig. 13 proper DVA's displacemets are shown.

It is possible to notice that at the insignificant worsening of vibroabsorbing properties of DVA (only in some narrow range of frequencies, that will exactly answer working frequency not necessarily) DVA vibration amplitude diminish on an order. That the risk of breakage of DVA diminishes considerably. In our construction of DVA it was attained by the use of containers filled by particles. Large damping at such family constructions of DVA does not bring to destruction a elastic element over in critical cases, when working frequency approaches own frequency of DVA, or when the transitional process of acceleration of pump is slow enough and DVA has time to collect large amplitudes of vibrations.

a)

b)

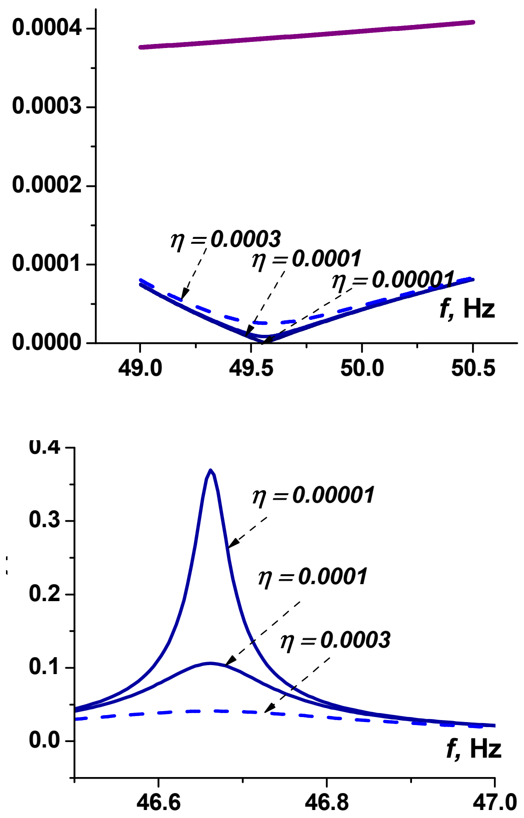


Fig. 13. Displacements by different DVA's damping: a) - base construction vibration; b) - DVA vibration

6. Concluding remarks

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 DVA's by narrow frequency 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. The discrete-continue models of machines dynamics of such machines as water pump with the attachment of dynamic vibration absorbers are offered. The large optimal DVA's eigen-frequency shift may be seen from the DVA's action zones. In spite of the conventional resonance DVA, the maximum DVA amplitude is moved from the working frequency (50Hz). The algorithms for vibration decreasing are received. The new vibroabsorbing elements are proposed. The first eigen-frequencies 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. Finally, present research develops the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling. By the increasing of DVA damping the risk of breakage of DVA diminishes considerably by small decreasing the DVA effectivities.

Additional information

Selected issues related to this paper are planned to be presented at the 22nd Winter International Scientific Conference on Achievements in Mechanical and Materials Engineering Winter-AMME'2015 in the framework of the Bidisciplinary Occasional Scientific Session BOSS'2015 celebrating the 10th anniversary of the foundation of the Association of Computational Materials Science and Surface Engineering and the World Academy of Materials and Manufacturing Engineering and of the foundation of the Worldwide Journal of Achievements in Materials and Manufacturing Engineering.

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