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MANY MASS ELASTIC BUFFERED IMPACT DAMPER

An impact damper is a passive control device which takes the form of a freely moving in the container mass, 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. A many mass elastic buffered impact damper can overcome some limitations of ordinary dynamic vibration absorber (DVA) by using impact masses as the damping medium and inter- masses an barrier interaction as the damping mechanism. . In this paper, an efficient numerical approach is proposed to maximize the damping of modes in a prescribed frequency range for general systems. The primary structure is modelled as a spring-mass system The influence of DVA's 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 and impact forced system.
Keywords: dynamic vibration absorber, impact damper, buffered damper, spring-mass system, optimization

Introduction Impact dampers have been extensively studied and investigated to counter vibrations in industrial machinery and structural systems. This is due to the fact that they are simple in design and easy to implement. In the article the methods of calculation and optimization of much of mass shock type DVA's are examined for diminishing of vibration at small frequencies of vibrations of base construction. The algorithms of diminishing of vibration of base construction are got. Absorption of energy is taken into account for an account to the movement of the rolling masses on the curved surface, to the blow of the masses to the resilient elastic barriers and the masses impacts between itself.

The dynamic response and performance of single unit impact dampers has been studied extensively. Pioneering research was conducted in [1]. Further research by [2] determined the existence of the optimal distances between the primary mass and the auxiliary mass for the impact damper. In [3] a piecewise analytical solution for the dynamics of an impact damper, and determined that the most effective damping condition occurred with two symmetric collisions per cycle is presented. In [4,5] further the relation between the coefficient of restitution and damping ratio of the impact and found the optimum damping effect by changing the mass ratio of the damper to the structure are provided. Moreover, many kinds of impact dampers have since been introduced, Among them with resilient buffers [6,7].

An improved scheme for detecting the time of impact has been developed in order to prevent negative collisions, which represent an intolerable scenario for large amplitude vibrations [8]. Detailed experiments with a horizontal impact damper explain the general performance and the resonance vibration of the integrated system, which occurs at a frequency, which is different from the original resonance frequency. 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 [9-13].

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

Consider now the DVA with 3 different impact masses in one container (Fig. 1)

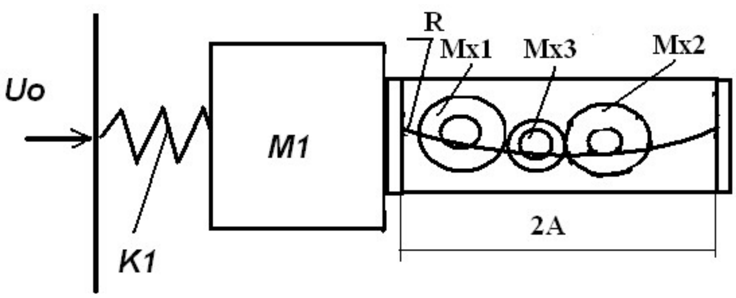


Fig. 1. DVA with 3 different impact masses

The system of equations is now

$$m_1 \frac{d^2 u_1}{dt^2} + k_1(u_1 - u_0) + k_A(u_1 - u_A) - {}^{m_{X1}}\!/_{R_{X1}}(u_{X1} - u_A) + k_{X1} F_1(u_1 - u_{X1})$$
$$- \ldots - {}^{m_{XN}}\!/_{R_{XN}}(u_{XN} - u_A) + k_{XN} F_N(u_1 - u_{XN}) = F(t), \tag{1}$$
$$m_{X1} \frac{d^2 u_{X1}}{dt^2} + {}^{m_{X1}}\!/_{R_{X1}}(u_{X1} - u_A) - k_{X1} F_1(u_1 - u_{X1}) + F_{12}(u_{X1}, u_{X2}) + F_{13}(u_{X1}, u_{X3}) = 0,$$
$$m_{X2} \frac{d^2 u_{X2}}{dt^2} + {}^{m_{X2}}\!/_{R_X}(u_{XN} - u_A) - k_X F_N(u_1 - u_{X2}) - F_{12}(u_{X1}, u_{X2}) + F_{23}(u_{X2}, u_{X3}) = 0,$$

$$m_{X3} \frac{d^2 u_{X3}}{dt^2} + {}^{m_{X2}}\!/_{R_X}(u_{XN} - u_A) - F_{13}(u_{X1}, u_{X3}) - F_{23}(u_{X2}, u_{X3}) = 0 .$$

Here three DVA's masses are considered. Parameters m_1, k_1 of the prime system may be found by means of FEM or experimentally [12]. The nonlinear functions are

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

Were A – clearance 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.

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

Let us consider the optimization of this DVA's by criterion

$$CiL = Max(x_1(t)),\; t > t_P \tag{3}$$

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. 2.

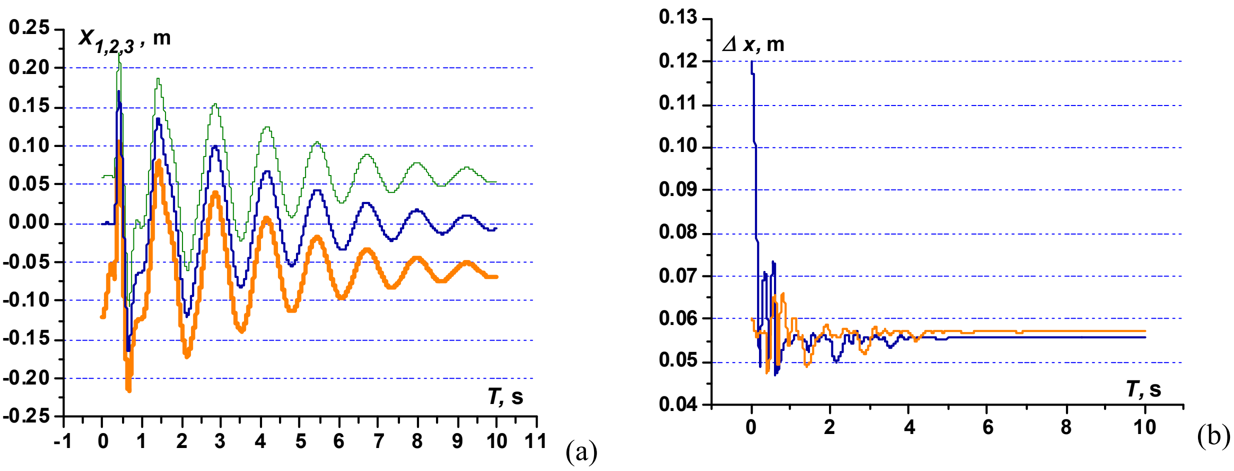


Fig. 2. Coordinates x_1, x_2, x_3 of the impact masses (a);
the differences between this coordinates x_1, x_3 and x_2, x_3 (b).

In Fig. 3 the results of DVA's application is shown

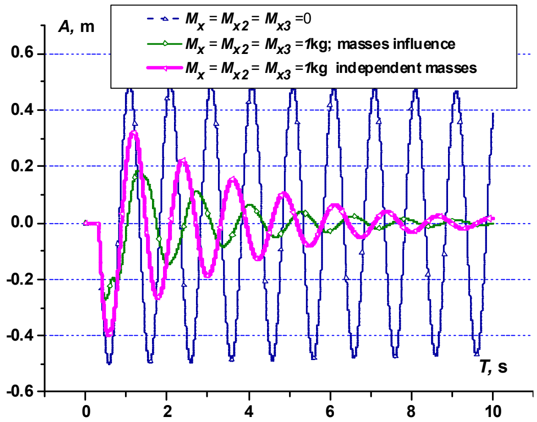


Fig. 3. Results of DVA's application

The 3 mass impacts DVA seems to be better then independent 3 DVA's with the same masses. Here the optimization in the real time is done.

Present research develops the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling [9-13]. The process of geometrical DVA's parameters evolution for different stage of impulse loading and different base system damping is shown in Fig. 4.

N = 1

Dx .317E-01 DG .192E-01 Mx3 .279E+01 Mx .107E+00 CiL .242E-01
fx .532E+00 fKK .983E+00 fEkx .101E+01 Ax .218E+02

N = 2874

Dx .432E-01 DG .433E-01 Mx3 .282E+01 Mx .923E-01 CiL .161E-01
fx .456E+00 fKK .105E+01 fEkx .101E+01 Ax .140E+02

N= 3892

Dx .426E-01 DG .434E-01 Mx3 .282E+01 Mx .911E-01 CiL .161E-01
fx .455E+00 fKK .105E+01 fEkx .101E+01 Ax .140E+02

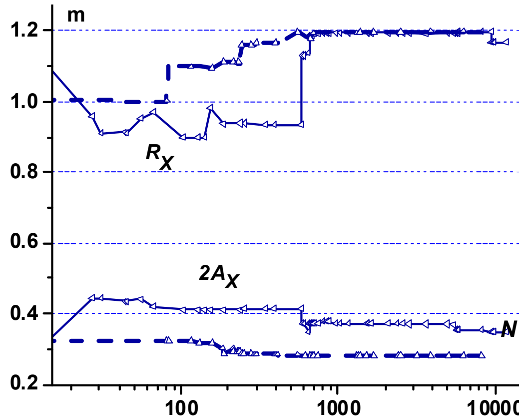


Fig. 4. Process of geometrical DVA's parameters evolution for different stage of impulse loading

Here 8 parameters of optimization are used: fx, fx2 DVA's eigenfrequencies; Dx, DG – proportional viscous damping in container and in barrier (added to all equations terms $k_{Xi} D_{Xi} \, {du_i}/{dt}$), $Mx3$ – less DVA' mass, fKK – DVA's masses inter-collision and fKx – DVA's masses on barrier collision eigenfrequencies . Ax is clearance half length. The prime system mass is $m_1 = 10$kg, the prime system eigenfrequency – $f_R = 1$Hz$=6.28$ Rad/s, the proportional damping $-D1=0.03$.

In Fig. 5 results of one-mass DVA and 3 mass DVA optimization.

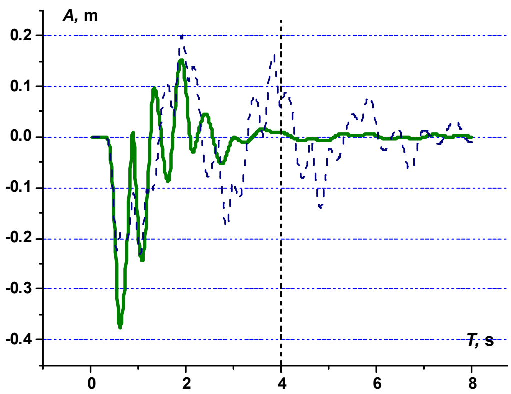


Fig. 5. Results of one-mass DVA (dash line) and 3 mass DVA optimization

The one-mass DVA is worse than 3-mass. The upper results are achieved with the Boltzman approximation for contact forces.

Impact masses DVA with different radiuses of sending plates. Let us consider new 3-mass DVA (Fig. 6).

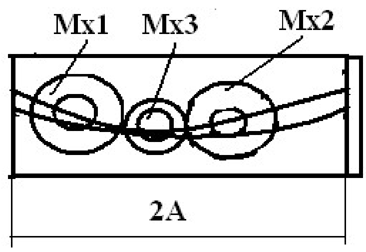


Fig. 6. DVA with 3 different radiuses of sending plates

Here the curvatures of flat springs of DVA's masses are different. That prevents them to move synchronous motion. In Fig. 7 the optimization result are shown

N= 1

Dx .559E-01 Dx2 .137E+00 DG .277E-01 Mx3 .425E+00 CiL .143E+00
fx .819E+00 fx2 .687E+00 fEkx .155E+01 Ax .231E+02

N= 16

Dx .485E-02 Dx2 .120E+00 DG .155E+00 Mx3 .673E+00 CiL .728E-01
fx .568E+00 fx2 .188E+01 fEkx .327E+00 Ax .232E+02

N= 17

Dx .483E-02 Dx2 .134E+00 DG .645E-01 Mx3 .743E+00 CiL .344E-01
fx .120E+00 fx2 .183E+01 fEkx .206E+00 Ax .231E+02

N=23408

Dx .204E-01 Dx2 .119E+00 DG .106E+00 Mx3 .222E+00 CiL .682E-02
fx .161E+00 fx2 .157E+01 fEkx .993E-01 Ax .232E+02

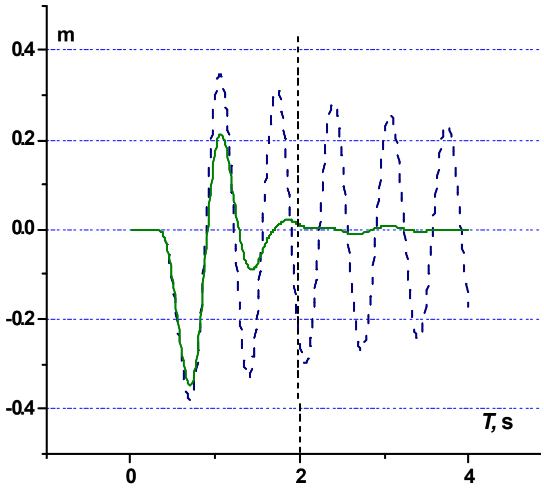


Fig. 7. Optimization results

The evaluation time was 2s. In Fig. 8 comparison of optimization processes for masses contact and without contact are presented .

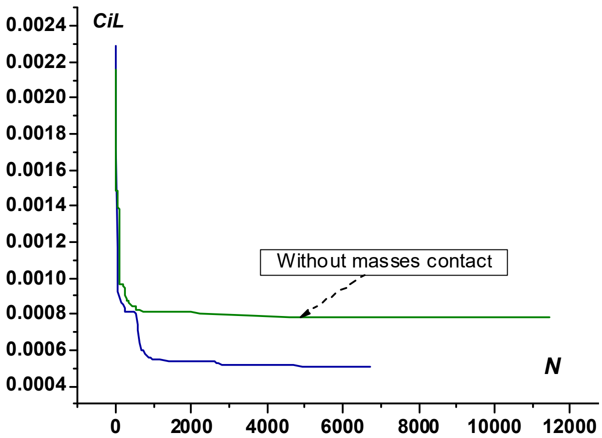
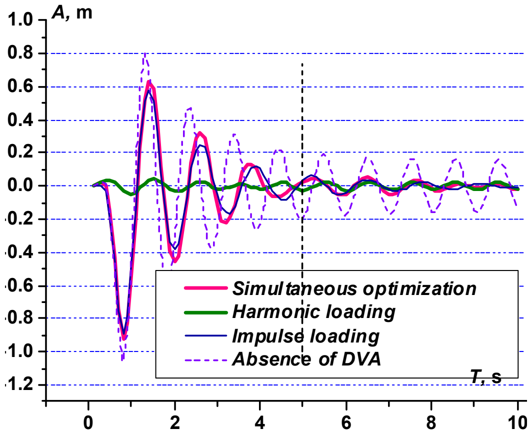
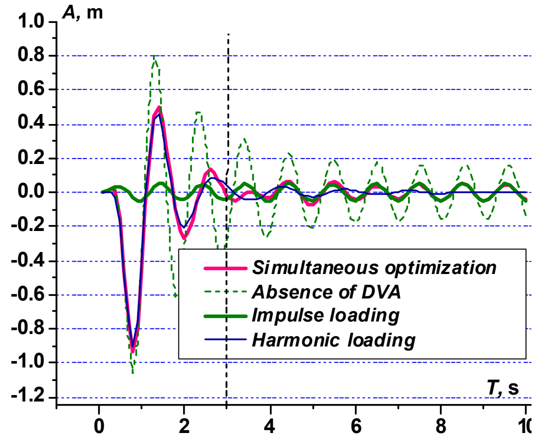


Fig. 8. Comparison of optimization processes for masses contact and without contact

Simultaneous optimization for shock and oscillation loading. Consider now simultaneous optimization by impulse and harmonic loading. Let us now consider the optimization of this DVA's by criterion (3) for simultaneous shock oscillation loading. In Fig. 9 results of optimization for various initial time are presented. For the optimization the best results are achieved (as was shown by calculations) by high damping in the container and soft highly damping barriers.



(a)



(b)

Fig 9. Results of optimization for initial time 5s (a) and 3s (a)

Conclusion. In order to determine the optimal parameters of impact multy-mass 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 impact and narrow frequency excitation. The new vibro-absorbing elements are proposed. Few parameters numerical schemes of vibration analysis are under discussion. The influence of geometric, elastic and damping properties of the basic construction and dynamic vibration absorbers are considered. The algorithms for vibration decreasing are received. The energy dissipation results from the exchange of momentum during impacts between the mass and the stops, mass friction during its motion and masses collegian as the structure vibrates. Finally, present research develops the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling.

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Дівеєв Б.М., Вікович І.А., Дорош І.Р., Коваль Т.Б., Мартин В.Є. **Багатомасовий ударний демпфер з еластичними бар'єрами.**

Ударний демпфер це пасивний контрольний пристрій, який має форму маси, що вільно рухається у контейнері, прикріпленим до підконтрольної конструкції, тобто первинної структури. Демпфування спричиняється обміном імпульсів протягом зіткнень між масою і бар'єрами, оскільки структура вібрує. Багатомасова демпфуюча система, може подолати деякі обмеження звичайного динамічного гасника коливань (ДГК) використовуючи ударні маси як демпфуюче середовище і взаємодію мас з бар'єром як демпфуючого механізму. . У статті запропонований ефективний числовий підхід для максимізації демпфування мод коливань в заданому діапазоні частот для первинної структури. Первинна структура моделюється як пружно-масова система. Розглядається вплив динамічних пружних і демпфуючих властивостей ДГК. Розроблено методику для надання оптимальних ДГК для усунення надмірної вібрації при синусоїдальному і ударному навантаженні системи.

Ключові слова: динамічний гасник вібрації, ударний демпфер, пружно-масова система, оптимізація.

Дивеев Б.М., Викович И.А., Дорош И.Р., Коваль Т.Б., Мартын В.Е. **Многомассовый ударный демпфер с эластичными барьерами.**

Ударный демпфер это пассивное контрольное устройство, которое имеет форму массы, которая свободно двигается в контейнере, прикрепленным к подконтрольной конструкции, то есть первичной структуры. Демпфирование влечется обменом импульсов на протяжении столкновений между массой и барьерами, поскольку структура вибрирует. Многомассовая демпфирующая система, может преодолеть некоторые ограничения обычного динамического гасителя колебаний (ДГК) используя ударные массы как демпфирующую среду и взаимодействие масс с барьером как демпфирующего механизма. В статье предложен эффективный числовой подход для максимизации демпфирования мод колебаний в заданном диапазоне частот для первичной структуры. Первичная структура моделируется как упруго-массовая система. Рассматривается влияние динамических упругих и демпфирующих свойств ДГК. Разработана методика для предоставления оптимальных ДГК для устранения избыточной вибрации при синусоидальной и ударной нагрузке системы..

Ключевые слова: динамический гаситель вибрации, ударный демпфер, упруго-массовая система, оптимизация.

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