Scientific Review – Engineering and Environmental Sciences (2017), 26 (3), 394–400
Sci. Rev. Eng. Env. Sci. (2017), 26 (3)
Przegląd Naukowy – Inżynieria i Kształtowanie Środowiska (2017), 26 (3), 394–400
Prz. Nauk. Inż. Kszt. Środ. (2017), 26 (3)
http://iks.pn.sggw.pl
DOI 10.22630/PNIKS.2017.26.3.38

Ivan KERNYTSKYY[1], Bohdan DIVEYEV[2], Orest HORBAJ[2], Mykhajlo HLOBCHAK[3], Marta KOPYTKO[4], Oleh ZACHEK[5]

[1]Department of Engineering and Geodesy, Warsaw University of Life Sciences – SGGW
[2]Department of Transport Technologies, Lviv Polytechnic National University
[3]Department of Operation and Repair of Motor Vehicle, Lviv Polytechnic National University
[4]Department of Economy, Lviv State University of Internal Affairs
[5]Department of Police, Lviv State University of Internal Affairs

Optimization of the impact multi-mass vibration absorbers

Key words: dynamic vibration absorber (DVA), impact damping system, multi-mass DVA, results of DVA's application, genetic algorithms, Boltzman approximation, DVA optimization, simultaneous optimization, frequency characteristics, energy dissipatio

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 by Paget (1937). Further research by Grubin (1956) determined the existence of the optimal distances between the primary mass and the auxiliary mass for the impact damper. In Masri (1968) 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 Bagpat & Sankar (1985) and Ema & Marui (1994) 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 (Chen & Wang, 2003; Li & Darby, 2006).

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 (Park, Wang & Crocker, 2009).

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 (Kernytskyy, Diveyev, Pankevych & Kernytskyy, 2006; Stocko, Diveyev & Topilnyckyj, 2007; Diveyev, Vikovych, Dorosh & Kernytskyy, 2012; Cherchyk, Diveyev, Martyn & Sava, 2014; Diveyev, Vikovych, Martyn & Dorosh, 2015).

Impact masses DVA

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

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

The system of equations is now:

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

$$
\begin{gathered}
m_{X1} \frac{d^2 u_{X1}}{dt^2} + {m_{X1}}/{R_{X1}} \left(u_{X1} - u_A\right) - \\
- k_{X1} F_1 \left(u_1 - u_{X1}\right) + F_{12} \left(u_{X1}, u_{X2}\right) + \\
+ F_{13} \left(u_{X1}, u_{X3}\right) = 0
\end{gathered}
$$

$$
\begin{gathered}
m_{X2} \frac{d^2 u_{X2}}{dt^2} + {m_{X2}}/{R_X} \left(u_{XN} - u_A\right) - \\
- k_X F_N \left(u_1 - u_{X2}\right) - F_{12} \left(u_{X1}, u_{X2}\right) + \\
+ F_{23} \left(u_{X2}, u_{X3}\right) = 0
\end{gathered}
$$

$$
\begin{gathered}
m_{X3} \frac{d^2 u_{X3}}{dt^2} + {m_{X2}}/{R_X} \left(u_{XN} - u_A\right) - \\
- F_{13} \left(u_{X1}, u_{X3}\right) - F_{23} \left(u_{X2}, u_{X3}\right) = 0
\end{gathered}
$$

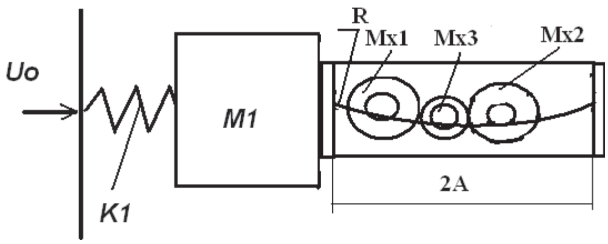

FIGURE 1. DVA with three different impact masses

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 (Kernytskyy et al., 2006). The non-linear functions are:

$$\begin{aligned} &F_i = -K_{vi}\left(x_i - A_i\right) \quad \left|x_x\right| > A_i \\ &F_i = 0 \quad \left|x_i\right| < A_i \\ &F\left(t\right) = a\sin\left(\omega t\right) \end{aligned} \tag{2}$$

were:
A – clearance;
K_{vi} – boundary elements rigidity.

The non-linear functions $F_{13}(u_{X1}, u_{X3})$, $F_{23}(u_{X2}, u_{X3})$, of DVA's masses interaction may be defined analogously.

$$\begin{aligned} &F_{13} = F_{13}\left(x_1 - x_3\right) \quad \left|x_1 - x_3\right| < R_1 + R_3 \\ &F_{13} = 0 \quad \left|x_1 - x_3\right| > R_1 + R_3 \end{aligned}$$

$$\begin{aligned} &F_{23} = F_{23}\left(x_2 - x_3\right) \quad \left|x_2 - x_3\right| < R_2 + R_3 \\ &F_{23} = 0 \quad \left|x_2 - x_3\right| > R_2 + R_3 \end{aligned}$$

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

$$CiL = \max\left[x_1(t)\right], t > t_P \tag{3}$$

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 Figure 2.

In Figure 3 the results of DVA's application are shown.

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

Present research develops the genetic algorithms for optimal design search-

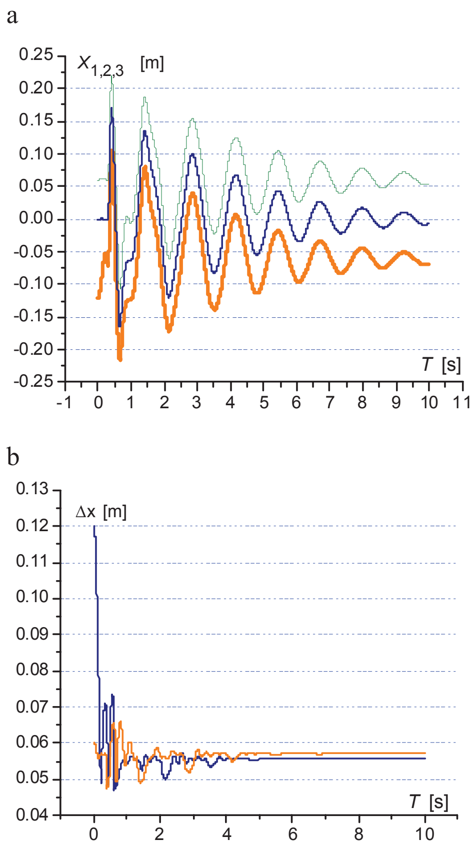

FIGURE 2. Coordinates x_1, x_2, x_3 of the impact masses (a); the differences between this coordinates x_1, x_3 and x_2, x_3 (b)

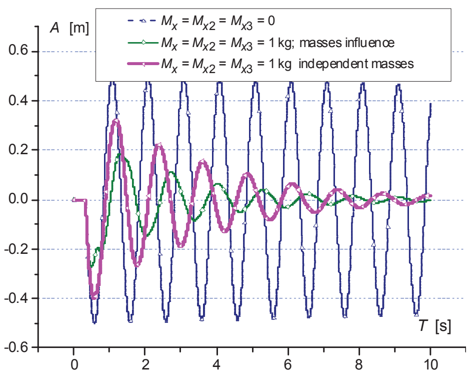

FIGURE 3. Results of DVA's application

ing by discrete-continuum DVA's system – base system modeling (Chen & Wang, 2003; Kernytskyy et al., 2006; Li & Darby, 2006; Stocko et al., 2007). The process of geometrical DVA's parameters evolution for different stage of impulse loading and different base system damping is shown in Figure 4.

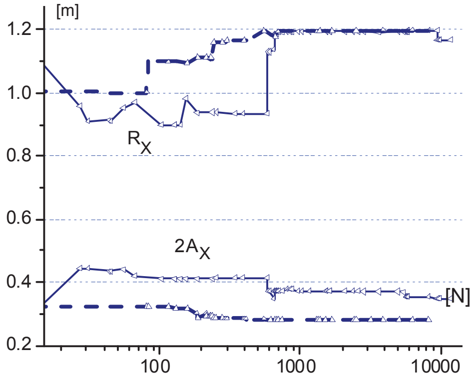

FIGURE 4. Process of geometrical DVA's parameters evolution for different stage of impulse loading

Here eight parameters of optimization are used: *fx*, *fx*2 – DVA's eigenfrequencies; *Dx*, *DG* – proportional viscous damping in container and in barrier (added to all equations terms $k_{Xi}D_{Xi}\, {du_i}/{dt}$); Mx_3 – 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 is f_R = 1 Hz = = 6.28 rad·s^{-1}, the proportional damping is $D1$ = 0.03.

In Figure 5 results of one-mass DVA and three-mass DVA optimization.

The one-mass DVA is worse than three-mass.

The upper results are achieved with the Boltzman approximation for contact forces (Fig. 6).

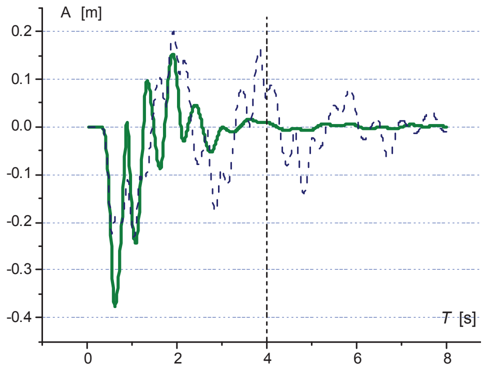

FIGURE 5. Results of one-mass DVA (dash line) and three-mass DVA optimization

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

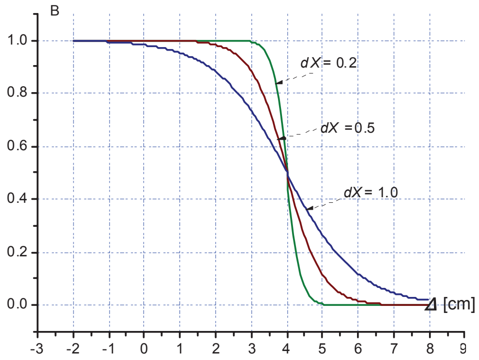

FIGURE 6. Boltzman contact forces (B) approximation

Here $A_1 = 1$, $A_2 = 2$, $X_0 = 4$, $x = 2X_0 - \Delta$ where:

Δ – distance between centers of rolling masses;

X_0 – width of contact zone.

Impact masses DVA with different radiuses of sending plates

Let us consider new three-mass DVA (Fig. 7).

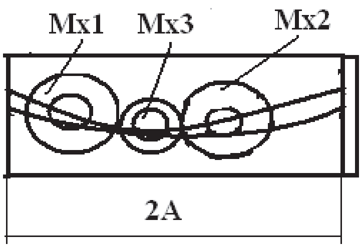

FIGURE 7. DVA with two different radiuses of sending plates

Here the curvatures of flat springs of DVA's masses are different. That prevents them to move synchronous motion. In Figure 8 the optimization results are shown.

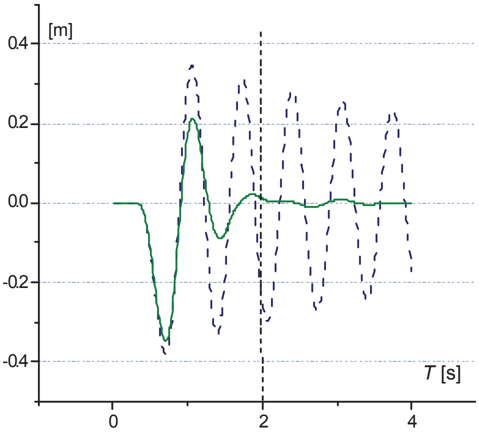

FIGURE 8. Optimization results

The evaluation time was 2 s. In Figure 9 the masses rejections are presented.

Here dX = 0.5 in Boltzman approximation.

In Figure 10 comparison of optimization processes for masses contact and without contact are presented.

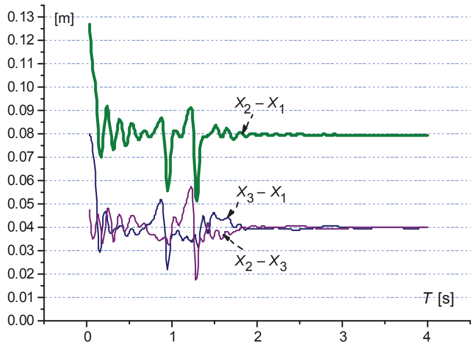

FIGURE 9. Masses rejections

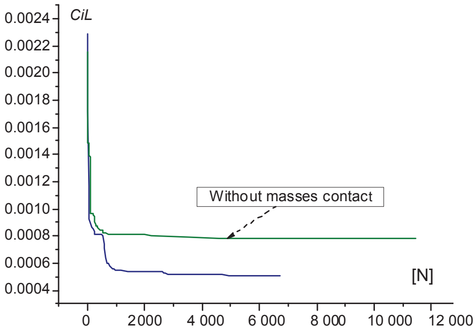

FIGURE 10. Comparison of optimization processes for masses contact and without contact

Simultaneous optimization for shock 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 Figure 11 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.

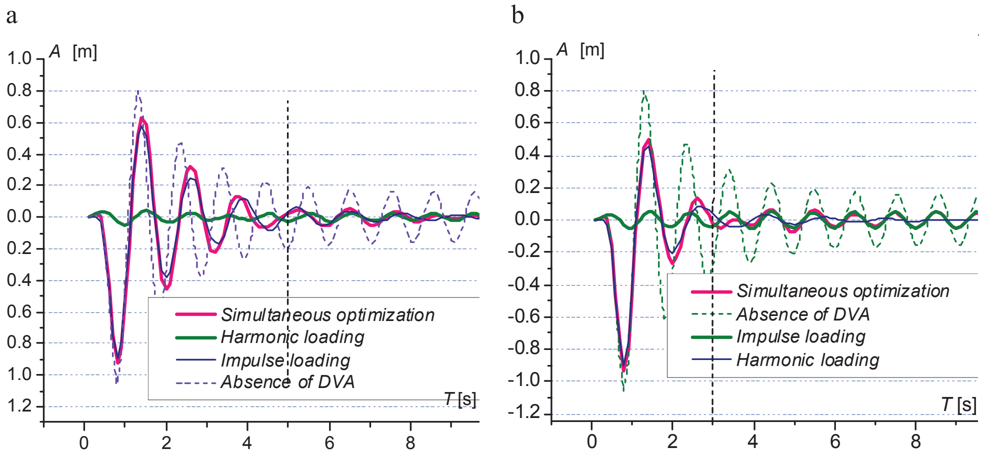

FIGURE 11. Results of optimization for initial time 5 s (a) and 3 s (b)

Conclusion

In order to determine the optimal parameters of impact multi-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.

References

Bagpat, C.N. & Sankar, S. (1985). Single unit impact damper in free and forced vibration. *Journal of Sound and Vibration*, *99*, 85-94.

Chen, C.C. & Wang, J.W. (2003). Free vibration analysis of a resilient impact damper. *International Journal of Mechanical Science*, *45*, 589-604.

Cherchyk, H., Diveyev, B., Martyn, V. & Sava, R. (2014). Parameters identification of particle vibration absorber for rotating machines. *Proceeding of ICSV21, Beijing, China* (electronic edition).

Diveyev, B., Vikovych, I., Dorosh, I. & Kernytskyy, I. (2012). Different type vibration absorbers design for beam-like structures. *Proceeding of ICSV19, Vilnius, Lithuania 2,* 1499-1507 (electronic edition).

Diveyev, B., Vikovych, I., Martyn, V. & Dorosh, I. (2015). Optimization of the impact and particle vibration absorbers. *Proceeding of ICSV21, Florence, Italy*, *2* (electronic edition).

Ema, S. & Marui, E. (1994). A fundamental study on impact dampers. *International Journal of Machine Tools and Manufacturers*, *34*, 407-421.

Grubin, C. (1956). On the theory of acceleration damper. *Journal of Applied Mechanics, Transactions of the ASME*, *78,* 373-378.

Kernytskyy, I., Diveyev, B., Pankevych, B. & Kernytskyy, N. (2006). Application of variation-analytical methods for rotating machine dynamics with absorber. *Electronic Journal of Polish Agricultural Universities, Civil Engineering*, *9(4)*. Available Online http://www.ejpau.media.pl.

Li, K. & Darby, A.P. (2006). An experimental investigation into the use of a buffered impact damper. *Journal of Sound and Vibration*, *291*, 844-860.

Masri, S.F. (1968). Analytical and experimental studies of multi-unit impact dampers. *Journal of the Acoustical Society of America*, *45*, 1111-1117.

Paget, A.L. (1937). Vibration in steam turbine buckets and damping by impacts. *Engineering*, *143,* 305-307.

Park, J., Wang, S. & Crocker, M.J. (2009). Mass loaded resonance of a single unit impact damper caused by impacts and the resulting kinetic energy influx. *Journal of Sound and Vibration*, *323*, 877- 895.

Stocko, Z., Diveyev, B. & Topilnyckyj, V. (2007). Discrete-cotinuum methods application for rotating machine-absorber interaction analysis. *Journal of Achievements in Materials and Manufacturing Engineering, 20, 1-2,* 387-390.

Summary

Optimization of the impact multi--mass vibration absorbers. The problem of attaching dynamic vibration absorber (DVA) to a discrete multi-degree-of-freedom or continuous structure has been outlined in many papers and monographs. 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 with the several of the impact masses. Such originally designed absorbers reduce vibration selectively in maximum vibration mode without introducing vibration in other modes. 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. The paper contemplates the provision of the impact multi-mass DVA's with masses collisions for additional damping. For some cases of DVA optimization such a design seems more effective than conventional multi-mass DVA with independent mass moving. A technique is developed to give the optimal DVA's for the elimination of excessive vibration in harmonic stochastic and impact loaded systems.

Authors' addresses:

Ivan Kernytskyy
Szkoła Główna Gospodarstwa Wiejskiego
Wydział Budownictwa i Inżynierii Środowiska
Katedra Inżynierii Budowlanej
Nowoursynowska 159, 02-776 Warszawa
Poland

Bohdan Diveyev, Orest Horbaj
Lviv Polytechnic National University
Department of Transport Technologies
St. Bandery 12, Lviv
Ukraine
e-mail: divboglviv@yahoo.com

Mykhajlo Hlobchak
Lviv Polytechnic National University
Department of Operation and Repair of Motor Vehicle
St. Bandery 12, Lviv
Ukraine

Marta Kopytko
Lviv State University of Internal Affairs
Department of Economy
St. Horodocka 29, Lviv
Ukraine

Oleh Zachek
Lviv State University of Internal Affairs
Department of Police
St. Horodocka 29, Lviv
Ukraine