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OPTIMIZATION OF IMPACT DYNAMIC VIBRATION ABSORBERS IN THE FREQUENCY BAND

The paper deals with the methods of calculation and optimization of constructions of the impact dynamic vibration absorbers (DVA) for the elongated structures. 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 the tuned-mass impact systems. Methods of decomposition and numerical synthesis are considered on the basis of an 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 systems.

Key words: dynamic vibration absorber, impact, optimization, frequency range

Introduction. 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.

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 order to determine the optimal parameters of an absorber the need for complete modelling 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 [17–19].

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.

Dynamic equations. Problem of vibration fields modeling of complicated designs deformation and strain is considered for the purposes of dynamic absorption. 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. The essence of the first method consists in reviewing knots of junctions as compact discrete elements A_i^n for which inertial properties are taken into account without reviewing their strain, and massive connected parts - as deformable elements A_i^c, their inertion being taken into account on the basis of modal expansion.

For every point X=(x, y, z) of A_i^c we have

$$U_i(t,X)=\begin{bmatrix} q_{1i}(t)\varphi_{1i}(X) \\ \dots \\ q_{ni}(t)\varphi_{ni}(X) \end{bmatrix}. \quad (1)$$

Here $\varphi_{1i}(X),\dots,\varphi_{ni}(X)$ are coordinate functions, $q_{1i}(t),\dots,$ $q_{ni}(t)$ – corresponding independent time functions. By variation of strain U_i^c and kinetic K_i^c energies for A_i^c we have

$$\delta U_i^c=(K_i^{uc}\cdot q_i)^T\cdot\delta q_i\ ,\ \delta K_i^c=(M_i^{uc}\cdot q_i)^T\cdot\delta q_i\ . \qquad q_i=[q_{1i},q_{2i},\dots,q_{ni}]^T . \quad (2)$$

By variation of strain U_i^n and kinetic K_i^n energies for connecting and attached discrete element A_i^n we have

$$dU_i^n=k_{ij}(q_{ij}^n(t)-q_j(t)j_j(X_{ij}))\times(dq_{ij}^n(t)-dq_j(t)j_j(X_{ij})). \quad (3)$$

Here X_{ij} are point of contact of discrete element A_i^n and continual element A_j^c and k_{ij} – corresponding rigidity of connection. For the mass-less joints of continual elements we must add to the strain energy such terms

$$dU_i^n=k_{ij}(q_i(t)j_i(X_{ij})-q_j(t)j_j(X_{ij}))\quad\cdot(dq_i(t)j_i(X_{ij})-dq_j(t)j_j(X_{ij})). \quad (4)$$

Kinetic energy variation of discrete one-mass element A_i^n is

$$dK_i^n=m_i\,\dot{q}_i^n\cdot d\,\dot{q}_i^n\ . \quad (5)$$

By Hamilton-Ostrogradsky variation equation

$$\int_{t_0}^{t_1}(dU-dK)dt=F\ ,$$

equating terms by independent variation parameters in (2-5) we obtain [20–22]

$$M\ddot{q}+C\dot{q}+\cdot Kq=F\ , \quad (6)$$

a set of ordinary differential equations. Here the viscous damping is added by means of matrix *C*. *M* is the mass matrix and *K* – rigidity matrix.

Elongated element with the impact DVA. Let us consider condensed model of DVA-elongated element system. For the element modeling let us consider uniform Timoshenko beam. The kinematical hypothesis are (for pure bending) are

$$U(X,Y,Z,t) = g\,(x,t)\cdot Z, \quad W(X,Y,Z,t) = w(x,t). \tag{7}$$

By substitution of (7) into the variation Hamilton-Ostrogradsky equation

$$\int_0^L \left(EI\frac{\partial g}{\partial x} d\frac{\partial g}{\partial x} + GF\left(g + \frac{\partial W}{\partial x}\right)dg + rI\frac{\partial^2 g}{\partial t^2}dg + GF\left(g + \frac{\partial W}{\partial x}\right)d\frac{\partial W}{\partial x} + rF\frac{\partial^2 W}{\partial t^2}dW \right)dx = F \tag{8}$$

and taking the modal series expansion for the functions:

$$g(x,t) = \sum_1^N q_i(t) g_i(x), \;\; w(x,t) = \sum_1^N p_i(t) g_i(x), \tag{9}$$

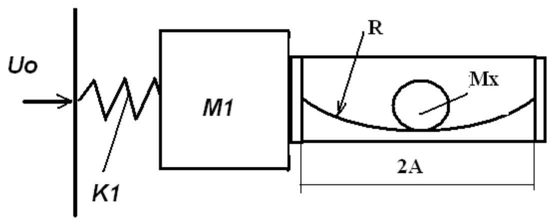
we obtain a set of ordinary differential equations for unknown time dependent functions (written in matrix form)

$$[M]\frac{d^2\vec{r}}{dt^2} + [C]\vec{r} = \vec{f}\,. \tag{10}$$

Here [M] and [C] are well known mass and rigidity matrix, $\vec{r} = \left(\vec{q}, \vec{p}\right)$ – vector of unknown functions, $\vec{f}$ vector of outer forces. Vectors *F* or *f* consists of two parts 1: *Fe* or *f* e –beam dynamic loading; 2: *Fz* or *f* z – beam DVA connections terms ($F = F_e + F_Z$ *or* $f = f_e + f_Z$).

Condensed model. In Fig. 1 the condensed model: elongated element – impact mass DVA is presented with an additional impact mass in container with elastic barrier elements

Fig. 1. Pendulum type DVA with the additional elements



Here *M1* and *K1* are mass and rigidity parameters of condensed model. These parameters may be found theoretically, considering, for example, the first mode of beam vibration in (7-10), or by FEM, or by combining experimental and theoretical results of investigations [20-23]. The system of equations in the condensed rangy is obtained

$$\begin{aligned} &m_1\frac{d^2u_1}{dt^2} + k_1(u_1 - u_0) + k_A(u_1 - u_A) - {m_{X1}}/{R_{X1}}(u_{X1} - u_A) + k_{X1}F_1(u_1 - u_{X1}) - \\ &-\ldots - {m_{XN}}/{R_{XN}}(u_{XN} - u_A) + k_{XN}F_N(u_1 - u_{XN}) = F(t), \end{aligned} \tag{11}$$

$$m_{X1}\frac{d^2u_{X1}}{dt^2}+{m_{X1}}\Big/{R_{X1}}\left(u_{X1}-u_A\right)-k_{X1}F_1\left(u_1-u_{X1}\right)=0,$$

$$\ldots$$

$$m_{XN}\frac{d^2u_{XN}}{dt^2}+{m_{XN}}\Big/{R_{XN}}\left(u_{XN}-u_A\right)-k_{XN}F_N\left(u_1-u_{XN}\right)=0\,.$$

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

$$F_i=-K_{vi}\left(x_i-A_i\right)\quad \left|x_x\right|>A_i,\qquad F_i=0\quad \left|x_i\right|<A_i;\; F(t)=a\sin(wt). \tag{12}$$

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

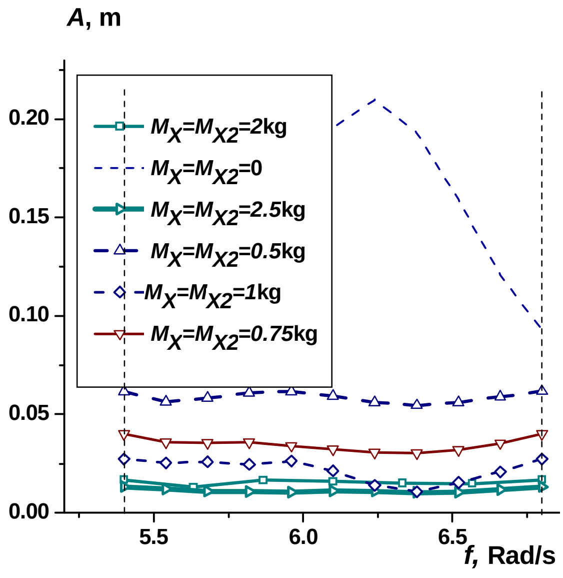
Numerical results, optimization. DVA's are appropriately optimized by genetic algorithms near the beam first eigen-frequency f_R. The evaluation function is

$$CiL=Max\left(u_1\left(f\right)\right),\ af_R<\mathrm{f}<bf_R, \tag{13}$$

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: 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 – *D1=0.03*. For system with two dangerous frequency intervals the grate number of DVA's may be used (Fig. 3)

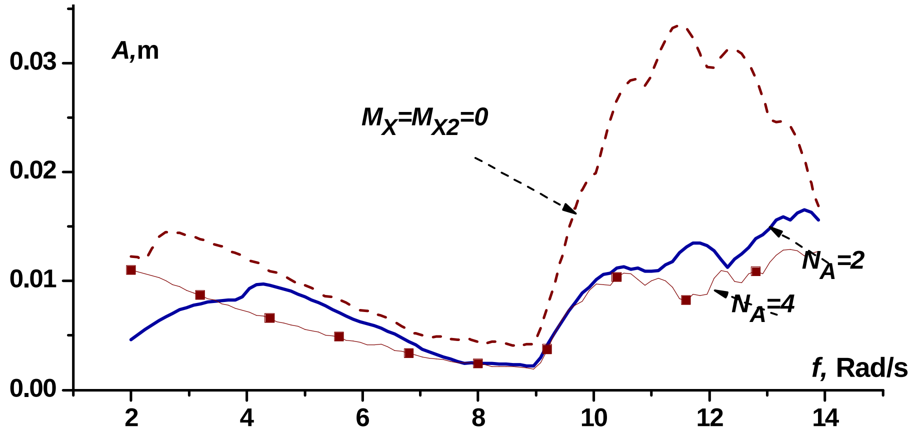


Fig. 3. The results of optimization for system with two frequency intervals by number of DVA's $N_A = 2, 4$
For $N_A = 4$ *the better result may be seen*

For the four DVA's better result may be seen then for two in the two frequency intervals. The DVA's mass sum is equal to 4kg in both cases.

Conclusions. In order to determine the optimal parameters of DVA the complete modeling of dynamics of devices should be made. Traditional design methodology, based on decoupling models of structures and machines are not effective for vibration decreasing since they do not give a possibility to determine vibration levels. Paper deals with the new methods for the explicit determination of the frequency characteristics of the impact dynamic vibration absorbers by narrow frequency excitation. Few parameters numerical schemes of vibration analysis are under discussion. The new vibroabsorbing elements are proposed. Present research develops a modern prediction methodology, based on coupled theory. The result may be highly improved by applying the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling.

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