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OPTIMIZATION OF PARTICLE DYNAMIC VIBRATION ABSORBERS IN THE FREQUENCY BAND

Few parameters numerical schemes of dynamic vibration absorbers (DVA) and base construction vibration analysis are under discussion. The discrete-continue models of machines dynamics of such machines as water pump with the attachment of dynamic vibration absorbers are offered. The algorithms for vibration decreasing of pump are received. The new vibroabsorbing elements with particle filled containers are proposed.

Key words: *dynamic vibration absorber, vibration decreasing, pump, discrete-continue models, particle filled containers.*

Abstract. The paper contemplates the provision of DVA's or any number of such absorbers. Such originally designed absorbers reduce vibration selectively in maximum vibration mode without introducing vibration in other modes. In the classical theory of DVA, the primary structure is modelled as a spring-mass system; however, other models also have high interesting research and engineering application. 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.

1. Introduction. Noise and vibration are of concern with many mechanical systems including industrial machines, home appliances, transportation vehicles, and building structures. Many such structures are comprised of beam and plate like elements. The vibration of beam and plate systems can be reduced by the use of passive damping, once the system parameters have been identified. Rotating machines in the most performance with movable parts is often affected by a rotor-base interaction in associatation with dissipative effects.

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.

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 [17-19].

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 [20-22]. 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.

2. Dynamic equations for the pump with DVA's. In Fig.1 the scheme of pump structure *P* with absorbers attachment is presented

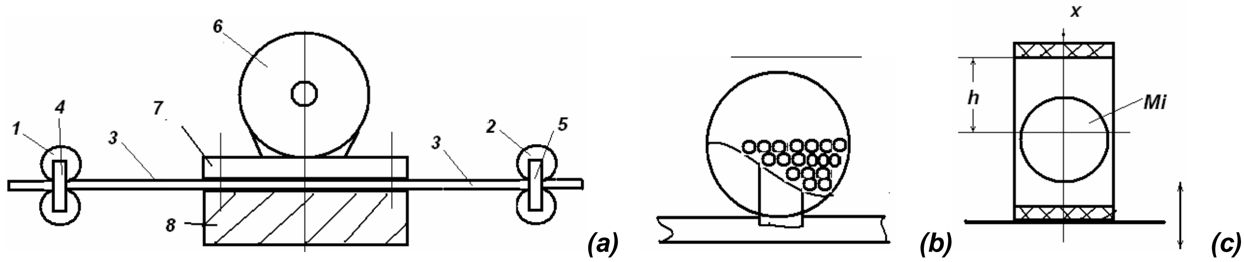


Fig. 1. Pump – DVA scheme (a); DVA filled container(b); container model (c)

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 [20-22]. The system of equations in the condensed rangy is obtained:

$$m_1 \ddot{w}_0 + (k_1 D_K + k_A D_A + k_{A2} D_{A2})\dot{w}_0 + (k_1 + k_A + k_{A2}) w_0 -$$
$$- k_A D_A \dot{w}_A - k_A D_A \dot{w}_A - k_A w_A - k_{A2} w_{A2} = F$$

$$m_A \ddot{w}_A + k_A D_A \dot{w}_A + k_A w_A - k_A D_A \dot{w}_0 - k_A w_0 = 0 ; \quad (1)$$

$$m_{A2} \ddot{w}_{A2} + k_{A2} D_{A2} \dot{w}_{A2} + k_{A2} w_{A2} - k_{A2} D_{A2} \dot{w}_0 - k_{A2} w_0 = 0.$$

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

$$m_i \ddot{w}_i + C_i \dot{w}_i + k_G(x)(w_i - w_0) + C_G(x)\left(\dot{w}_i - \dot{w}_0\right) =$$
$$= 0 \quad |w_i - w_0| > |h - R|$$

$$m_i \ddot{w}_i + C_i \dot{w}_i = 0 \qquad |w_i - w_0| \le |h - R| , \quad (2)$$

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.

Even though there have been some investigations (for example [17-19]) into the dynamic behaviour of particle dampers over a wide range of frequencies, the excitation amplitude and mass ratio of the auxiliary mass have normally been restricted so that either a shift in the resonance frequency does not occur, or the shift in the value is so small that the shift is not noticeable.

Elastic parameters of DVA's springs may be corrected on the refined estimation of mechanical properties of beams [23-28], as well on the refined calculation of clamp conditions [29-31]. The important part of damping for such DVA's takes clamp friction.

3. Optimization. Genetic algorithms (GA). 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

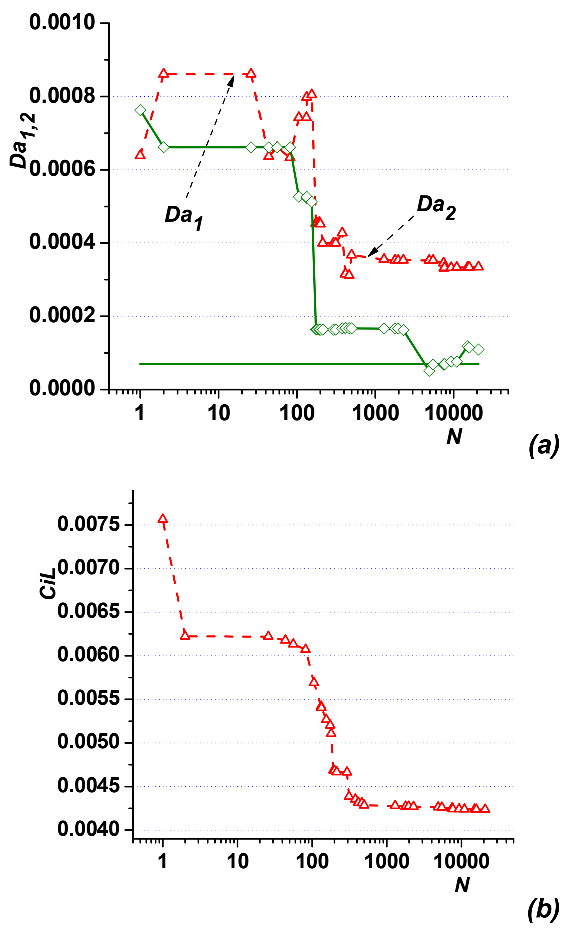
$$Fcil = \max_{f_1<\omega_1<f_2}\left(\int_{f_1}^{f_2} |u_1(f)| P(f) df\right) \qquad (3)$$

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

$$m_A + m_{A2} = 3.8\,kg$$

In Fig. 2. results of optimization are presented.



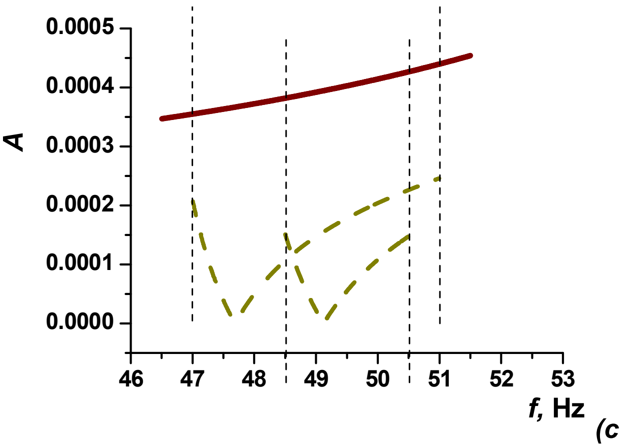


Fig. 2. Result of optimization:
(a) – DVA's damping coefficients evolution;
(b) – Fcil evolution; (c) – optimal FRF of base (for different frequency band), solid line system without DVA;s)

By the appropriate weight function $P(f)$ in (3) more uniform as in Fig. 2c absorbing may be reached. For example, such weight function

$$P(f) = Max(u_1) + (u_1(f) - 0.0001)**3./10.$$
$$f_1 + {(f_2 - f_1)}/{2} < f < f_2$$

$$P(f) = Max(u_1).\quad f_1 < f < +{(f_2 - f_1)}/{2} \qquad (4)$$

Result of optimization for such weight function is presented in Fig. 3.

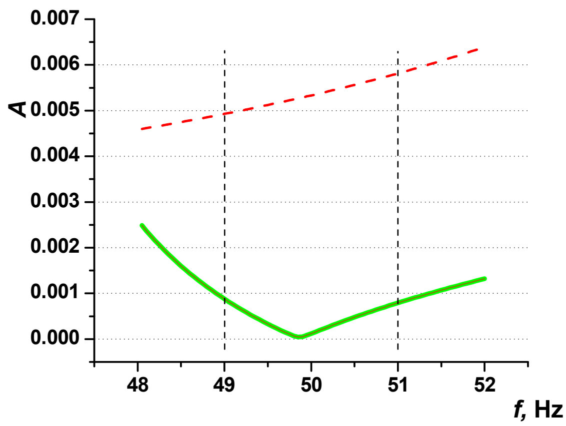
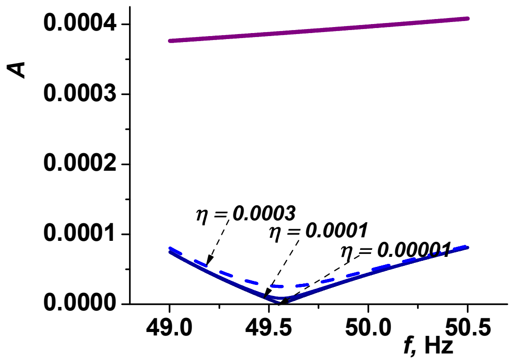


Fig. 3. Result of optimization for weight function (3)

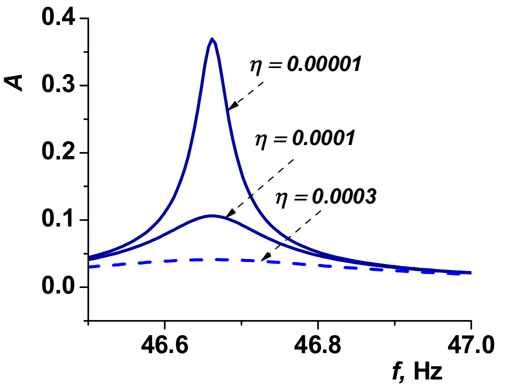
Finally, let us discuss the question of DVA strength. Influence of different DVA parameters was before explored on his efficiency. If rigidity parameters influence very substantially, damping parameters considerably less influential. At the same time it follows to expect the considerable influencing 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

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

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.4a are shown displacements of base construction at small and some greater damping. In Fig. 4b proper DVA's displacemets are shown.



(a)



(b)

Fig. 4. Displacements by different DVA's damping: (a) – base construction vibration; (b) – DVA vibration

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.

Conclusions. In order to determine the optimal parameters of DVA the complete modelling 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 dynamic vibration absorbers 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 influence of dynamic vibration absorbers and basic design elastic and damping properties is under discussion. The one of the tasks of this work is to analyze parameters identification of the dynamic vibration absorber and develop new numerical schemes. The discrete-continue models of machines dynamics of such machines as water pump with the attachment of dynamic vibration absorbers are offered. The algorithms for vibration decreasing of pump are received. 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 modelling.

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ОПТИМІЗАЦІЯ ЧАСТИНОК ДИНАМІЧНИХ ГАСНИКІВ КОЛИВАНЬ В ДІАПАЗОНІ ЧАСТОТ

Анотація. *Розглядаються малопараметричні схеми аналізу вібрації динамічних гасників коливань (ДГК) та основної конструкції. Представлені дискретно-континуальні моделі динаміки таких машин як водяні насоси з приєднаними динамічними гасниками коливань. Отримані алгоритми зменшення вібрації насосів. Запропоновані нові віброполинаючі елементи – ДГК з заповненими частинками контейнерами.*

Ключові слова: *динамічний гасник коливань, зменшення вібрації, насос, дискретно-континуальні моделі, заповнені частинками контейнери.*

ОПТИМИЗАЦИЯ ЧАСТИЦ ДИНАМИЧЕСКИХ ГАСИТЕЛЕЙ КОЛЕБАНИЙ В ДИАПАЗОНЕ ЧАСТОТ

Аннотация. *Рассматриваются малопараметрични схемы анализа вибрации динамических гасителей колебаний (ДГК) и основной конструкции. Представлены дискретно-континуальные модели динамики таких машин как водяные насосы с присоединенными динамическими гасители колебаний. Полученные алгоритмы уменьшения вибрации насосов. Предложены новые виброполглощающие элементы - ДГК с заполненными частицами контейнерами.*

Ключевые слова: *динамический гасители колебаний, уменьшение вибрации, насос, дискретно-континуальные модели, заполненные частицами контейнеры.*