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USING OF DYNAMIC VIBRATION ABSORBERS FOR REGULATION OF VIBRATING COMPACTOR VIBRATION PROPERTIES

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Abstract. A new mathematical model for complicated machine constructions dynamic properties calculation and optimization by DVA is proposed. The essence of the method is to introduce the few parameter models such that the differential equations for machines construction and there dynamic properties optimization can be effectively analyzed.

Key words: mathematical model, vibrating compactor, dynamic vibration absorber (DVA)

INTRODUCTION

The development of models for flexible multibody systems aims at an accurate description of deformations and high accuracy as well as efficiency of the used discretization method. The discrete models are totally inadequate to calculate the natural frequencies of vibration of the complicated machines constructions 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 these constructions through more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination. The numerical schemes (NS) row is considered for such complex vibro-exitated constructions. Methods of decomposition and the NS synthesis are considered on the basis of new methods of modal synthesis. Complex NS are provided of discretely-continual type that enables in the adaptive mode to calculate tension not only in the construction elements, but in places of most their concentration – in joints. It is considered also numerical schemes

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for research of joint elements that also are got on the basis of kinematics hypotheses. On the basis of simple and more complex NS research of local tensions on verge of stratified structure at the different kinds of its fixing is conducted. Traditional design methodology, based on discontinuous models of structures and machines is not effective for high frequency vibration. Present research develops a modern prediction and control methodology, based on complex continuum theory and application of special frequency characteristics of structures. Complex continuum theory allows to take into consideration system anisotropy, supporting structure strain effect on equipment motions and to determine some new effects that are not described by ordinary mechanics of the continuum theory.

In order optimal parameters of dynamic vibration absorber (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.

DYNAMIC EGUATION

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:

$$U_i\left(t, X\right) = \begin{bmatrix} q_{1i}(t)\varphi_{1i}(X) \\ \ldots \\ q_{ni}(t)\varphi_{ni}(X) \end{bmatrix} \tag{1}$$

where: $\varphi_{1i}(X), \ldots, \varphi_{ni}(X)$ are coordinate functions,
$q_{1i}(t), \ldots, q_{ni}(t)$ – corresponding independent time functions.

By variation of strain U_i^c and kinetic K_i^c energies for A_i^c we have:

$$\delta U_i^c = (K_i^{uc} \cdot q_i)^T \cdot \delta q_i \, ; \; \delta K_i^c = (M_i^{uc} \cdot q_i)^T \cdot \delta q_i \tag{2}$$

where $q_i = \left[q_{1i}, \, q_{2i}, \, \ldots, \, q_{ni}\right]^T$.

By variation of strain U_i^n and kinetic K_i^n energies for connecting and attached discrete elementwe have:

$$\delta U_i^n = k_{ij}\left[q_{ij}^n(t) - q_j(t)\varphi_j\left(X_{ij}\right)\right]\cdot\left[\delta q_{ij}^n(t) - \delta q_j(t)\varphi_j\left(X_{ij}\right)\right] \tag{3}$$

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:

$$\delta U_i^n = k_{ij}\left[q_i(t)\varphi_i\left(X_{ij}\right) - q_j(t)\varphi_j\left(X_{ij}\right)\right]\cdot\left[\delta q_i(t)\varphi_i\left(X_{ij}\right) - \delta q_j(t)\varphi_j\left(X_{ij}\right)\right] \tag{4}$$

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

$$\delta K_i^n = m_i\,\dot{q}_i^n \cdot \delta\dot{q}_i^n \tag{5}$$

By Hamilton-Ostrogradsky variation equation:

$$\int_{t_0}^{t_1}(\delta U - \delta K)dt = 0$$

equating terms by independent variation parameters in (2)–(5) we obtain:

$$(M\cdot\ddot{q} + \overline{K}\cdot q)\cdot\delta q = 0 \tag{6}$$

a set of ordinary differential equations.

NUMERICAL EXAMPLE

In Figure 1a the scheme of vehicle with DVA system is presented. In Figure 1b the corresponding one-half discrete model is presented.

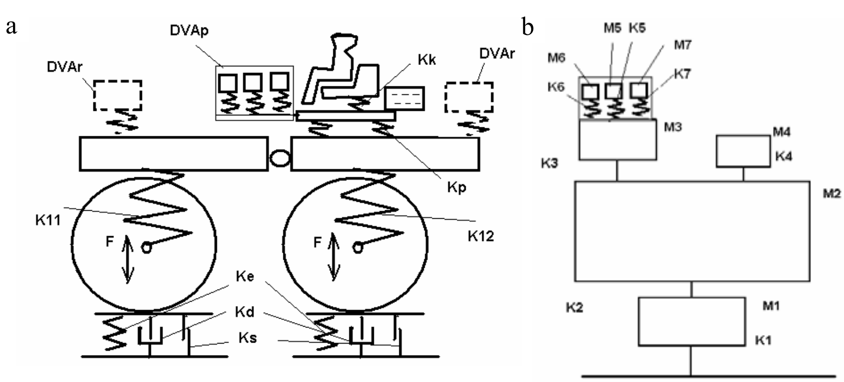

Fig. 1. Scheme of the vehicle with the DVA system (a), discrete scheme of one-half vehicle (b)
Rys. 1. Schemat maszyny z systemem dynamicznej amortyzacji wibracji (a), model dyskretny jednej części maszyny (b)

Consider now the classical discrete scheme of one-half vehicle (Fig. 1b). At the first step of investigation such an effects, as nonlinear effects in dynamic wheel-ground contact zone, horizontal vibration interaction of roller elements, dynamic properties of human body are not considered. The governing equations are:

$$
\begin{aligned}
& m_1 \frac{d^2 u_1}{dt^2} + k_1 (u_1 - u_0) + k_2 (u_1 - u_2) = F(t) \\
& m_2 \frac{d^2 u_2}{dt^2} + k_2 (u_2 - u_1) + k_3 (u_2 - u_3 + w_1) + k_4 (u_2 - u_4) = 0 \\
& m_3 \frac{d^2 u_3}{dt^2} + k_3 (u_3 - w_1 - u_2) + k_5 (u_3 + w_1 \varphi_1 (x_5, y_5) - u_5) + \\
& + k_6 (u_3 + w_1 \varphi_1 (x_6, y_6) - u_6) + k_7 (u_3 + w_1 \varphi_1 (x_7, y_7) - u_7) = 0 \\
& m_{W_1} \frac{d^2 w_1}{dt^2} - k_3 (u_3 - w_1 - u_2) + k_5 (u_3 + w_1 \varphi_1 (x_5, y_5) - u_5) \varphi_1 (x_5, y_5) + \\
& + k_6 (u_3 + w_1 \varphi_1 (x_6, y_6) - u_6) \varphi_1 (x_6, y_6) + k_7 (u_3 + w_1 \varphi_1 (x_7, y_7) - u_7) \varphi_1 (x_7, y_7) + \\
& + \omega_1{}^2 m_{W_1} w_1 = 0 \\
& m_4 \frac{d^2 u_4}{dt^2} + k_4 (u_4 - u_2) = 0 \\
& m_5 \frac{d^2 u_5}{dt^2} + k_5 (u_5 - u_3 - w_1 \varphi_1 (x_5, y_5)) = 0 \\
& m_6 \frac{d^2 u_6}{dt^2} + k_6 (u_6 - u_3 - w_1 \varphi_1 (x_6, y_6)) = 0 \\
& m_7 \frac{d^2 u_7}{dt^2} + k_7 (u_7 - u_3 - w_1 \varphi_1 (x_7, y_7)) = 0
\end{aligned}
\tag{7}
$$

Parameters m_i, k_i are shown in Figure 1b, u_i are s of masses m_i, m_{W1} is an effective dynamic mass of deformable form of platform vibration (displacement w_1), $\varphi_1(x_i, y_i)$ are the kinematic coefficient of DVA's connection points (the ratio connection point – maximum platform displacement.

Here only one continual element is considered – the driver seat platform and only two form of deformation of this element thirst – rigid body vertical vibration mode and second – first mode bending vibration – orthogonal to the first mode:

$$\int_V \varphi_1 dV = 0$$

In Figure 2 two first mode of seat platform is presented (obtained by the FEM program APM WinMachin).

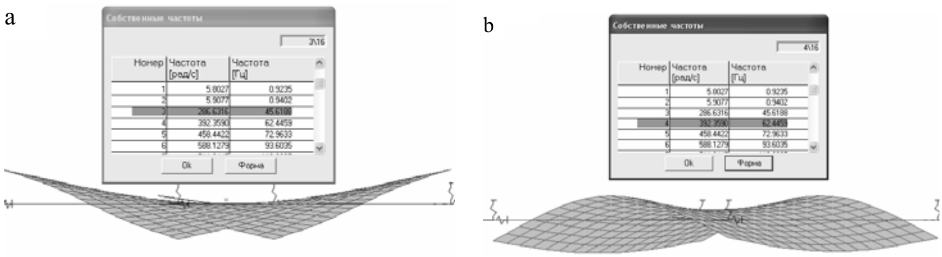

Fig. 2. First mode antisymmetrical vibration mode (a), First mode symmetrical vibration mode (b)
Rys. 2. Moda pierwsza antysymetrycznej mody wibracyjnej (a), Moda pierwsza symetrycznej mody wibracyjnej (b)

The symmetrical type of vibration (Fig. 2b) in the case of vertical vibration exists only. By small amplitude vibrations and for viscous damping k_i may be written in such a form:

$$k_i(u) = K_i u + D_i \frac{du}{dt} \tag{8}$$

If vibrations are unifrequency, we obtain

$$-[M]\omega^2 \bar{u} + [K]\bar{u} = \bar{F} \tag{9}$$

the system of ordinary linear equations (here $[M]$ – mass matrix, $[K]$ – stiffness matrix, value with upper risk are amplitudes). By solving this system we obtain the frequency dependent amplitude values.

The vibrating roller with such parameters was considered: $M_1 = 300$ kg, $M_2 = 600$ kg, $M_3 = 200$ kg, $K_1 = 1000$ kN·m^{-1}, $K_2 = 1000$ kN·m^{-1}, $K_3 = 600$ kN·m^{-1}.

OPTIMIZATION. GENETIC ALGORITHMS (GA)

A possible solution is the application of an optimization strategy to determine the specific parametric values that best fit. In general, a parameter estimation method is an iterative process that takes a known input *u(t)* of a system, evaluates it into a mathematical model with starting sets of parameter values, compares the result with the optimization function (thus obtaining a value of error between them), and then it modifies the values of the model parameters, expecting the new set to get closer to the optimum solution. 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:

$$Fcil = \max_{f_1 < \omega_1 < f_2} \left(\int_{f_1}^{f_2} \left| u_3(f) \right| P(f)\, df \right) \tag{10}$$

where: u_3 – vibration level of driver platform,
f_1, f_2 – boundaries of observed frequency domain,
P – wait function,
ω_1 – first eigen-frequency.

Parameters of optimization are: mass of absorbers $M_4 - M_7$, elastic constants $K_4 - K_7$, damping constants $C_4 - C_7$. The vibrating roller with such parameters was considered: M_1 = 300 kg, M_2 = 600 kg, M_3 = 200 kg, K_1 = 1000 kN·m^{-1}, K_2=1000 kN·m^{-1}, K_3 = 600 kN·m^{-1}. In Figure 3 the vibration levels on the driver seat platform are presented after optimization. In Figure 4 DVA configuration is shown as a three-mass system with flat elastic elements configured for the stress minimization properties.

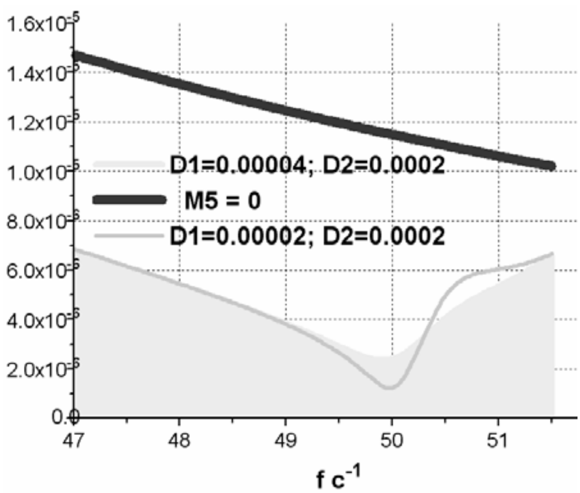

Fig. 3. Vibration levels by various damping parameters D1, D2
Rys. 3. Poziomy wibracji przy zmiennych parametrach tłumienia drgań D1, D2

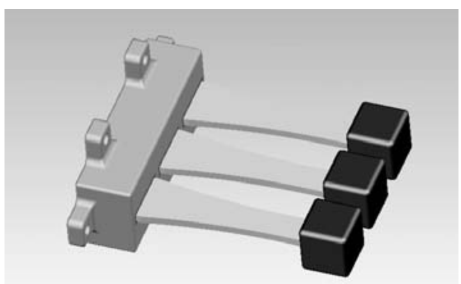

Fig. 4. Optimal DVA configuration
Rys. 4. Konfiguracja optymalna dynamicznego amortyzatora wibracyjnego

In Figure 5 the vibration levels on the driver seat platform after optimization and optimized form of absorbers flat spring's elements are presented for frequency domain $f_1 = 40\ s^{-1} - f_2 = 60\ s^{-1}$ and various eigen-frequencies $f_w = \omega_1$.

a b

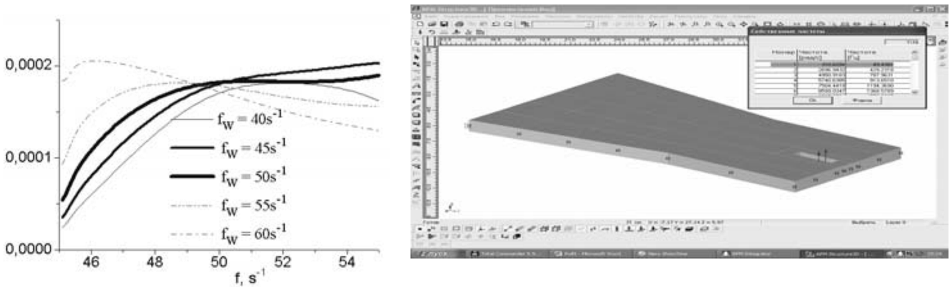

Fig. 5. Platform vibration levels by various eigen-frequencies (a), absorbers springs (b)
Rys. 5. Poziomy wibracji platformy przy różnych częstotliwościach drgań (a), sprężyny amortyzatorów (b)

CONCLUSION

In order the optimal parameters of DVA for driver seat platform are determinate the complete modeling of dynamics of vehicle should be made. Traditional design methodology, based on decoupling models of structures and machines are not effective for high frequency vibration. They do not give a possibility to determine vibration levels. Present research develops a modern prediction methodology, based on coupled theory. This allows take into consideration vehicle suspension, carrying frame and other factors. The result may be much improved by applying the genetic algorithms for optimal design searching by discrete-continuum machine modeling.

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WYKORZYSTANIE DYNAMICZNYCH AMORTYZATORÓW WIBRACYJNYCH W CELU REGULOWANIA CHARAKTERYSTYK WIBRACYJNYCH WALCÓW WIBRACYJNYCH

Streszczenie. W artykule zaproponowany został model matematyczny niezbędny dla rozliczania i optymalizacji dynamicznych amortyzatorów wibracyjnych maszyn skomplikowanych konstrukcyjnie. Za pomocą opracowanego modelu opartego na równaniach różniczkowych staje się możliwa efektywna analiza i optymalizacja różnych parametrów maszyn oraz ich właściwości dynamicznych.

Słowa kluczowe: model matematyczny, walec wibracyjny, dynamiczny amortyzator wibracyjny

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