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VIBRO- AND SOUND PROTECTION IN ROAD MACHINES BY MEANS OF VIBRATION ABSORBING

In most cases, road machines emit both acoustic and vibration energy into the fluids or structures surrounding the machinery. This is dangerous for the construction strength of the machinery, and is harmful for human health. There are two general classes of tools used to assess and optimize acoustic performance of a vehicle: test based methods, and Computer-Aided Engineering based methods. The second one is discussed in this paper.

1. Method of modeling

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 [1-5]. The discrete models are totally inadequate to accurately calculate the natural frequencies of vibration of complicated machines constructions, and therefore, for a sufficiently accurate determination of their dimensional characteristics so as to determine such frequencies. It is therefore necessary in practice to dimension these constructions through a more complex modeling [6-10]. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination. The numerical schemes (NS) is now considered for the use in such complex vibro-exitated constructions. Methods of decomposition and the NS synthesis are considered on the basis of new methods of modal synthesis [7-10]. Complex NS are of discretely-continual type that makes it possible, in the adaptive mode, to calculate tension not only in the construction elements, but also in places of concentration of tension – in the joints. One also considers numerical schemes for investi-

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

In order to determine optimal parameters of the dynamic vibration absorber (DVA), one obviously needs the complete modeling of dynamics of the machine [7-10]. The model with two degrees of freedom model is totally inadequate to calculate the vibration frequencies of the construction with adequate 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 a more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination.

2. Dynamic equation

The problem of modeling of vibration fields in complicated structures subjected to deformation and strain is considered for the purposes of dynamic absorption. The problem is solved on the basis of modified methods of modal synthesis. The methods are based on deriving the solving set of equations in a normal form with minimal application of matrix operations. The essence of the first method consists in treating the knots of junctions as compact discrete elements A_i^n for which inertial properties are taken into account without considering their strain, and treating the massive connected parts – as deformable elements A_i^c, whose inertia is taken into account on the basis of modal expansion.

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

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

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

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

Here

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

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

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

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

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

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

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

From the 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 [1-4]

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

a set of ordinary differential equations.

3. Damping in layered structures

The loss factors in layered beams (plates in cylindrical bending) are found by analytical solutions [11,12] and comparison of deformations energy:

$$\eta_\Sigma = \frac{1}{\eta_1} \frac{\frac{L^3}{3}\sum\limits_{i=1}^{n}\int\limits_{H_p^{(i)}}^{H_p^{(i+1)}} \eta_i \sigma_{xx}^{(i)} \varepsilon_{xx}^{(i)} dz + L\sum\limits_{i=1}^{N}\int\limits_{H_p^{(i)}}^{H_p^{(i+1)}} \eta_i \frac{\tau_{xz}^{(i)2}}{G_i} dz}{\frac{L^3}{3}\sum\limits_{i=1}^{N}\int\limits_{H_p^{(i)}}^{H_p^{(i+1)}} \sigma_{xx}^{(i)} \varepsilon_{xx}^{(i)} dz + L\sum\limits_{i=1}^{N}\int\limits_{H_p^{(i)}}^{H_p^{(i+1)}} \frac{\tau_{xz}^{(i)2}}{G_i} dz}, \tag{7}$$

where η_i is the loss factor of i-th layer. Let us consider three-layered beam with a thick rigid layer, a damping layer and a thin cover layer (the constrained damping layer). Such a construction presents high damping properties and high rigidity. Fig.1 presents the loss factor, and Fig.2. illustrates lamina rigidity (first eigenfrequency) of a three-layered beam with changing depth of the cover layer for various layers rigidity.

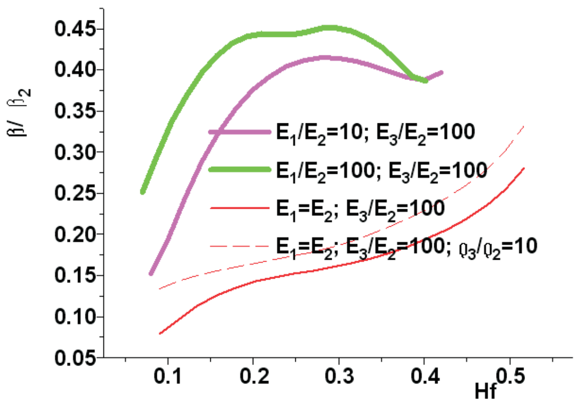

Fig. 1. Loss factor in a three-layered beam

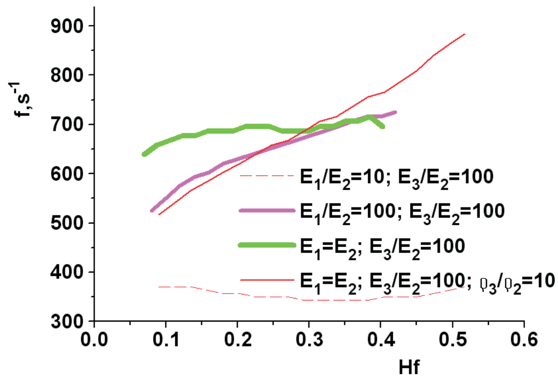

Fig. 2. Rigidity (first eigenfrequency) in a three-layered beam

In Figs.1,2, one can see that the maximum damping properties are achieved by applying specific geometrical and mechanical properties.

4. Numerical example

Fig.3. shows the scheme of a vibration roller with DVA system for drive platform. The corresponding discrete model of a half of the vehicle is presented in Fig.4.

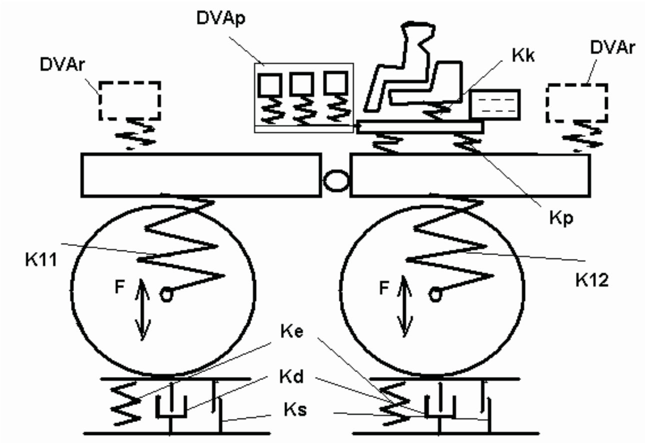

Fig. 3. Scheme of vibration roller with DVA system

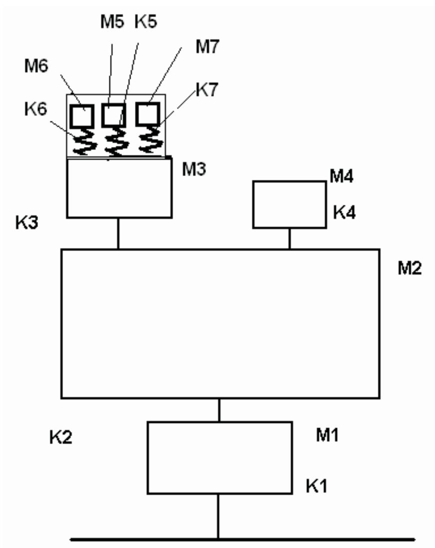

Fig. 4. Scheme of a half of the vehicle

Consider now the discrete-continuum scheme of a half of the vehicle (Fig.4). The governing equations are:

$$
\begin{aligned}
& m_1\frac{d^2u_1}{dt^2} + k_1(u_1 - u_0) + k_2(u_1 - u_2) = F(t),\\
& m_1\frac{d^2u_2}{dt^2} + k_2(u_2 - u_1) + k_3(u_2 - u_3 + w_1) + k_4(u_2 - u_4) = 0,\\
& m_1\frac{d^2u_3}{dt^2} + k_3(u_3 - w_1 - u_2) + k_5(u_3 + w_1\varphi_1(x_5, y_5) - u_5)\\
& +k_6(u_3 + w_1\varphi_1(x_6, y_6) - u_6) + k_7(u_3 + w_1\varphi_1(x_7, y_7) - u_7) = 0,\\
& m_{W_1}\frac{d^2W_i}{dt^2} - k_3(u_3 - w_1 - u_2) + k_5(u_3 + w_1\varphi_1(x_5, y_5) - u_5)\varphi_1(x_5, y_5)\\
& +k_6(u_3 + w_1\varphi_1(x_6, y_6) - u_6)\varphi_1(x_6, y_6) + k_7(u_3 + w_1\varphi_1(x_7, y_7) - u_7)\varphi_1(x_7, y_7)\\
& +\omega_1^2 m_{W_1} w_1 = 0,
\end{aligned}
\tag{8}
$$

$$
m_1\frac{d^2u_4}{dt^2} + k_4(u_4 - u_2) = 0, \quad \alpha_5\frac{d^2u_5}{dt^2} + k_5(u_5 - u_3 - w_1\varphi_1(x_5, y_5)) = 0,
$$

$$
m_1\frac{d^2u_6}{dt^2}+k_6(u_6-u_3-w_1\varphi_1(x_6, y_6)) = 0, \quad m_1\frac{d^2u_7}{dt^2}+k_7(u_7-u_3-w_1\varphi_1(x_7, y_7)) = 0.
$$

Only one continual element is considered here – the driver seat platform, and only two forms of deformation of this element; the first one – the rigid body vertical vibration mode, and the second one – the first-mode bending vibration – orthogonal to the first mode.

$$
\int_V \varphi_1 dV = 0\,.
$$

The symmetrical type of vibration (b) exists only in the case of vertical vibration.
Assuming small amplitude of vibrations and viscous damping, we can write k_i in the form

$$
k_i(u) = K_i u + C_i\frac{du}{dt}\,. \tag{9}
$$

If vibrations are at a single frequency, we obtain

$$
-[M]\omega^2\bar{u} + [K]\bar{u} = \bar{F}\,, \tag{10}
$$

the system of ordinary linear equations (here $[M]$ – mass matrix, $[K]$ – stiffness matrix, the values denoted with upper bar are amplitudes). By solving this system, we obtain the amplitude values as functions of frequency.
The considered vibrating roller had the following parameters: $M_1 = 300kg$, $M_2 = 600kg$, $M_3 = 200kg$, $K_1 = 1000kN/m$, $K_2 = 1000kN/m$, $K_3 = 600kN/m$.

5. Optimization. Genetic algorithms

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 [13]. The optimization function is

$$Fcil = \max_{f_1<\omega_1<f_2}\left(\int_{f_1}^{f_2} | u_3(f) | P(f)df\right) \tag{11}$$

u_3 – vibration level of driver platform, f_1,f_2 – boundaries of observed frequency domain, P – wait function, ω_1 . – first eigen-frequency.
The parameters of optimization are: masses of absorbers M_4-M_7, elastic constants K_4-K_7, damping constants C_4-C_7.

In Fig.5. one can see the vibration levels on the driver seat platform after optimization and for the optimized form of flat spring elements of the absorbers. These parameters are presented in the frequency range $f_1 = 40s^{-1} - f_2 = 60s^{-1}$ and for various eigenfrequencies $f_w = \omega_1$

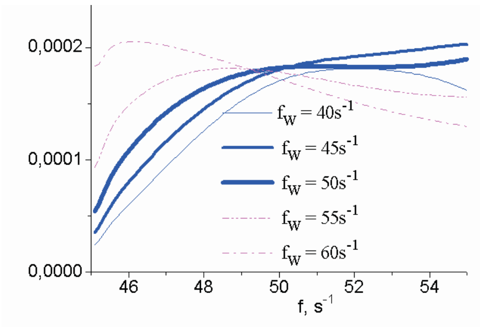

Fig. 5. Platform vibration levels by for various eigenfrequencies

6. Conclusion

In order to determine optimal parameters of the DVA for a driver seat platform, a complete modeling of dynamics of the 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 any possibility to determine the vibration levels. In the presented study, the authors developed a modern prediction methodology, based on the coupled theory. This makes it possible to take into consideration the vehicle suspension, the carrying frame and other factors. The result can be significantly improved by applying genetic algorithms, which allow for finding an optimal design by discrete-continuum machine modeling.

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Wibracyjna i akustyczna ochrona maszyn drogowych za pomocą tłumienia drgań

S t r e s z c z e n i e

Większość maszyn drogowych emituje energię akustyczną i energię wibracji do otaczającego środowiska – płynów lub struktur stałych. Jest to niebezpieczne dla wytrzymałości konstrukcji i szkodliwe dla zdrowia ludzkiego. Istnieją generalnie dwie klasy narzędzi stosowanych do oceny i optymalizacji właściwości akustycznych pojazdu: metody oparte na testach, oraz metody projektowania wspomaganego komputerowo (Computer-Aided Engineering). Metoda należąca do tego drugiego rodzaju jest omawiana w przedstawionej pracy.