666. Controllable vibro-protective system for the driver seat of a multi-axis vehicle

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Abstract. The paper considers a controlled vibration protection of the multi-axis vehicle with hydraulic shock absorbers that are based on “smart” fluid. The dependence is experimentally determined for the viscous and elastic components of the resistance force of the electrorheological fluid as a function of intensity of the electric field. Investigation of vertical oscillations of the system “workplace-driver” with a controlled vibro-protection system is provided. The dependence is established for the relative displacement of weight as function of external perturbation at different values of the control signal.

Keywords: controlled vibration protection system, multi-axis vehicle, smart fluids, vertical oscillations, simulation, experiment.

Introduction

Analysis of studies on a vibration damping device (workplace-driver) in all-terrain vehicles has been conducted on a real road and on test sites. It revealed that in the range of oscillation frequencies 1-32 Hz passive vibration protection systems are not adequate in terms of vibration acceleration. In the frequency range of 1-5 Hz elements of the vibration protection system not only failed to provide the necessary reduction of vibration, but sometimes even reinforced them. To significantly reduce the vibration level and improve the smoothness and stability of motion of the multi-supporting machine it is necessary to use active vibration damping means, which include control systems of vibration protection with a microprocessor in the control loop or using smart fluids. The paper discusses algorithms of adaptive control systems in individual systems of multi-supporting machine with feedback. Given the wide range of the exchange rate of the machine and the stimulating effects, it should be noted that the most effective vibration protection is achieved only when the control system parameters are modified in accordance with external conditions. Such a system is referred to as adaptive automatic control system.

The aim of this work is to perform theoretical and experimental studies of a controlled vibration protection of a multi-axis vehicle, in order to establish dependence of relative displacement of weight on the external perturbation at different values of the control signal.

Formulation of the problem and the mathematical description of a controlled vibration protection system of a multi-axis vehicle

Control of vertical oscillations in the system “seat-driver” is as follows (Fig. 1): perturbations that are proportional to acceleration on the driver's seat $\ddot{z}_s$ and floor of the cabin $\ddot{z}_c$ come from the feedback sensors in the control unit (CU), which generates control signals u for actuators (electro-hydraulic shock absorbers with “smart” fluid) of the seat and the suspension. Feedback sensors D_i transmit signals proportional to accelerations and displacements of regulatory systems, actuators and seat supports, which are also coming into the control unit (CU).

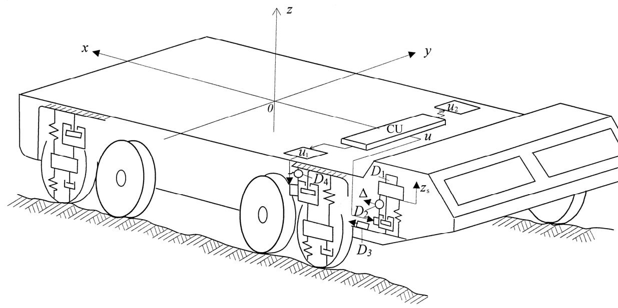

Fig. 1. Control scheme of oscillations of a multi-axis vehicle

The object of vibration protection is the mass m, moving along the axis OZ, which is acted with the kinematic displacement $q = q\ (t)$ and internal forces due to stiffness of the spring C_s and the resistance of the shock absorber K. Rigidity of the system C determines the natural frequency of cushioning of the system $\omega_0 = \sqrt{C/m}$, damping properties of the system are characterized by the damping coefficient $\eta = K/m$ and the relative damping $\beta = \eta/\omega_0$ of a vibration protection system.

The effectiveness of the vibration protection system of the multi-axis vehicle is determined by the degree of vibration protection purposes. At kinematic harmonic excitation $q(t) = q_0 \sin\omega t$ (where q_0 and ω – amplitude and frequency of displacement, respectively) the purpose of vibration protection may be to reduce the absolute acceleration of mass m and relative vibro-displacement δ. Quantitatively the degree of realization of the purposes of vibration protection are characterized by dimensionless coefficients of effectiveness of vibration isolation $K_R = \ddot{z}/(\omega^2 q)$ and dynamics $K_R = \delta/q_0$.

When the criterion of vibration protection is formulated, and also when the kinematic excitation $q\ (t)$ is specified, we can determine the optimum parameters C and K, or the variation of the restoring force F in time. For a linear system the restoring force is equivalent to relative displacement of F~δ, hence, $F = K\dot{\delta} + C\delta$, where $\delta = z - q$. This implies that a given restoring force F can be obtained only by varying the parameters C and K. Algorithms to determine the force F depends on the adopted performance criteria. Under harmonic kinematic excitation it is proposed to minimize the performance criterion:

$$I_1 = \delta_m + \rho_1 F_m \tag{1}$$

where δ_m – peak (maximum) value of the relative masses of the insulated vibration protection system; ρ_1 – weighting factor reflecting the degree of preference of a small vibro-peak values (for small ρ_1) or the smallest peak value of acceleration (at high ρ_1); $F_m = m\,\ddot{z}_m$ – peak force transmitted by the vibration protection system as a result of the maximum acceleration of mass $\ddot{z}_m$.

If the kinematic excitation $q(t)$ is random and known for its statistical characteristics, it minimizes the criterion:

$$I_2 = M\left[\delta^2\right] + \rho_2 M\left[F^2\right] \tag{2}$$

where M – the mathematical expectation; ρ_2 – weighting factor.

I_2 criterion is widely used in the design of optimal dynamic systems. Based on the theory of optimal control (for example, using dynamic programming) and the quadratic performance criteria one can receive recommendations on choosing the control law for the creation of the algorithm of changing the restoring force F. The optimal control consists of introducing a linear combination of all the state variables of vibration protection in a feedback loop of vibration protection control system of vehicle. Assuming that the kinematic excitation has the components q and $\dot{q}$, it is required to minimize the criterion:

$$I_3 = \rho_3 M\left[\delta^2\right] + \rho_4 M\left[\dot{z}^2\right]. \qquad (3)$$

From which the control law is as follows:

$$F = K\dot{z} + C\delta. \qquad (4)$$

Values K and C depend on constant multiplying factors ρ_3 and ρ_4 from expression (3).

Control law with the implementation of the restoring force $F = K\dot{z}$ is theoretically determined by means of passive vibration protection of the type "inertial damper". Active vibration protection from electro-hydraulic control can be used in vibration isolation devices that allow the modulation generated by the restoring force *F*, for example, actuators with electric, electro or electro-dynamic principle of control, as well as elements with electro- or magneto-rheological energy converter. Fig. 2 shows an active vibration protection "seat - driver" with a device, where the restoring force *F* is regulated by a control system based on information about the object.

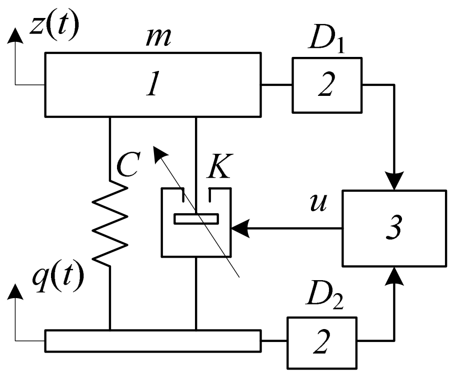

Fig. 2. A scheme of active vibration protection with the hydraulic shock absorber: 1 – vibration protection object with the mass *m*; 2 – sensor; 3 – control unit

To allow energy dissipation due to the work of restoring force *F*, the following condition should be satisfied:

$$F\dot{\delta} \geq 0. \qquad (5)$$

The basic information signals for the control unit are the feedback signals from vibration acceleration and the relative displacement of the insulated mass from vibro-sensors D_1 and D_2. Converting these signals, the control unit generates a control signal, which affects the hydraulic shock absorber and controls the flow of a nearly incompressible "smart" fluid at the entrance of the cylinder and the exit from it. A rod of the hydraulic cylinder is connected to the insulated mass *m* directly (Fig. 2). Thus, by kinematic excitation, which is transferred to the base of the piston mechanism its vertical displacement occurs. Hydraulic resistance of the fluid in the shock absorber is regulated by the resulting control signal *u*. Under the influence of the pressure drop the rod of the actuator will move the mass at a distance proportional to the input control signal. This way, the vibration protection occurs at displacement.

Let the flow of the working fluid in the channel of the shock absorber to be characterized by an average consumption of $V_{av} = S_p\dot{\delta}$, where S_p – effective area of the piston of shock absorber. Average consumption of the fluid in the channel after Laplace transform is defined as:

$$G = S_p p\Delta, \qquad (6)$$

where Δ – relative vibro-displacement δ determined by Laplace; p – Laplace operator.

Through the appropriate conversion of feedback signals it can be arranged that the flow rate of the fluid in the channel of the shock absorber becomes proportional to the linear combination of vibro-acceleration $\ddot{z}$, relative vibration speed $\dot{\delta}$ and relative vibro-displacement δ. After Laplace transform of this sum of components we obtain:

$$G=-(k_{\ddot{z}}\, p^2 z + k_{\dot{\delta}}\, p\Delta + k_{\delta}\Delta), \tag{7}$$

where $k_{\ddot{z}}$, $k_{\dot{\delta}}$ and k_{δ} – amplifying coefficients in the feedback loop of the control unit accordingly to the vibro-acceleration, relative vibration speed and relative displacement of the isolated mass.

By uniting (6) and (7), we will get:

$$k_{\ddot{z}}\, p^2 z + \left(k_{\dot{\delta}} + S_{\ddot{\imath}}\right) p\Delta + k_{\delta}\Delta = 0. \tag{8}$$

From this equation we will find expressions for transfer functions of absolute $R_x(s)=z(p)/q(p)$ and relative $W_\delta(p)=\Delta(p)/q(p)$ vibro-displacement:

$$W_z(p)=(\left(k_{\dot{\delta}} + S_{\ddot{\imath}}\right)p + k\,\delta)/(k_{\ddot{z}}\, p^2 + \left(k_{\dot{\delta}} + S_{\ddot{\imath}}\right)p + k_{\delta}), \tag{9}$$

$$W_\delta(p)=-\, k_{\ddot{z}}\, p^2 /(k_{\ddot{z}}\, p^2 + \left(k_{\dot{\delta}} + S_{\ddot{\imath}}\right)p + k_{\delta}). \tag{10}$$

Control vibro-protection system with electro-hydraulic shock absorber is able to provide dynamic properties superior than the working dynamic properties in comparison with passive vibration protection. However, the controlled vibro-protection system uses an external electrical energy and is more expensive, more complicated and less reliable. Therefore, the active vibro protection system is reasonable only when the improvement of dynamic characteristics justifies the cost and complexity of the design.

Shock absorber resistance force is generated by a power amplifier of control signals *u*. For the amplifier work it is necessary to use a small source of energy, since the main part of the efforts of control shock absorber is created by a passive way of vibration protection.

Restoring force *F*, adjustable by a control unit should change the sign of vibration speed $\dot{\delta}$ in accordance with the compelling influence of *q* (*t*) and distribute the energy fluctuations. When $\dot{z}\delta = 0$, the following two cases are possible: 1) when velocity $\dot{z}=0$, then it is necessary that $F=0$, 2) $\dot{z} \neq 0$ if $\dot{\delta}=0$, $F = K\,\dot{z}$. Restoring force *F* can be so large that the system will be "locked", and $\dot{\delta} = 0$ during finite period of time. In this case $\ddot{z} = \ddot{q}$ and restoring force of elastic element $F=C\delta$, where $C=C_s+C_f$, C_s – spring stiffness, C_f – fluid elasticity. Then the restoring force of the shock absorber is:

$$F = -m\,\ddot{q} - C\delta. \tag{11}$$

In this case the controlled vibration protective system will be locked, when $\dot{\delta}=0$ and the value of the required force $F=K\,\ddot{z}$ will be greater than the value (11). The aforementioned conditions can be written as:

$$F = \begin{cases} K\dot{z}, \quad if \quad \dot{z}\,\dot{\delta} > 0; \\ 0, \quad if \quad \dot{z}\,\dot{\delta} = 0, \dot{z} = 0, \dot{\delta} \neq 0; \\ 0, \quad if \quad \dot{z}\,\dot{\delta} < 0; \\ -m\ddot{q} - C\delta, \quad if \quad \dot{z}\,\dot{\delta} = 0, \dot{z} \neq 0, \dot{\delta} = 0. \end{cases} \tag{12}$$

Thus, the controlled vibro-protection system according to conditions (12) will be moved from one possible state to another (for example, from a condition of oscillation damping without an electric field into a condition of active oscillation damping under influence of operating electric field and, respectively, at the predetermined value of resistance factor of a fluid), depending on the type of kinematic excitation *q* (*t*) and the nature of forced oscillations.

Theoretical and experimental study. Analysis and discussion

On the basis of experimental data on vibration protection of systems of precision equipment for multi-axis vehicles in conditions of single harmonic exciting vertical oscillations we have obtained numerical dependences of viscous and elastic components of the resistance of electrorheological fluid (ERF) in the hydraulic shock absorber depending on the electric field strength (Fig. 3).

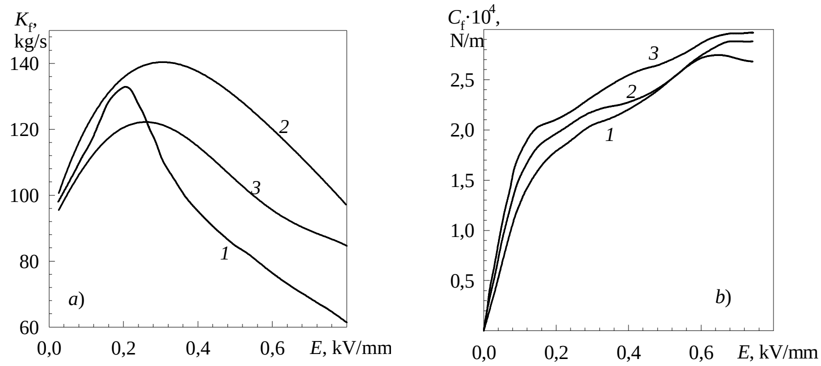

Fig. 3. Dependences of viscous K_f (*a*) and elastic C_f (*b*) components of resistance force of ERF as a function of intensity of the electric field strength. 1 – $q_m\omega$ = 0.3; 2 – 1; 3 – 2 m/s

Numerical calculations performed for vertical oscillations on the example of electrorheological vibration protection system (mass m=550 kg, C_s=18.4·10^4 N/m, E=0.3 kV/mm, z_m=10^{-6} m, δ=10^{-7} m) demonstrated that characteristics of ERF are crucial for the relative maximum displacement z_m/q_m (Fig. 4). Subsequent experiments provided the efficiency of hydraulic shock absorbers based on "smart" fluid systems of vibration protection of precision equipment for the vehicles at $T\cdot\omega_0$>>1 and T_0/T<<1.

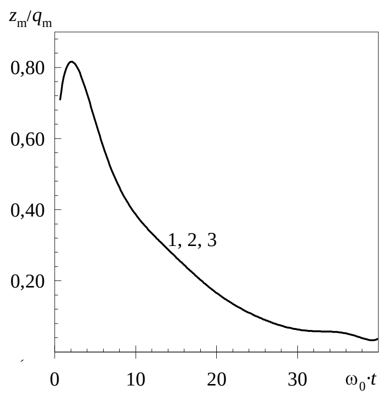

Fig. 4. Dependence of relative displacement of weight z_m/q_m on the parameter of external perturbation $\omega_0 t$ at E = 0.3 kV/mm for q_m: 1 – 100·10^{-6}, 2 – 400·10^{-6}, 3 – 300·10^{-6} m

Controlled vibration protection systems for multi-axis vehicles with hydraulic shock absorbers based on the "smart" fluid have such advantages as speed, ability to program the control of the process of energy dissipation, simple design, the absence of intermediate mechanical assemblies. They may find wide applications for the effective damping of oscillations in dynamic conditions when vehicle is moving along the road profile.

Conclusions

Experimental dependences were obtained for the viscous and elastic component of the resistance force of ERF as function of intensity of the electric field. The results of research on vertical oscillations of the system “workplace-driver” with the controlled vibration protection system were demonstrated. The dependence of relative displacement of weight on the external perturbation at different values of the control signal was established. Minimization criteria for the design of optimally controlled vibro-protection systems for multi-axis vehicles have been proposed.

“Smart” working fluids used in hydraulic shock absorbers of the controlled vibration protection system for vehicle must have a combination of special qualities: a considerable increase in viscosity under influence of external electric or magnetic field, high aggregate stability and ability to maintain stable performance characteristics at temperatures corresponding to an intense energy dissipation in damping devices. However, it is worth to pay serious attention on the constructive development of damping actuators (hydraulic shock absorbers), which will also enhance the efficiency of use of electro- and magneto-rheological fluids.

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