The active system of vibration isolation with electrodynamic actuator

Yu A Burian, D V Sitnikov
Omsk State Technical University, 11 Mira ave., Omsk, 644050, Russia

Abstract. The work is devoted to the study of the possibility of developing an active vibration isolation system in which in order to compensate for the dynamic forces on the hull arising due to the oscillations of an elastically suspended vibroactive mass, the effect of inertial forces in antiphase is used. The vibroactive force compensator is an electrodynamic actuator. It is mounted on the hull close to vibroactive mass supports and produce an inertial force by controlling oscillations of the moving mass, which compensates the force on the hull from the vibroactive mass.

1. Introduction

Passive vibration isolation systems are effective at frequencies above the resonant frequency of an oscillating system. At the near-resonant and pre-resonant frequencies the passive vibration isolation systems practically do not reduce the vibro-active forces, which are transmitted to the hull from the vibroactive device. The problem of creating vibration isolation systems for low frequencies is currently relevant.

Active power devices (actuators) are widely used in vibration protection systems. These devices use hydraulic, electrodynamic, piezoelectric and other types of actuators to suppress oscillations in the low-frequency range (5-20 Hz). The problem of effective vibration isolation at low frequencies (2–10 Hz) is currently unsolved. Its solution is required in some industries, for example, in shipbuilding. Active systems are not used usually in vibration isolation problems.

Active vibration protection and vibration isolation systems were considered in papers [1–5]. The capabilities of active vibration protection systems with various actuators which are located between the vibroactive devices and the base are presented in paper [1]. The control system of these systems used signals from acceleration and (or) forces sensors.

The using of actuators increase the efficiency of vibration isolation systems in a fairly narrow band beyond the system resonance and can to reduce the resonant frequency value [3]. The system can have the tuning frequency in the superresonance range with a minimum vibration isolation value too [5].

The oscillation amplitude decrease is achieved by increasing the force on the base from the actuator. This contradicts the main goal of vibration isolation. So the mounting of actuator between the vibroactive device and the base in vibration isolation systems is fundamentally impossible.

The actuator mounted on the base and creating inertial dynamic force antiphase to the vibroactive force is possible to use to reduce the total vibration force acting on the base in the low-frequency range [6]. The principle of dynamic inertial compensation of the vibration force acting on the base is that compensating force occurs during reciprocating movement of the movable mass of the compensator in antiphase to the movement of the mass of the vibroactive device mounted on the elastic elements of the passive part of the vibration isolation system.

Journal of Physics: Conference Series **1441** (2020) 012089 doi:10.1088/1742-6596/1441/1/012089

2. Theory

The diagram of an active vibration isolation system with an electrodynamic compensator (EDC), which is controlled by a signal from force sensors, shows at fig.1. The mass of the vibroactive device is mounted on the spring-damping element 1 of the passive part of the vibration isolation system, installed on the force sensor 2 and the base 5. The moving mass of the electrodynamic compensator includes a permanent magnet 8 and a magnetic circuit 7 mounted on a fixed guide. The springs 3 are used to centre the movable part relative to coil 6, which is fixed in the EDC hull, which also performs the function of the fixed magnetic circuit.

The force sensors measure the action of the passive system R_b and the compensating force from the EDC R_c. The difference of forces ΔR is boosted by the amplifier 4 and applied to the coil 6. So the force, which is proportional to the current in the coil, applies to the magnet.

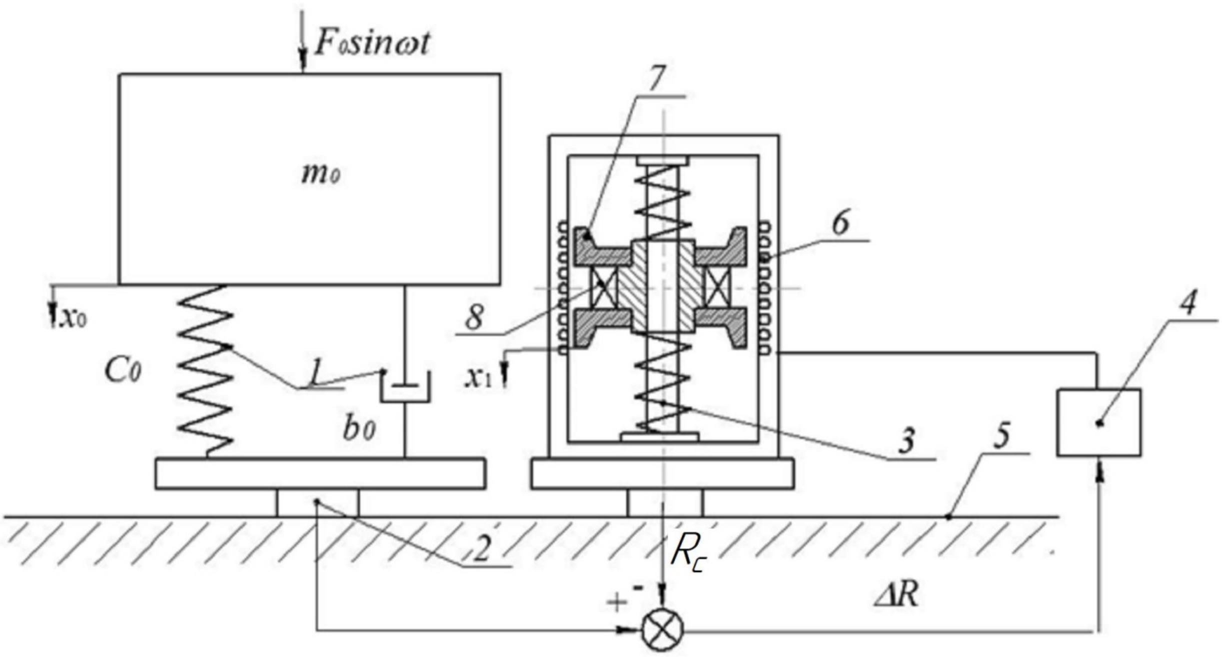


Figure 1. Active vibration isolation system with an electrodynamic compensator:

1 – spring-damping element of the passive part; 2 – force sensors; 3 – spring; 4 – amplifier gain K; 5 – base; 6 – coil; 7 – movable magnetic circuit; 8 – permanent magnet; $F_0 \sin \omega t$ – vibroactive force; R_b – base reaction from the passive system; R_C – compensating force from EDC

The following assumptions were used while a mathematical model building:

– the system oscillates along the vertical axis only and there are no angular oscillations;

– the mass of the vibroactive device m_0 movement of and the mass of the moving parts of the EDC m_1 is considered relative to the equilibrium positions;

– there is no interaction between the force sensors in the considered frequency range.

The dynamics of the EDC in the servo mode is described by the following differential equations [6]:

$$\left.\begin{aligned} & m_1\ddot{x}_1 + b_1\dot{x}_1 + c_1 x_1 = Bli \\ & L\frac{di}{dt} + Ri + Bl\dot{x}_1 = u \\ & u = \left(R_\text{b} - R_\text{c}\right)\cdot K \end{aligned}\right\}, \qquad (1)$$

where u is the control voltage on the coil winding; i is current strength; Bli is electrodynamic force; L, R are inductance and resistance of the coil; B is magnetic induction; l is the total length of the conductor; b_1 is the coefficient of viscous friction; K is gain;
The block diagram of the automatic control system according to Fig. 1 is shown in Fig. 2. It is taken into account that $\Delta R = R_b – R_c$, where R_b is the force acting on the base through the passive system, $R_c = m_1\ddot{x}_1$ is the inertial EDC effect.

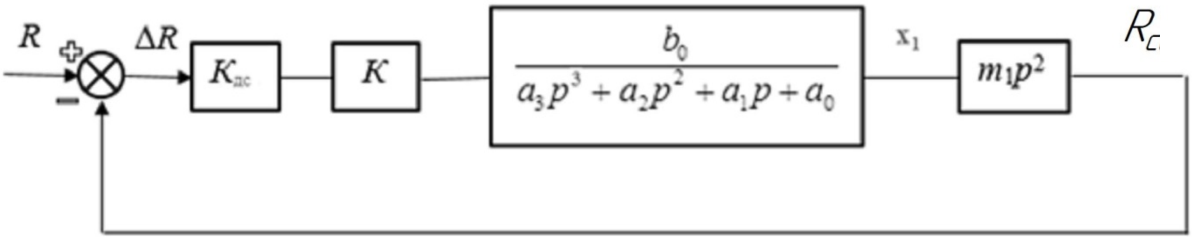


Figure 2. Block diagram of the automatic control system: $a_0 = Rc_1$, $b_0 = Bl$ - for EDC without servosystem

The passive system motion’s equations:

$$m_0\ddot{x}_0 + b_0\dot{x}_0 + c_0x_0 = F_0 \sin \omega t. \tag{2}$$

The force R_b transmitted to the base:

$$R_b = c_0x_0 + b_0\dot{x}_0. \tag{3}$$

The stability Hurwitz’s criterion for the automatic control system in Fig. 2 follows to inequality:

$$\left(Lb_1 + Rm_1 + KBlm_1\right)\left(Lc_1 + b_1R + B^2l^2\right) > Lm_1 \cdot Rc_1. \tag{4}$$

So the inertial compensation system is stable for any positive values of K.
The control system analysis (Fig. 2) showed the capability of the high compensation efficiency of vibroactive forces in the frequency range of 1–20 Hz.
For the values of m_0 = 100 kg; m_1= 1 kg; c_0 = 3,56·10^4 N/m; c_1 = 300 N/m, Bl= 10 T·m, L = 5·10^{-3} H, b_0 = 300 N/m, R = 10 Ohm, b_1 = 2.5 N·s/m the amplitude-frequency characteristics for $K_p(\omega) = \frac{|\Delta R(i\omega)|}{|F(i\omega)|}$ at different values of K are built in paper [12]. For example, for K = 10 for a frequency of 3 Hz, the K_p is 15 dB. However, this work did not take into account the amplitude limits of the mass m_1 movement, which should significantly reduce the effectiveness of vibration isolation at low frequencies.
The constructive constraints of the displacement x_1 amplitude limits and the maximum drive force limits will always be even if the electrodynamic actuator perfectly performs the dynamic signal $K \cdot \Delta R(\omega)$.
The maximum value of $\bar{x}_1$ was taken 0.02 m, and the drive force was taken not more than 50 N. the principle of dynamic compensation is disturbed if the mass m_1 will be on the stops during a part of the oscillation period while actuator’s operation. The inertial load is not created in this case and this shock mode can damage the actuator structure.
So it is necessary to include a block into the actuator control system, the block’s operation would exclude such a mode.
The actuator’s allowable voltage u_{perm} required to displacement x_1 with mass m_1, spring’s stiffness c_1 and damping factor b_1 is determined by the expression

$$u_{\mathrm{perm}} = \frac{\overline{x}_1}{Bl} \cdot \sqrt{\left(Lb_1\omega^2 - Rc_1 + Rm_1\omega^2\right)^2 + \omega^2 \cdot \left((Bl)^2 - Lm_1\omega^2 + Lc_1 + Rb_1\right)^2} \qquad (5)$$

Taking for the model example $\overline{x}_1 = 2 \cdot 10^{-2}$ m, the allowable values of u_{perm} are shown in Fig. 3

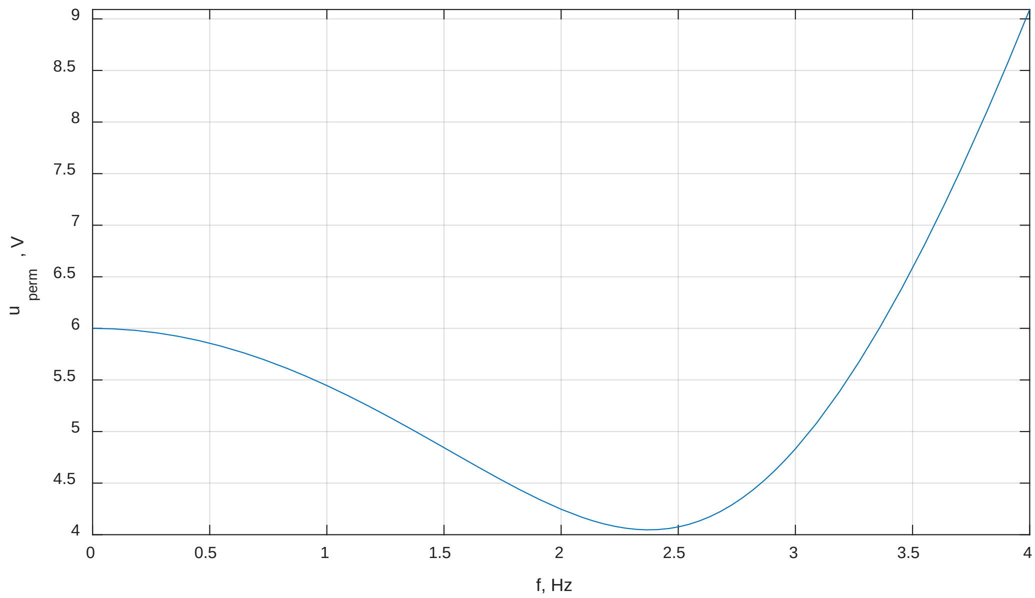
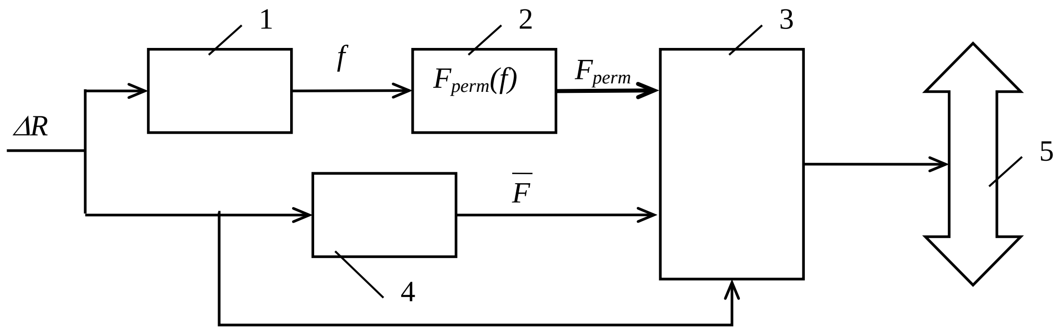


Figure 3. The dependence of the actuator's allowable voltage on the frequency for the displacement amplitude of 2 cm

The adaptation block's diagram is shown in fig. 4. The adaptation block in the automatic mode limits the force in the actuator and, accordingly, the displacement x_1 for the harmonic mode.



1 – frequency meter f; 2 –$F_{\mathrm{perm}}(f)$ meter; 3 – variable gain;
4 – amplitude meter; 5 – actuator

Figure 4. Force limiter block diagram

The including logic block, which disables the adaptation block, if $\frac{\overline{F}_{\mathrm{perm}}}{\overline{F}} \geq 1$, in the block diagram (Fig. 4) is necessary.

The model in the Matlab/Simulink is shown in fig. 5.

The amplitude measurement in the adaptation block is implemented by passing the signal through a half-wave rectifier and a low-pass filter, and the logic block is implemented by the switch.

To build the frequency characteristic $K_{\text{p}}(\omega) = \dfrac{|\Delta R(i\omega)|}{|F(i\omega)|}$, a constant amplitude signal with slowly varying frequency (0.01 rad/(s·s)) was set to the input of the oscillating system.

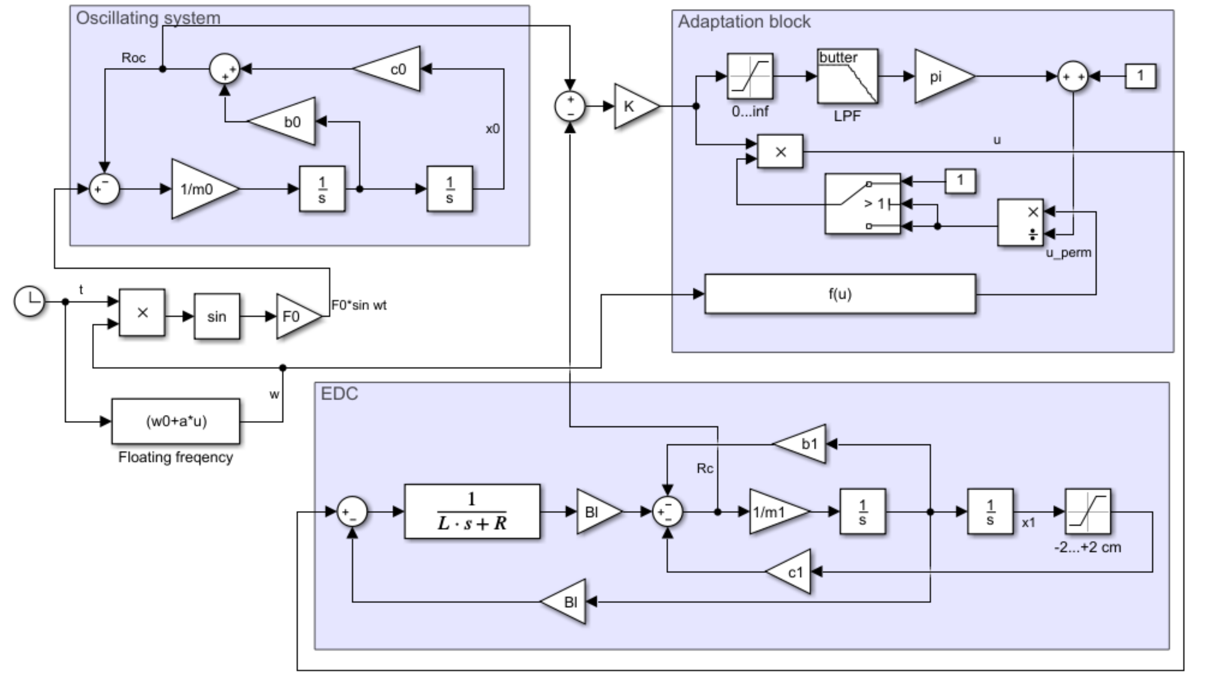


Figure 5. Model in Matlab/Simulink

3. Results

The frequency characteristics of $R_b(f)$ and $\Delta R(f)$ with m_0 = 100 kg; b_0 = 300 Ns/m; c_0 = 3,56·10^4 N/m; m_1 = 1 kg; b_1 = 2.5 Ns/m, F_0 = 2 N, K = 5 for different values of c_1 (50 and 300 N/m) are shown in fig. 6-7.

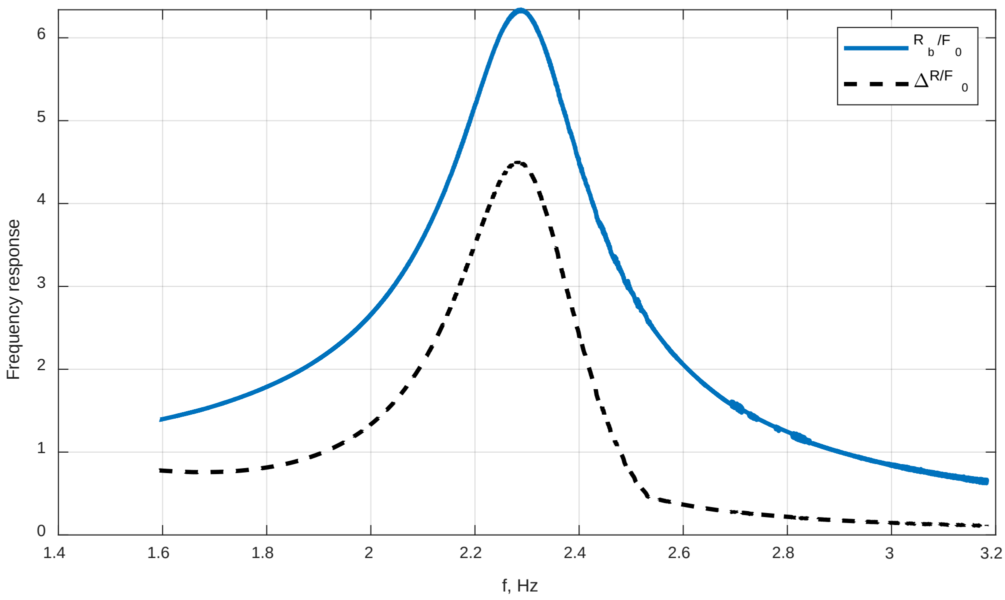


Figure 6. Frequency response of R_b and ΔR forces at c_1 = 50 N/m

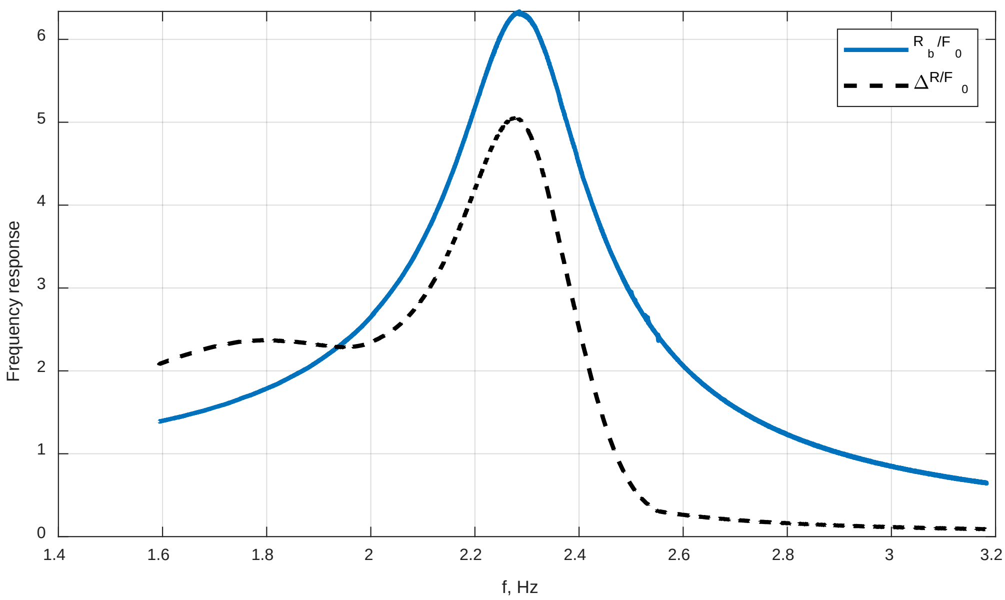


Figure 7. Frequency characteristics of R_b and ΔR forces at c_1 = 300 N/m

4. Summary and conclusions

The adaptation's algorithm that automatically provides structurally acceptable displacement of the moving mass of the EDC at an arbitrary value of the control signal is introduced in the mathematical model of the active vibration isolation system.

The vibration isolation system's mathematical model with active inertial compensation of vibroactive forces using the EDC installed on the base shows was researched. It is established that the decrease of the vibration isolation effectiveness at frequencies below 2.6 Hz occurs due to limitations on the value of m_1 and x_1.

The reduction of the forces on the base from the spring-suspended vibroactive unit is 20–40 dB at frequencies higher than 2.6 Hz.

Analysis of the stiffness influence of the moving EDC mass suspension shows that in order to increase the vibration isolation effectiveness, the stiffness should be low and provide only the moving mass centering.

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