The inertial compensation of the vibroactive force in the oscillating system

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

Abstract. Actuators used in active vibration isolation systems of various types are either installed between the oscillating mass and the base, or next to the elastic-dissipative elements of a passive system. The actuator is usually controlled according to information from force-measuring devices. This significantly complicates the implementation of such systems. An active system is considered in which an electrodynamic drive (actuator) is used to compensate for vibroactive forces transmitted to the base. An electrodynamic drive is mounted on the base and controlled by accelerometers mounted on the passive system moving masses and the actuator. Straight-line controlled vibrations of the actuator moving mass provide inertial compensation of vibroactive forces on the base.

1. Introduction

Active systems of vibration protection and vibration isolation have recently begun to be used in various branches of mechanical engineering. Passive systems with elastic-dissipative elements are not effective in the range of low frequencies 1 - 20 Hz.

The results of research and practice of using active power devices (actuators) of hydraulic, electrodynamic, piezoelectric and other operating principles for the purpose of vibration protection are known. In the review work [1], the history of the development of the field of application and the basic principles of constructing vibration protection systems are considered. In [2], the main schemes and methods of research and calculation of active vibration isolation systems are given for both regular and random influences from vibroactive forces. In [3], [4] various approaches to the construction of active vibration isolation systems are considered. It is assumed that the actuator is mounted between the passive system vibrating mass and the body. This makes it possible to effectively solve the problems of vibration protection in the low frequency range. Unfortunately, this way is unacceptable for vibration isolation, because a decrease in the vibration amplitude of the protected mass leads to an increase in the force on the base from the actuator.

Currently, there is an extensive scientific and technical literature on active vibration isolation and vibration protection systems. For example, in [5] an analysis of the vibration protection system operation of an object using an active hydraulic support was carried out. It is shown in [6] that it is advisable to use piezoelectric accelerometers with a high transmission coefficient for electronic control circuits of active vibration protection systems. In [7], a description and results of theoretical and experimental studies of an active damper for active vibration protection based on magnetorheological elastomers are given. For example, in [8] the original concept of an intelligent system of active vibration protection and high-precision aiming of the “Millimetron” observatory telescope is presented.

The given short list of works shows a wide area of active vibration protection systems application and their effectiveness at low frequencies.

Unfortunately, in the scientific and technical literature, there are practically no works devoted to a promising direction - inertial compensation of vibroactive forces, which will solve the problem of vibration isolation in the low frequency range.

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The solution to this problem is especially important, for example, for shipbuilding, because the high power-to-weight ratio of modern ships generates vibration and noise. Being transmitted into the aquatic environment, vibrations and noise disrupt the secrecy of the marine objects action. The hydroacoustic field (acoustic portrait) is an important informative feature that makes it possible to detect and classify marine objects at distances of hundreds of kilometers [9], [10]. The main frequency range for early detection is the range of 5 - 40 Hz, so the development of effective low-frequency vibration isolation systems is an urgent task.

In [11], [12], [13], the issues of dynamic compensation of vibroactive forces by an electrodynamic actuator controlled by signals from force measuring devices are considered. The effectiveness of this approach to the construction of active vibration isolation systems is shown.

This paper considers the construction of an active system with inertial compensation of vibroactive forces in which only acceleration sensors (accelerometers) are used to control the actuator.

2. Theory

Preliminary studies have shown that using information only about acceleration leads to an unstable system. It is necessary to form an estimate of the base response from a passive system. One of the options for solving this problem for the case of stationary single-frequency exposure from vibroactive units is considered in this work.

A basic diagram of a fairly simple active system with vibroactive forces inertial compensation and acceleration control is shown in Fig. 1.

![Figure 1. Basic diagram:](image)

- 1 – passive vibration isolation system; 2 – accelerometers; 3 – frequency meter; 4 – multiplication blocks; 5 – magnetic circuit; 6 – permanent magnet; 7 – control winding; 8 – spring; $\tilde{R}$ – base reaction estimate; $F_0 \sin \omega t$ – vibroactive force
When developing the basic diagram, it was taken into account that the signal from the accelerometer in the oscillatory system with mass \( m_0 \) has the opposite sign and practically coincides in frequency and phase with the base reaction \( R(t) \). It is assumed that the values \( c_0 \) and \( b_0 \) of the passive system are known with sufficient accuracy. If the current frequency \( \omega \) is measured by block 1, then an approximate base reaction estimate can be done by the expression

\[
R(t) = x_0(t) = x_{\infty} \frac{c_0}{\omega^2} + x_0(t) b_0.
\]  

(1)

The displacement \( x_0 \) in the steady regime:

\[
x_0(t) = x_\infty \sin(\omega t + \epsilon),
\]

where \( \epsilon \) is the phase shift between the disturbing force \( F(t) = F_0 \sin \omega t \) and \( x_0(t) \). Acceleration measured by an accelerometer with a single frequency impact will be determined as the second derivative of \( x_0(t) \)

\[
\ddot{x}_0(t) = -x_\infty \omega^2 \sin(\omega t + \epsilon).
\]

Calculation of the base reaction estimate according to the proposed algorithm is given by:

\[
\tilde{R}(t) = x_{\infty} (c_0 + b_0 \omega \sin(\omega t + \epsilon)).
\]  

(2)

The base reaction \( R(t) \) measured, for example, by a force-measuring device, is determined by the dependence

\[
R(t) = x_0(t)c_0 + \dot{x}_0(t)b_0 = \ddot{x}_0 c_0 \sin(\omega t + \epsilon) + \ddot{x}_0 b_0 \omega \cos(\omega t + \epsilon).
\]  

(3)

Expression (3) can be represented as:

\[
R(t) = \tilde{R} \sin(\omega t + \epsilon + \phi),
\]

where \( \tilde{R} = \ddot{x}_0 \sqrt{c_0^2 + b_0^2 \omega^2}, \phi = \frac{b_0 \omega}{c_0} \).

Comparing the amplitudes \( \tilde{R}(t) \) and \( R(t) \) it can be noticed that \( \tilde{R}(t) > R(t) \). Also, a phase mismatch at a small angle \( \phi \) is seen.

Fig. 1. also shows a structural diagram of an electrodynamic actuator with a movable niobium magnet (6) and a magnetic circuit (5), forming an inertial mass \( m_1 \). The mass \( m_1 \) can move translationally along the bearing slide. The control winding (7) is located on the actuator body.

When compiling a mathematical model, the following assumptions were made:

– single-frequency and unidirectional oscillations occur in the system;
– the motion of the masses \( m_0 \) and \( m_1 \) are considered relative to the equilibrium positions;
– the frequency meter is a non-inertial link.

Taking into account the assumptions made, the dynamics of the considered electromechanical system is described by the following equations:

\[
\begin{align*}
\sum_{i=0}^{m_0} m_i \ddot{x}_i + L \frac{di}{dt} + R_{Ohm} i + B l \dot{x}_i = u \\
\sum_{i=0}^{m_1} m_i \ddot{x}_i + L \frac{di}{dt} + R_{Ohm} i + B l \dot{x}_i = u \\
\end{align*}
\]

(4)

where

\[
\begin{align*}
\begin{array}{c}
u \text{ is control voltage at the coil winding;} \\
i \text{ is amperage;} \\
B l i \text{ is electrodynamic force;} \\
L, R_{Ohm} \text{ is inductance and active resistance of the coil;} \\
B \text{ is magnetic induction;} \\
l \text{ is conductor total length;} \\
K_0 \text{ is gain.}
\end{array}
\end{align*}
\]

The equations system (1) was analyzed in the Matlab/Simulink software package. The Matlab / Simulink model is shown in fig. 2., the vibration isolation coefficient \( K_{vi} = \frac{\Delta R(i\omega)}{\Delta F(i\omega)} \) frequency-response characteristics are constructed for the following variant of the system parameters: the
frequency meter is a non-inertial link; the values of \( m_0, c_0 \) and \( b_0 \) are known; \( K_0 = 100; \Delta R = \bar{R} - m_1 \ddot{x}_1 \).

**Figure 2.** Vibration isolation system model

The study of the control system stability for an electrodynamic drive in [14] showed that such a system is stable at any positive values of \( K_0 \). An increase in \( K_0 \) leads to the vibration isolation efficiency and the automatic control system oscillation increase.

Vibration isolation coefficient frequency-response characteristics for values \( m_0 = 100 \) kg; \( m_1 = 1 \) kg; \( c_0 = 3.56 \times 10^4 \) N/m; \( c_1 = 157.75 \) N/m; \( Bl = 10 \) T·m; \( L = 5 \times 10^{-3} \) H; \( R_{Ohm} = 10 \) Ohm; \( b_0 = 400 \) Ns/m; \( b_1 = 5 \) Ns/m; \( F_0 = 2 \) N are shown in Fig. 3.

Fig. 4 shows the values of the calculated estimate \( \bar{R}(t) \) and the base response \( R(t) \) for the external disturbance frequency \( f = 3 \) Hz. This frequency is equal to the natural frequency of the passive system \( f_0 = 3 \) Hz.

\( R(t) \) and \( \Delta R(t) \) graphs for the same values of \( f \) and \( f_0 \) are given in fig. 5. It corresponds to a point of 3 Hz on the frequency-response characteristic (Fig. 3).

Fig. 6 shows the inertial mass \( m_1 \) oscillations development in an electrodynamic drive with a gain \( K_0 = 100 \). Fig. 7 shows the inertial mass movement amplitude-frequency response characteristic.
Figure 3. $K_{\nu}(f)$ frequency-response characteristics

Figure 4. $R(t), \bar{R}(t)$ graphs: 1 - $\bar{R}$; 2 - $R(t)$; $f_0 = 3$ Hz; $f = 3$ Hz
Figure 5. \( R(t), \Delta R(t) \) graphs: 1 - \( R(t) \); 2 - \( \Delta R(t) \); \( f_0 = 3 \) Hz

Figure 6. Actuator inertial mass \( m_1 \) movement \( x_1(t), f = 3 \) Hz
3. Findings
The study of a sufficiently simple active vibration isolation system with inertial compensation of vibroactive forces showed that acceleration control by an electrodynamic drive is promising when evaluating the base response from the measured values of the passive system mass vibration frequency.

Studies have shown that the system has sufficient robustness, so if an error in setting $c_0$ and $b_0$ is 10%, the effectiveness of vibration isolation ($K_{vi}$ value) decreases by 1%.

The use of only accelerometers for control will ensure the introduction of such devices into the practice of vibration isolation. It does not require structural changes in the device of elastic-dissipative suspension of vibroactive units.

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