Electrodynamic long-stroke drive as an inertial compensator of low-frequency vibroactive forces

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

Abstract. The article discusses various possibilities of using an electrodynamic drive with an increased displacement of a movable permanent magnet as a low-frequency inertial compensator of vibroactive forces. The main attention is paid to the analysis of an active dynamic vibration damper (ADVD) when installing an electrodynamic compensator (EDC) on the base near vibration supports and when connecting the EDC moving mass with a vibroactive mass using a spring. It is shown that the considered ADVD is effective as a vibration isolator in the low-frequency range.

1. Introduction
The active vibration isolation system includes a passive system (the vibroactive mechanism mass located on the elastic-dissipative supports) and a power drive (actuator) with a control system [1 - 5].

To implement the dynamic inertial compensation principle [6] of vibration forces transmitted to the base through elastic-dissipative supports, it is advisable to use an electrodynamic drive. It can provide a sufficiently large displacement of the moving mass (7). As it is shown by the analysis of scientific and technical literature, for the low-frequency vibroactive forces range (0.5 - 20 Hz), a promising direction for the vibration isolation purposes is the use of vibroactive forces inertial compensation by an electrodynamic compensator. It ensures the movement of the mass in antiphase with the vibroactive mechanism oscillations (7 - 19).

2. Problem formulation
Application of the principle of vibroactive forces inertial compensation in the low-frequency range (1 - 20 Hz) for active vibration isolation systems has shown both the effectiveness of this principle and the need to create a fairly simple and cheap actuator that ensures the operation of the active vibration isolation system.

The principle of inertial compensation is that the actuator provides the displacement of the mass, which creates inertial forces in antiphase with vibroactive forces.

An electrodynamic drive with an increased displacement of a movable permanent magnet can be used as such an actuator.

A schematic diagram of such an actuator is shown in fig. 1.
Figure 1. Actuator schematic diagram

1 is the housing made of soft magnetic material, 2 is the base, 3 is the control coil, 4 is the guide made of non-magnetic material, 5 is the sleeve made of non-magnetic material, 6 is the permanent neodymium magnet, 7 is the magnetic circuit, 8 are the springs, 9 is the alternating voltage source.

When an alternating voltage from 9 is applied to the coil 3, the movable link (5, 6, 7) makes a translational movement along the guide 4 with an amplitude proportional to the current of 3 and with the frequency of the applied voltage.

The inertial force $F$ is transmitted to the base and is equal in amplitude $F = m \cdot x \cdot \omega^2$, where $m$ is the mass of the moving part, $x$ is the amplitude of the moving part, $\omega$ is the circular vibration frequency.

The use of the proposed electrodynamic actuator to create inertial forces is possible in the following three options.

The first option - the actuator is installed in the immediate vicinity to the passive system elastic-dissipative supports or is performed in one structure with the support. A schematic diagram of such an active vibration isolation system is shown in Fig. 2.

Figure 2. Schematic diagram

$F(t)$ is vibroactive force, 1 is the vibroactive unit mass, 2 is mass of the actuator moving part, 3 is electromotive force $F_{act}$ developed by the actuator, 4 is regulator with a transfer ratio $K$, 5 is force-measuring device, $m_0$ is the vibroactive unit mass, $m_1$ is the actuator moving part mass, $c_0, c_1$ are the elastic elements stiffness, $b_0, b_1$ are the viscous friction coefficients.

Force $F_{act}$ is determined in accordance with the following equations:

\[
\begin{align*}
F_{act} &= Bl \ i, \\
L \ \frac{di}{dt} + Ri + Bl \ \dot{x}_1 &= u, \\
u &= \Delta R \ K, \\
\Delta R &= c_0 x_0 + b_0 \dot{x}_0 - m_1 \ddot{x}_1.
\end{align*}
\]
where
- $u$ is voltage on the winding of the control coil,
- $i$ is current strength,
- $F_{act}$ is electrodynamic force,
- $L, R$ is inductance and resistance of the coil,
- $B$ is magnetic induction,
- $l$ is total length of the coil conductor.

Study of the active system efficiency in accordance with fig. 2 [7] [8] showed that the reduction in the transmission of vibration force to the base is $20 - 40$ dB in the frequency range $1 - 20$ Hz.

The next option for the actuator is shown in fig. 3, where the actuator is based at mass $m_0$.

![Design scheme](image)

**Figure 3.** Design scheme

1 is the vibroactive unit mass, 2 is mass of the actuator moving part, 3 is electromotive force $F_{act}$ developed by the actuator, 4 is regulator with a transfer ratio $K$, 5 is force-measuring device, $m_0$ is the vibroactive unit mass, $m_1$ is the actuator moving part mass, $c_0, c_1$ – are the elastic elements stiffness, $b_0, b_1$ – are the viscous friction coefficients

Fig. 3 designations correspond to fig. 1, and the force $F_{act}$ is determined by the following equations:

$$
\begin{cases}
L \frac{di}{dt} + R i + Bl (\dot{x}_1 - \dot{x}_0) = u, \\
u = K R_{base}, \\
R_{base} = c_0 x_0 + b_0 \dot{x}_0.
\end{cases}
$$

In this case, the electromechanical system in fig. 3 is a vibration isolation system with an active dynamic vibration damper. The efficiency of such a system has been studied in detail in [11] [14]. It is shown there that the reduction in force transmission to the base is $20-40$ dB in the frequency range $1-20$ Hz. In addition, it was shown that the system remains effective in the non-stationary mode of the vibroactive unit, for example, "start-stop" [14].

An active dynamic vibration damper with an electrodynamic actuator can be implemented by installing the EDC on the base and connecting the moving mass of the EDC by a spring with a vibroactive mass.

3. **Theory**

To compile a mathematical model of an active dynamic vibration damper, a schematic diagram is presented in Fig. 4.
Figure 4. Schematic diagram
Designations 1 - 8 correspond to Fig. 1, 9 is the force-measuring device, $K$ is the controller, $m_0$ is the vibroactive unit mass, $c_0,c_1$ – are the elastic elements stiffness, $b_0$ – is the viscous friction coefficients.

In accordance with fig. 4, the design scheme is shown in Fig. 5.

Figure 5. Schematic diagram
1 is regulator, 2 is electromotive force, 3 is force-measuring device.

Assuming that the two-mass system in fig. 6 performs unidirectional motion about the equilibrium position, the equations system describing the dynamics of the system will have the form:

$$
\begin{align*}
    m_0\ddot{x}_0 + b_0\dot{x}_0 + c_0x_0 + c_1(x_0 - x_1) + b_1(\dot{x}_0 - \dot{x}_1) &= F(t), \\
    m_1\ddot{x}_1 + b_1(\dot{x}_1 - \dot{x}_0) - c_1(x_1 - x_0) &= Bl \, i, \\
    L \frac{di}{dt} + R \, i + Bl \, (\dot{x}_1 - \dot{x}_0) &= u, \\
    u &= K \, R_{base}, \\
    R_{base} &= c_0x_0 + b_0\dot{x}_0 - Bl \, i.
\end{align*}
$$

(3)
The equations system (3) was analyzed by using the Matlab / Simulink program. The model in the Simulink program is shown in fig. 6. To assess the efficiency of the system under consideration, the following model parameters are taken: \(m_0 = 200 \, kg\), \(m_1 = 20 \, kg\), \(c_0 = 8,08 \cdot 10^4 \, \frac{N}{m}\), \(b_0 = 8 \cdot 10^2 \, \frac{N \cdot s}{m}\), \(c_1 = 3,2 \cdot 10^3 \, \frac{N}{m}\), \(b_1 = 50 \, \frac{N \cdot s}{m}\), \(BL = 10 \, T \cdot m\), \(L = 5 \cdot 10^{-3} \, H\), \(R = 10 \, \Omega\).

![Model in the Simulink program](image)

**Figure 6.** Model in the Simulink program

As a efficiency criterion, it is advisable to take the coefficient \(K_T\) of the transfer of force to the base, determined by the expression \(K_T(\omega) = \frac{|R_{\text{base}}(i\omega)|}{|F(i\omega)|}\). The \(K_T(\omega)\) dependency graph at \(K = 10\) and \(K = 100\) is shown in fig. 7.

![Bode Diagram](image)
For clarity, fig. 8 shows the values of the force on the base without the EDC and with the EDC turned on at a frequency of the vibroactive force of 1 Hz.

Analysis of the equations system (3) showed that the system is stable at any positive values of $K$. With an increase in $K$, the efficiency of vibration isolation in the low-frequency range increases, the oscillation index and the period of natural oscillations increase.

4. Conclusion
The study of the effectiveness of an active vibration damper when installing an EHD on the basis showed that this direction is promising for vibration isolation in the low-frequency region of 1 - 20 Hz, including at the natural frequency of a passive system.
Analysis of possible options for including EHD in the active vibration isolation system showed that when controlled from a force-measuring device, their efficiency is the same and is 20 - 40 dB in the range of 1 - 20 Hz.
The choice of one or another option for using the EDC depends on the design features of the construction of a passive vibration isolation system.

References
[1] Kiryukhin A V, Tikhonov V A, Chistyakov A G and Yablonsky V V 2011 Active vibration protection - purpose, principles, condition. 1. Purpose and principles of development Problems of mechanical engineering and automation No 2 pp 108–111
[2] Kiryukhin A V, Tikhonov V A, Chistyakov A G and Yablonsky V V 2011 Active vibration protection - purpose, principles, condition. 2. A history of development and a condition Problems of mechanical engineering and automation No 3
[3] Pat. 2556867 Russian Federation, IPC B63G 8/34. Active vibration isolation system of pipelines of the emergency cooling system of a nuclear reactor of a submarine / Kiryukhin A. V., Fedorov V. A., Milman O. O. “2013158496/07; declared 12/30/2013; publ. 07/20/2015, Bul. No. 20.
[4] Nijsse Gerard 2006 A subspace based approach to the design, implementation and validation of algorithms for active vibration isolation control Ph.D. Thesis (University of Twente – Netherlands, Enschede)
[5] Fowler L P, Buchner S P and Ryaboy V M 2000 Self-contained active damping system for pneumatic isolation tables Proceedings of SPIE - The International Society for Optical Engineering vol 3991 p 261; doi:10.1117/12.388168
[6] Burian Yu A, Shalai V V, Zubarev A V and Polyakov S N 2017 Dynamic compensation of vibroactive forces in the oscillatory system Mechantronics, automation, control No 3(18) pp.
[7] Burian Yu A, Sitnikov D V and Silkov MV 2020 Inertial compensation of vibroactive forces in the oscillatory system Dynamics of systems, mechanisms and machines vol 8 No 1 pp 16–23
[8] Burian Yu A, Sitnikov D V, Silkov M V and Belkov V N 2020 Active vibration isolation system with digital twin and acceleration control Dynamics of systems, mechanisms and machines Vol 8 No 1 pp 9–16
[9] Burian Yu A and Sitnikov DV 2019 Active vibration isolation system with electrodynamic drive Problems of engineering. Materials of the III International Scientific and Technical Conference pp 56–62
[10] Burian Yu A , Zubarev A V and Polyakov S N Dynamic vibration damper. Patent for invention RU 2654241 C2, 17.05.2018. Application No. 2016108574 dated 09.03.2016
[11] Burian Yu A, Sitnikov D V, Burian A A and Kalashnikov B A 2018 On the question of the influence of control laws on the efficiency of an active dynamic vibration damper Dynamics of systems, mechanisms of machines Vol 6 No 1 pp 17–26
[12] Burian Yu A, Zubarev A V, Silkov M V and Shalay V V 2017 Active low-frequency vibration isolation system with dynamic forces compensation Vestnik mashinostroeniya No 6 pp 18–22
[13] Burian Yu A, Shalai V V, Zubarev A V and Polyakov S N 2017 Dynamic compensation of vibroactive forces in the oscillatory system Mechatronics, automation, control Vol 18 No 3 P. 192-195.
[14] Burian Yu A, Sitnikov D V, Silkov M V and Burian A A 2021 Vibration isolation system with an active dynamic vibration damper during non-stationary operation of the piston machine Problems of mechanical engineering. Materials of the V International Scientific and Technical Conference pp 18–25
[15] Burian Yu A and Sitnikov D V 2020 The active system of vibration isolation with electrodynamic actuator Journal of Physics: Conference Series. XIII International Scientific and Technical Conference "Applied Mechanics and Systems Dynamics” 012089
[16] Kiryukhin A V, Milman O O, Ptakhin A V and Serezhkin L N 2018 Test results of an active system for reducing vibration forces and pressure pulsations Journal of technical physics vol 44; DOI:10.21883/PJTF.2018.24.47028.17443
[17] Khot S M, Yelve N P, Tomar R, Desai S and Vittal S 2012 Active vibration control of cantilever beam by using PID based output feedback controller Journal of Vibration and Control 18(3) pp 366–372
[18] Omidi E and Mahmoodi N 2016 Vibration suppression of distributed parameter flexible structures by Integral Consensus Control Journal of Sound and Vibration 364 pp 1–13
[19] Kiryukhin A V, Milman O O, Ptakhin A V, Serezhkin L N and Kondratev A V 2017 Development and Calculation-Experimental Analysis of Pressure Pulsations and Dynamic Forces Occurrence Models in the Expansion Joints of Pipelines with Fluid International Journal of Applied Engineering Research 12(19) pp 8209–8216
[20] Feng N S, Hahn E J and Randall R B 2004 Simulation of vibration signals from a rolling element bearing defect (University of New South Wales)
[21] Tinghsu S, Hattori S, Ishida M and Hori T 2002 Suppression control method for torque vibration of AC motor utilizing repetitive controller with Fourier transform IEEE Transactions on Industry Applications Vol 38, 15 pp 1316–1325