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Active Vibration Isolation Devices with Inertial Servo Actuators

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Abstract—The use of active vibration isolation devices (AVIDs) in aerospace engineering is subject to the following restrictions. First, the volume for installing additional devices is always limited in instrument racks and compartments. Secondly, in many cases, it is impossible to add supports for servo actuators for fundamental or design considerations. In the paper, it has been shown that this problem can be solved if the inertial servo actuators are used in AVIDs instead of reference actuators. A transfer function has been theoretically calculated for an AVID controlled by inertial actuators. It has been shown that the volume of a six-mode single-housing AVID with inertial actuators can be 2–2.5 times smaller than that of devices with support actuators.

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INTRODUCTION

Recently, modern analytical/measuring and technological equipment for aerospace and transport applications, as well as for laboratory/workshop conditions, has increasingly required active vibration isolation devices (AVIDs).

For example, an important research area on board the spacecraft that requires vibration isolation equipment is the study of the effect of residual microaccelerations on convection during the growth of crystals, which leads to the formation of various kinds of inhomogeneities [1–3].

The most popular designs of commercial AVIDs in the form of tables consist of a base plate, a bearing plate installed on it using elastic elements, and a group of accelerometers and servo actuators symmetrically located in the space between the plates and resting on the base plate. Automatic electrical circuits make it possible to suppress all six modes of oscillations of the bearing plate with the equipment to be protected on it [4–6]. Vibration isolation instrument racks ARIS and g-LIMIT devices installed in the US sector of the *ISS* also contain servo actuators resting on the base.

For stationary equipment, AVIDs in the form of the tables described above can be considered a universal design. However, in some cases, these AVIDs can prove unsuitable for aerospace and transport applications for a number of reasons, the main one of which is the large volume of the mechanical part of the device. As will be seen, the volume of the AVID with inertial servo actuators can be significantly (by 2–2.5 times) smaller than that with actuators resting on the base. In addition, in some cases, supports for servo actuators cannot be added for principle or design considerations.

INERTIAL ACTUATORS IN THE AVID CONTROL CIRCUIT

Figure 1 shows an AVID circuit with an inertial actuator, where m_1 is the inertial mass connected via the elastic element c_1 with mass m_2 that represents the bearing body of the vibration isolation device. The inertial mass is driven by the electrodynamic transducer represented on the circuit by the magnetic system MS, in the gap of which the electric coil EC is located. The body rests on the base with an elastic element c_2. In the circuit, $F_1(t)$ and $F_2(t)$ are the forces applied by the electrodynamic transducer to the inertial mass and the body, $a_1(t)$ and $a_2(t)$ are the corresponding accelerations. The signal $K_a a_2(t)$ of the accelerometer A installed on the carrier plate (K_a is the voltage transfer ratio of the accelerometer) and the voltage on the coil $u(t)$ are the output and input of the dynamical system, respectively.

A system with lumped parameters is considered below because it is assumed that transverse vibrations of the housing/carrier plate are not excited in the active frequency range.

The dynamics of the composite vibration system shown in Fig. 1 is determined by the housing resting on the elastic element c_2 with the mass m_2, the motion of which is controlled by the inertial mass m_1 (Newtonian inertial force $F = m_1 a_1$) driven by an electrodynamic transducer. The desired complex transfer function of this circuit $W_2(i\omega)$ that reflects its dynamic characteristics is determined by the ratio a_2/u, where a_2 and u are Fourier transforms of the functions $a_2(t)$ and $u(t)$, which represent the housing acceleration and the voltage at the input of the electrodynamic transducer, respectively.

Calculations of the transfer function $W_2(i\omega) = a_2/u$ are simplified because the inertial mass is much smaller than the mass of the housing, where $m_1 \ll m_2$. Therefore, when the vibrations are excited by the electrodynamic transducer, we can consider $a_2 \approx 0$ and $F_2 = -F_1 = F = m_1 a_1$. This approximation becomes even more accurate with the live automatic circuit because of the multiple increase in effective mass of the bearing plate.

In this approximation, the transfer function of acceleration of the electrodynamic transducer loaded with mechanical impedance $z_1(i\omega)$ is given by the following formula [7, 8]:

$$W_1(i\omega) = \frac{a_1}{u} = -i\omega\left(\frac{Z(i\omega)z_1(i\omega)}{K} + K\right)^{-1}. \qquad (1)$$

Here, a_1 is the acceleration of the inertial mass, u is the voltage at the input of the electrodynamic transducer, $K = Bl$ is the coefficient of the electromechanical coupling of the transducer (B is the induction in the gap of the magnetic system and l is the wire length in the magnetic field), and $Z(i\omega) = R + i\omega L$ is the electric impedance of the transducer coil (R is its electrical resistance and L is inductance). The mechanical impedance $z_1(i\omega) = r_1 + i\omega m_1 + 1/i\omega c_1$ includes the coefficient of friction r_1, the inertia mass m_1, and the elastic compliance of the support of the inertial mass c_1.

Acceleration of the housing controlled by the Newtonian inertia force $F = m_1 a_1$ is determined by the following ratio:

$$a_2 = i\omega\frac{F}{z_2(i\omega)} = i\omega\frac{m_1 a_1}{z_2(i\omega)}. \qquad (2)$$

Here, $z_2(i\omega) = r_2 + i\omega m_2 + 1/i\omega c_2$ is the mechanical impedance of the vibration system, which includes the coefficient of friction r_2, the mass of the housing m_2, and the elastic compliance of the supports c_2, on which the housing is mounted.

From Eqs. (2) and (1), we obtain the transfer function of a dynamical system, the input of which is the voltage on the coil of the electrodynamic transducer, and the output is the acceleration of the bearing housing as follows:

$$W_2(i\omega) = \frac{a_2}{u} = \frac{m_1\omega^2}{z_2(i\omega)}\left(\frac{Z(i\omega)z_1(i\omega)}{K} + K\right)^{-1}. \qquad (3)$$

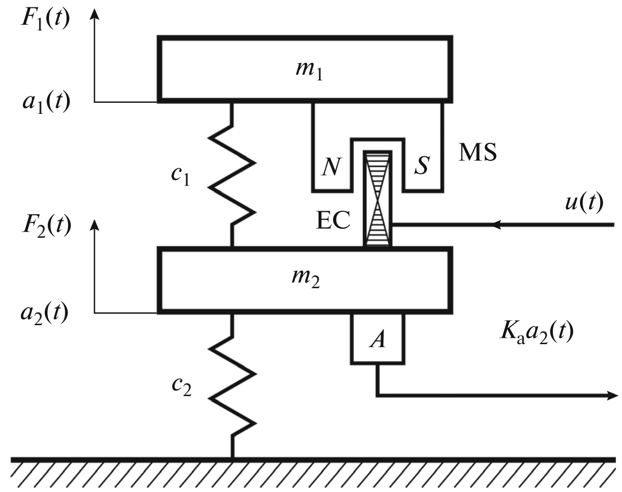

Fig. 1. Single-mode diagram of an active vibration isolation device with an inertial servo actuator.

The parameters of all design elements of AVIDs with inertial actuators can be obtained using Eq. (3). The inertial mass m_2, the elastic compliance of the suspension c_2, and the parameters of the electrodynamic transducer can be determined based on the specified active frequency range of AVIDs, the mass of the carrier plate m_1, and the elastic compliance of the supports c_1.

The calculation of $W_1(i\omega) = a_1/u$ and $W_2(i\omega) = a_2/u$ transfer functions defined by Eqs. (1) and (3) for a specific vibration isolation device designed for a protected object weighing up to 100 kg is given below as an example. In this case, a carrier plate with a mass of $m_2 \approx 23$ kg can be installed on four springs so that the resonance of the mode of vibration in the direction normal to the plane of the plate is ~20 Hz. Four inertial actuators located in the corners of the plate that control the movement of the plate have a mass of $m_1 = 0.5$ kg and a natural resonance frequency of ~1 Hz. Coils of electrodynamic transducers have a resistance of $R = 12$ Ohm and inductance of $L = 22$ mH. Magnetic induction in the magnet gap is $B \approx 0.2$ T, and the wire length in the magnetic field is $l = 50$ m so that the coefficient of electromechanical coupling of actuators is $K = Bl = 10$.

Figure 2 shows the frequency dependences of the moduli of the transfer functions of the inertial actuator $W_1(i\omega) = a_1/u$ and the housing $W_2(i\omega) = a_2/u$ controlled by the inertial actuator determined by Eqs. (1) and (3) with the above parameters of the specific vibration isolation device.

As can be seen from Fig. 2, a plane section is observed on the curve $|W_1(i\omega)|$ in the frequency region between the resonance of the actuator of ~1 Hz, and the frequency is ~20 Hz. In the case of

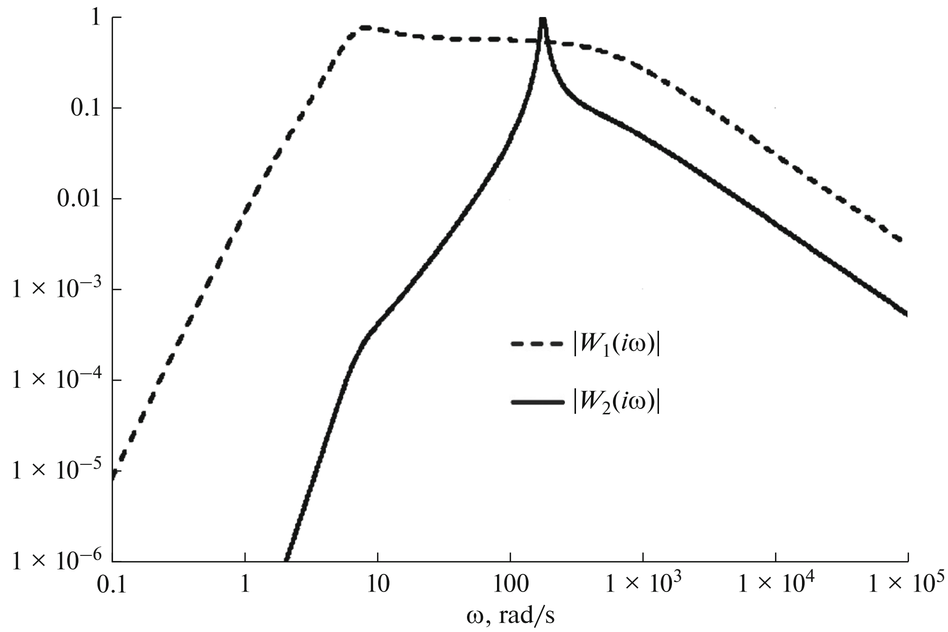

Fig. 2. Frequency dependences of the acceleration transfer function modulus of (1) inertial drive $W_1(i\omega) = a_1/u$ (dotted curve) and (2) housing controlled by inertial drive $W_2(i\omega) = a_2/u$ (solid curve).

the frequency above ~20 Hz, a straight line with a slope of –20 dB/dec is observed and below ~1 Hz, there is a straight line with a slope of 40 dB/dec. As shown in [7], this is a typical form of the transfer function of an electrodynamic transducer loaded with a large mass. As can be seen from Eq. (3), the transfer function $W_2(i\omega)$ is a product of the transfer functions of two dynamic elements, i.e., the mechanical system $i\omega/z_2(i\omega)$ (resonance of the carrier plate) and the inertial drive (Eq. (1)). Since, on the presented logarithmic scale, the modulus of the full-circuit transfer function is the sum of the moduli of transfer functions of the individual element, the resonance maximum is at ~20 Hz and lines with slopes of –20 dB/dec above ~20 Hz, 40 dB/dec in range of ~1–20 Hz, and 80 dB/dec lower than ~1 Hz are observed.

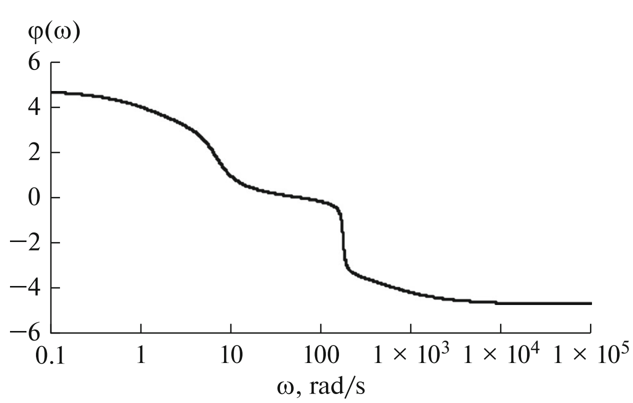

Fig. 3. Frequency dependence of phase angle φ(ω) of the transfer function of the carrier plate $W_2(i\omega) = a_2/u$.

In Fig. 3, the large total phase decrease is associated with the fact that the series-connected dynamic links shown in Fig. 1 contain three resonances, each of which lowers the phase by 180°, including (1) the resonance of the carrier plate at a frequency of ~20 Hz, (2) the resonance of the inertial mass at a frequency of ~1 Hz, and (3) the electromechanical resonance in the electric circuit of the actuator at a frequency of ~4.5 Hz [7].

It can be concluded from Figs. 2 and 3 that the lower limit of the active range of AVIDs with inertial actuators is limited by the resonance frequency of the inertial mass. Indeed, as can be seen in Fig. 2, when the frequency is lowered in the region below ~1 Hz, the modulus of the transfer function $W_2(i\omega)$ decreases rapidly (slope of ~80 dB/dec). Figure 3 shows that, in the frequency range above ~1 Hz, the phase can be lowered in order to ensure its stability using electric adjusters in the automatic circuit [9]. Thus, for an AVID with the specified parameters, the active frequency range can have limits of 1–200 Hz (or 1 Hz to 2 kHz) and a maximum vibration suppression factor of ~50 dB.

It should be noted that the total volume of the mechanical part of single-housing AVIDs can be significantly (2–2.5 times) smaller compared to conventional designs due to the exclusion of the base plate and the space between the base and bearing plates in which support actuators are located.

CONCLUSIONS

Single-housing AVIDs with inertial servo actuators placed in the bearing plate can be used as the main element in vibration isolation devices with different configurations. Thus, AVIDs in the form of a carrier plate (single housing) can be fixed on any suitable surface (surfaces) of the protected device already equipped with elastic supports. This arrangement of AVID facilitates its adaptation to instruments located in racks or in instrument compartments that limit the dimensions of additional devices. Since the volume of the mechanical part of the single-housing AVID is significantly (in 2–2.5 times) less compared to the conventional design, their prospects are promising for aerospace applications. In addition, a single-housing AVID can be attached to vehicle components, housing parts, and overhauls in order to suppress their lateral vibrations.

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