Active vibration isolation of high-vacuum nanotechnology equipment

V P Mikhailov, A M Bazinenkov, A V Kazakov, A P Rotar’ and D A Ivanova
Bauman Moscow State Technical University, 105005, Moscow, Russia

E-mail: mikhailov@bmstu.ru

Abstract. The results of experimental studies of the transients’ parameters during the operation of a high-precision drive with a closed-loop control, which at the same time is an active damper, are presented. The amplitude-frequency characteristics of the damper and the vibration-isolating platform in semi-active mode based on four dampers and four elastic suspension assemblies are also shown.

Current trends towards miniaturization of micro- and nanoelectronics and micromechanics devices and instruments establish key requirements for minimizing the size of elements in integrated circuits and micromechanical components. Nanolocal treatment with ionic and electron beams, micro- and nanoscale lithography in vacuum, intermediate surface control in vacuum are the main technological operations for manufacturing of micro- and nanoelectronics and micromechanics products.

As an example, figure 1 shows a high-vacuum scanning electron microscope (SEM) diagram. The SEM contains an electron-optical system (including an electron gun 1, a focusing system 2, a scanning system 3, a projection lens 4) providing the formation and scanning of an electron beam 5, a high-vacuum chamber 6, an active vibration isolation system 7, fixed on the coordinate table 8 sample 9 which reflects the secondary electrons flux 10, the detector of secondary electrons 11. Mechanical isolated mounting system of the main SEM units, including electron-optical system assemblies 1, 2, 3, 4, high-vacuum chamber 6, coordinate table 8, sample 9, detector of secondary electrons 11, has low rigidity and low resonant frequencies generated by these inertial elements. The SEM resonant frequencies manifest themselves in the range of 0.5 to 10 Hz and higher, i.e. in the range in which the existing passive and active vibration isolation systems (pneumatic, hydraulic, piezoelectric, etc.) do not work effectively enough [1–8]. Solving the problem of effective protection from external vibration disturbances in the low-frequency region requires fundamentally new approaches based on using of high-precision devices that provide active vibration isolation and positioning of the protected object in the submicron range with millisecond response speed [9–11].

The results of experimental studies of the transients’ parameters during the operation of a high-precision drive with a closed-loop control, which at the same time is an active damper (figure 2), are presented. The amplitude-frequency characteristics of the damper and vibration-isolating platform with an open-loop control system in a semi-active mode based on four dampers and four elastic suspension assemblies are also shown. The purpose of the first series of experiments was to evaluate the quality parameters of transients in a closed-loop control system of the active damper, depending on the regulator transmission coefficient. A personal computer with a LabView software, a National Instruments USB-6009 ADC/DAC unit, an amplifier unit consisting of 4 amplifying modules based on
Operational amplifiers, a DL6220/ECL2 capacitance displacement sensor with a measurement error of 0.1 micron were used to control the active damper. The active damper (figure 2) is an electromagnetic drive with damping properties, containing an electromagnetic coil, magnetic circuit and membrane based on a magnetorheological (MR) elastomer with a rigid core [9]. Creating a magnetic field in the membrane, one can adjust the elastic properties of the membrane made of MR elastomer and move the rigid core in the axial direction. As a result of experiments, transient response plots were obtained for different values of the transmission coefficient k of the control system regulator (figure 3). From the presented plots it can be seen that with an increase in the transmission coefficient k of the regulator, the oscillation and the system overshoot value increase, while the transient time of the positioning process remains almost unchanged.

Figure 1. High-vacuum scanning electron microscope (SEM) diagram.

Figure 2. Active damper with sensor mounted.

Figure 3. Transient plots when moving the active damper with a closed-loop control system over a distance of 200 μm for different values of the regulator transmission coefficient k: 1 – 0.002; 2 – 0.005; 3 – 0.01.

The purpose of further studying was to evaluate the dampers operating performance in semi-active mode. It is known that the use of the MR-effect allows one to adjust the stiffness coefficient of the elastic membrane made of MR elastomer by changing the value of magnetic induction and, accordingly, the frequency and accuracy characteristics of the active damper [10]. Due to this, it is possible to achieve a shift in the resonant frequency of the sample. This allows one to configure the device to work in the desired frequency range. To perform studies of the damper in semi-active mode on an experimental test bench, the scheme of which is shown in figure 4, the following instruments were used: an amplifier 1, a personal computer 2, ADC 3, piezoelectric accelerometers 4 and 5, an active damper 6, a source of vibration disturbances 7 (Data Physics Vibrator V300), a power supply unit 8 (Mastech). A feature of the piezoelectric accelerometers 4 and 5 is the built-in integrator, which allows to translate the obtained vibration acceleration values into vibration displacement values.
Figure 5 shows a general view of the test bench and its main elements: an active damper 1, a tool 2 for clamping the damper, a source of vibration disturbances 3.

![Figure 4. Scheme of the test bench for studies of the damper in semi-active mode.](image)

![Figure 5. General view of the test bench for studies of the damper.](image)

The experiment procedure is as follows: a fixed control current in the range of 0 – 1 A is applied from the power supply unit 8 to the damper 6, the control software is launched on the personal computer 2, activating the operation of the source of vibration disturbances 7. Measurement parameters are selected (frequency range of 15 – 200 Hz, measurement time 2 min.) Accelerometer 4 measures the acceleration of the source of vibration disturbances. Accelerometer 5 measures the acceleration of the rigid core of the active damper 6 membrane. The signal from the sensors enters the ADC 3 and then it is stored in the control software on the personal computer 2. As a result, experimental values of vibration displacement in the frequency range of 15 – 200 Hz for each active damper were obtained. Data processing was performed in Microsoft Excel. The resulting plots of the transmission coefficient $K$ of the vibration displacement amplitude versus frequency are shown in figure 6.

![Figure 6. Plots of the transmission coefficient $K$ of the damper vibration displacement amplitude versus frequency in semi-active mode for the values of the control current: 1 – 0A; 2 – 0.5A; 3 – 1.0A.](image)

The resulting dependence curve can be conditionally divided into two sections. The first section of the resonance is 15 – 76 Hz, with the peak of the resonance located at a frequency of about 48 Hz, which corresponds to the natural frequency of the damper. An increase in the control current contributes to a decrease in the transmission coefficient $K$ from a value of 3.18 with zero control action to a value of 2.77 for control current of 1A, and to a shift of the resonant region to the left (resonance region at a control current of 1A is 15 – 70 Hz). The second section of effective vibration damping 77 – 200 Hz is characterized by a tendency for the transmission coefficient $K$ to decrease with an increase in the vibration frequency (minimum value $K = 0.29$ without control action, minimum value $K = 0.28$ for a
current of 1 A at a frequency of 200 Hz). Thus, the dampers in the semi-active operation mode are most effective for the frequency range of 80 – 200 Hz, the natural frequency of the damper is 48 Hz.

The next stage of study is to evaluate the performance of the vibration-isolating platform in semi-active mode for the frequency range of 15 – 100 Hz. The resonance region of the dampers is in the frequency range of 15 ... 80 Hz, so it is necessary to evaluate the platform performance in this region.

To perform study of the platform in semi-active mode on the test bench, the scheme of which is shown in figure 7, the following instruments and devices were used: an amplifier 1, a personal computer 2, ADC 3, accelerometers 4 and 5, a platform 6, power supplies 7 and 8; a source of vibration disturbances 9. Vibration-isolating platform 6 contains four active dampers and four elastic suspension assemblies.

The technique of the experiment is as follows. The fixed control current from the power supply units 7 and 8 is applied to the electromagnetic coil of each damper. On the personal computer 2, the control software is launched and activate the operation of the source of vibration disturbances 9. Measurement parameters are selected (frequency range of 15 – 100 Hz, measurement time 2 minutes). Accelerometer 4 measures the acceleration of the platform bottom plate. Accelerometer 5 measures the acceleration of the platform top plate. The signals from the sensors 4, 5 enter the ADC 3, and then they are stored in the control software on the personal computer.

Figure 8 shows a general view of the test bench, containing a platform 1, accelerometers 2 and 3, tool 4 for clamping the platform, a vibrator 5, power supply units 6. As a result, the experimental values of vibration movements for the frequency range of 15 – 100 Hz were obtained. Data processing was performed in Microsoft Excel. The resulting plots of the transmission coefficient K of the vibration displacement amplitude versus frequency are shown in figure 9.

The resulting dependence curve can be conditionally divided into two sections. The first section of the resonance 15 – 39 Hz contains two peaks at frequencies of 21 and 28 Hz. An increase in the control current contributes to a decrease in the transmission coefficient K at the first peak (from a value of 3.26 with zero control action to a value of 3.22 for a control current of 0.55A) and to a shift of the resonance region to the left (resonance region at a control current of 0.55A is 15 – 38 Hz).

The second section of effective vibration damping 40–100 Hz is characterized by a tendency for K to decrease with an increase in the vibration frequency (minimum value K = 0.18 without control action, minimum value K = 0.16 for a current of 0.55 A at a frequency of 100 Hz).
Figure 9. Plots of the transmission coefficient K of the platform vibration displacement amplitude versus frequency in semi-active mode for the values of the control current: 1 – 0A; 2 – 0.55A.

Thus, the platform in the semi-active operation mode is most effective for the frequency range of 40 – 100 Hz, the natural frequencies of the platform are 21 and 28 Hz. Ensuring effective vibration isolation in the region of resonance frequencies can be achieved by using a closed-loop control system in the platform position stabilization mode, which is the subject of further studying.

Acknowledgement
The study was supported by the Ministry of Science and Higher Education of the Russian Federation as part of conducting research and development in the framework of the basic government assignment in the field of science no. 9.8503.2017/8.9. The authors would like to thank M.S. Ondrin and Makeev I.V., students of Bauman Moscow State Technical University, for their contribution to preparing the article for publication.

References
[1] Wigglesworth W and Jordan S 2009 Semicond. Int. 32(10) 4–26
[2] Active Vibration Isolation. Accurion. http://www.accurion.com
[3] Würzburg B H and Grossenlupnitz R R US Patent No. 20080318045 A1. Appl. No. 11/574397, 25.08.2005, Date of Patent 27.08.2004
[4] Gruzevich Yu K, Soldatenkov V A, Achil’diev V M, Levkovich A D, Bedro A N, Komarova M N and Voronin I V 2018 Journal of Optical Technology 85(5) 308–13
[5] Ovchinnikov I and Brancevich P 2017 Procedia Engineering 176 610–7
[6] Krestnikovskiy K V, Panovko G Ya and Shokin A E 2016 Vibroengineering Procedia 8 208–12
[7] Panovko G, Shokin A and Eremeykin S 2016 Vibroengineering Procedia 8 174–8
[8] Chernikov S A 2015 Journal of Machinery Manufacture and Reliability 44 439
[9] Mikhailov V P and Bazinenkov A M 2017 Journal of Magnetism and Magnetic Materials 431 266–8
[10] Mikhailov V P, Bazinenkov A M, Dolinin P A and Stepanov G V 2018 Instruments and Experimental Techniques 61(3) 427–32
[11] Mikhailov V P, Bazinenkov A M, Dolinin P A and Stepanov G V 2018 Russian Engineering Research 38(6) 434–7