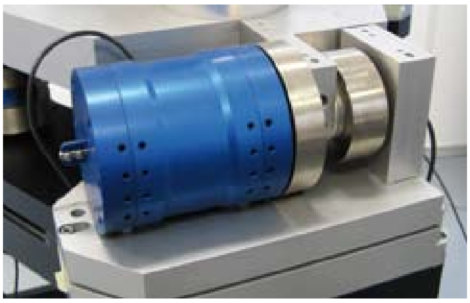
The "ultimate performance" in floor vibration isolation

Commercially available active vibration isolation systems basically are a combination of a passive compliant suspension with active damping of the low-frequency suspension modes. However, with the exception of the semiconductor industry they have not yet found widespread application. Until now performance gain was limited and horizontal-tilt crosstalk demanded for complex control strategies. Therefore, the actual benefits appear not sufficient to accept the added cost. MI-Partners has developed a new 6-DoF active vibration isolation platform with high performance, limited control complexity and high cost efficiency. Fundamental improvements concern crosstalk elimination and noise reduction for better performance. Dynamic Error Budgeting was used in the design phase to predict the integral influence of vibrations.



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Authors' note

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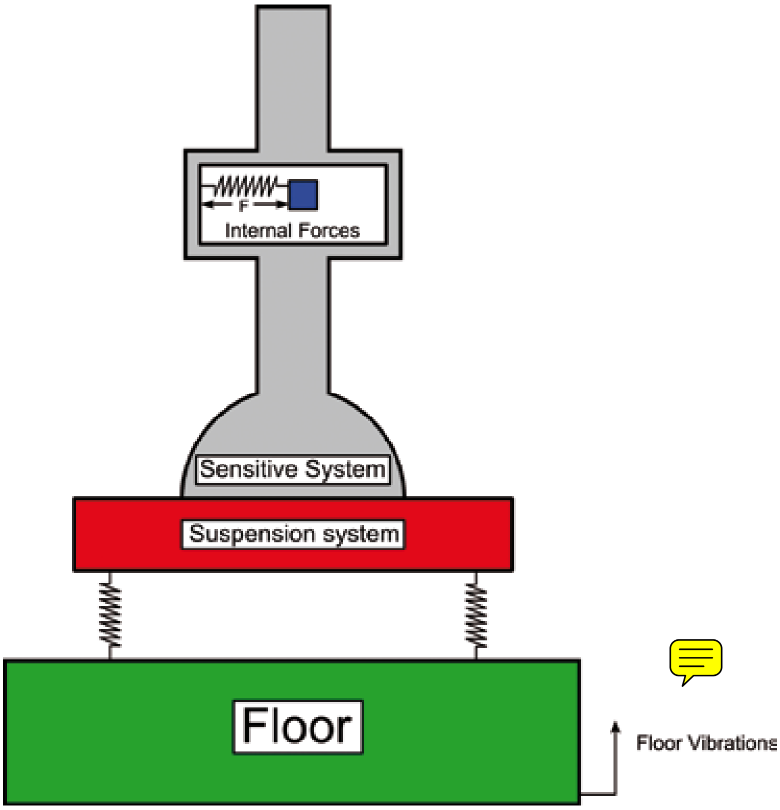


Figure 1. Two disturbance sources for a sensitive instrument: internal forces and floor vibrations.

Introduction

The rise of nanoscale science and technology is creating new challenges in the field of instrument and equipment design. The relative positions of parts of the system must be maintained to subnanometer levels. When imaging at a resolution of 50 pm, the allowable relative motions will be in the order of 10 pm. System vibrations lead to internal deformations – therefore, they must be avoided or sufficiently reduced. The matter has been addressed in recent Mikroniek issues [1] [2].

Figure 1 shows the schematic representation of a sensitive instrument, subject to two types of disturbances. Internal forces may result from driving forces needed to move parts of the instrument. In that case the total system will start to move on the suspension and the resulting forces will lead to internal deformations of the instrument. However, in this article the focus is on the other disturbance source, floor vibrations. These vibrations will be transmitted through the suspension to the instrument and will generate internal deformations.

The design of the suspension system can be aimed at reducing the amount of vibration that is transmitted. In that case, the suspension is designed to be a floor vibration isolator. Both passive and active designs have been proposed and used. To assess the quality of such a

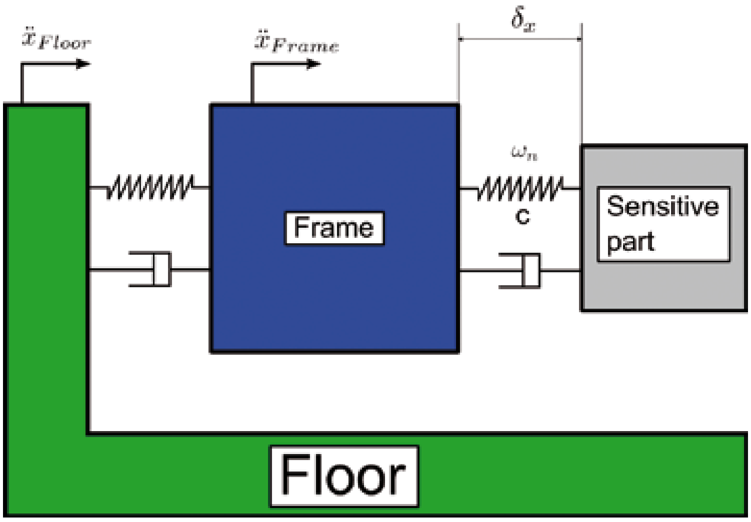


Figure 2. A precision system with a sensitive part connected to a frame by a finite stiffness. Upon force transmission this spring will deform.

suspension system, preferably the final positioning performance of the instrument should be considered.

From vibration to performance

In precision systems the performance is mostly expressed in terms of positioning errors. In an imaging system, however, the mean value of a position error is not as important as the position variations during imaging. Such variations can be characterised using the standard deviation or RMS value of the position error.
Relative motion of parts in a system occurs when forces are transmitted through mechanical parts. Figure 2 shows the schematic of a precision system with a sensitive part connected to a frame. The connection is not infinitely stiff and thus a deformation of this spring will occur when a force is transmitted.

For frequencies below the resonance frequency ω_n of this system the deformation is calculated as $\delta_x = F / c$. When the frame is vibrating with frequencies below the resonance frequency, the force transmitted by the spring is calculated with F = mass · acceleration. And thus the expression for the deformation becomes:

$$\delta_x = \text{acceleration} \cdot \text{mass} / c = \text{acceleration} / \omega_n^2$$

The ratio between stiffness and mass depends on the natural frequency of the mechanical system. Higher internal frequencies will lead to smaller errors. It can be assumed that system designers have optimised this, within the boundary conditions of the design. Once this is determined, the RMS value of deformation is directly proportional to the RMS value of the acceleration of the frame. Hence, the following statement can be made:

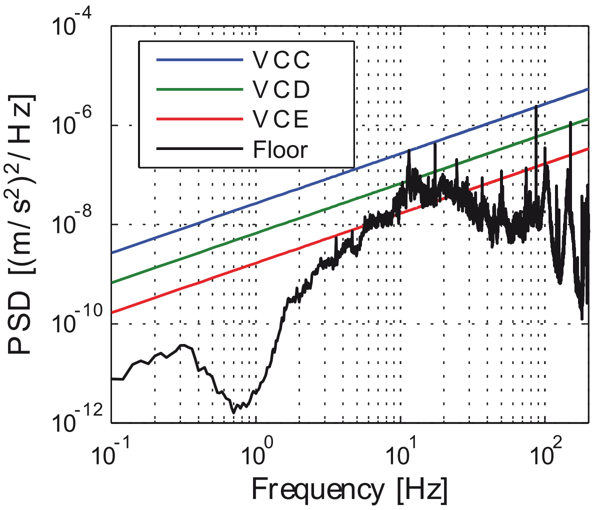


Figure 3. Power Spectral Density presentation of BBN-curves in acceleration units, combined with an actual PSD of measured floor vibrations.

The ultimate performance indicator for an isolated system is the RMS level of acceleration of that system.

Note 1. Of course, the RMS level is depending upon the level of floor vibrations present at a certain location. And comparison of systems must be done using the same external vibration input.

Note 2. This simple derivation is only valid for frequencies below the internal resonance frequency. In high-performance systems such frequencies are higher than 100 Hz. Specifications of floor vibrations are typically determined for frequencies between 1 and 80 Hz, so in many cases it will be acceptable to use this simplification. When interference of internal resonances with vibrations from the environment occurs, more detailed application-specific analysis will be required.

Specifications of floor vibrations

The floor specifications based upon the standards proposed by Bolt, Beranek and Newman (BBN) have been used for many years now. They specified curves with different vibration levels suited for different environments. The curves are indicated as BBN-A to BBN-E, or VC-A to VC-E (VC stands for Vibration Criteria) [3]. The last one is the quietest environment suited for the most sensitive tools. The vibration level is expressed in velocity and the RMS value for 1/3-octave bands is used to assess the vibration level.

In the design of a precision system the integral influence of vibrations can be predicted using the Dynamic Error Budgeting procedure; see the previous article [4], and [5]. Since floor vibrations are of a stochastic nature, the BBN curves can be converted to Power Spectral Density functions, PSDs. As instruments are sensitive to accelerations, using acceleration as the SI-unit is

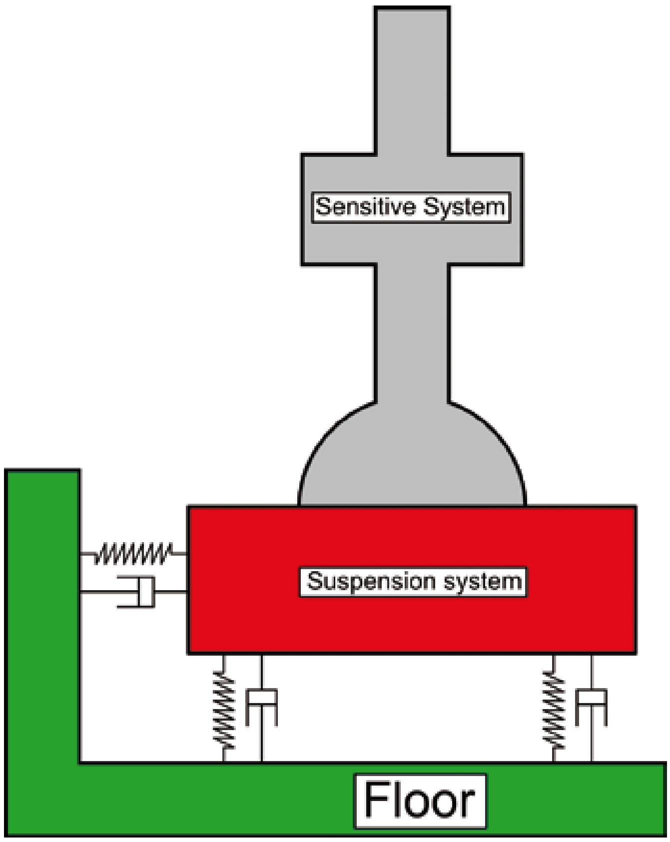


Figure 4. Passive vibration isolation using low-stiffness springs.

recommended. Figure 3 shows such curves together with an actual PSD of measured floor vibrations.

Based upon these graphs, it may be decided that the measured floor can be characterised as a VC-D floor. So, during design the VC-D disturbance level, adding up to about 6 mm/s^2 RMS for frequencies from 1 to 100 Hz, will be used for performance prediction. The actual level of accelerations in the measured floor, however, is adding up to only 0.8 mm/s^2. So, during design the influence of floor vibrations will be overestimated by a factor of 8. Therefore, another method of classifying floor vibrations is proposed. For a site, the PSD of accelerations, which indicates the specific frequency content, and the accumulated RMS level of acceleration for frequencies from 1 to 100 Hz must be provided. Then, a fair prediction of the impact of floor vibrations can be made.

Passive vibration isolation

The simplest suspension solution for avoiding floor to machine vibration transmission is to use low-stiffness springs, which in combination with the machine mass creates a mechanical filter; see Figure 4.

Through increasing the relative damping, the magnification at the resonance frequency can be reduced, although more damping will lead to more transmission above the

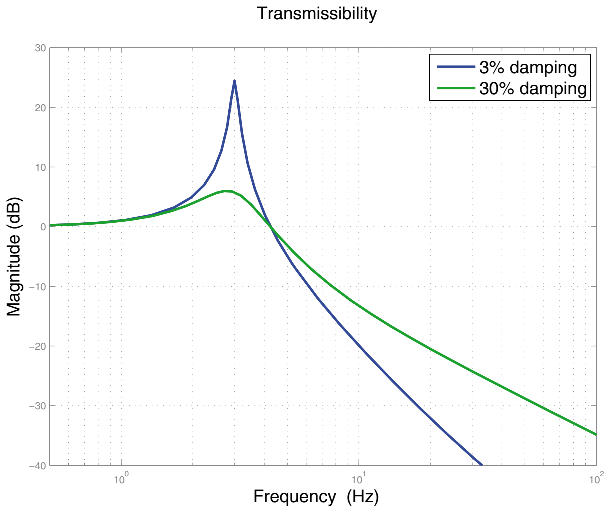


Figure 5. Transfer function describing floor to machine vibration transmission. Relative damping is 3% (blue line) and 30% (green).

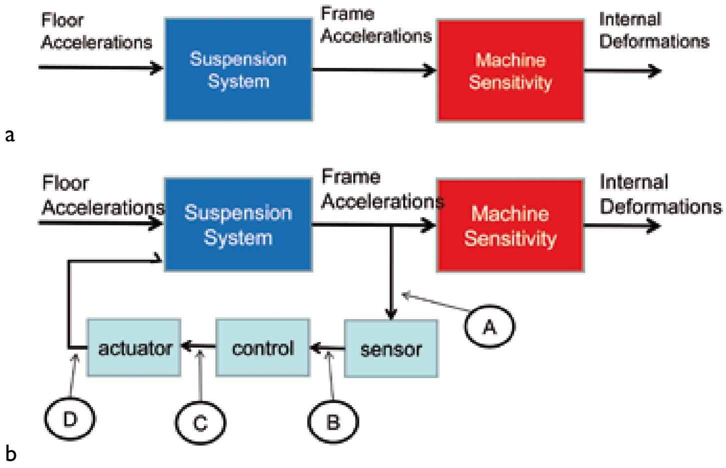


Figure 6. Principles of vibration isolation.
(a) Passive: floor vibrations are transmitted through the suspension; no additional disturbances are considered.
(b) Active: the frame motion is measured and the signal is used in the controller to create a counteracting force; noise present in the control loop causes additional disturbances (A-D).

resonance frequency. Figure 5 shows the effect of damping for a suspension having a resonance frequency of 3 Hz.

At different frequencies, this passive vibration isolation exhibits a different performance. When we consider the RMS value of acceleration in 1/3-octave bands around specific frequencies we can see the effect. Table 1 presents the RMS vibration of a VC-E specified floor and the resulting RMS level of machine vibration for two suspension solutions. Performance is mainly poor at low frequency. Therefore, active damping of the low-frequency suspension modes is required.
Another setback of passive vibration isolation using low-stiffness springs is the large static deflection at low frequencies due to gravity. This puts serious design restrictions on a passive system and once again calls for active vibration isolation.

Table 1. RMS vibration (μm/s^2) of a VC-E specified floor at three frequencies and the resulting RMS level of machine vibration (μm/s^2) for two suspension solutions.

	3 Hz	20 Hz	100 Hz
VC-E floor	60	360	1,800
3% damping	600	3.6	5
30% damping	120	36	30

Active vibration isolation

Active damping, or in general active vibration isolation, means incorporating a loop with sensor, control and actuator into the vibration isolation system. The principle, in comparison to passive vibration isolation, is shown in Figure 6.

The noise introduced in the control loop by the various electromechanical elements affects the actual accelerations and deformations. Now the “clean” transfer function from floor to frame accelerations does not completely cover the system performance. Especially for high-precision systems, placed in rather quiet locations, this becomes important. It may even turn out that adding active vibration isolation does not improve system performance as compared to the passive case; see the box on noise generation in active suspensions. It can be concluded that better sensors (and actuators) have to be available to obtain significant improvements.

Sensor and actuator

For active vibration control, MI-Partners developed dedicated sensors and actuators in collaboration with Magnetic Innovations [7]. Often geophones are used as sensors in active vibration control as a cost-efficient solution. A geophone comprises a permanent magnet that is suspended to the geophone casing; the magnet can move within a coil that is also fixed to the casing. The relative motion between magnet and coil induces a voltage proportional to the relative velocity. Hence, a geophone acts as a velocity sensor and its signal can be used as input for velocity-based feedback. It can be shown that the damping obtained is proportional to the absolute payload velocity; this is called sky-hook damping. Figure 8a shows a schematic of a geophone.
As a result of the joint development, geophones have become available that display excellent small signal behaviour down to frequencies as low as 1 Hz. For actuation a moving magnet actuator is used, wich has a very low parasitic stiffness; Figure 8b shows a schematic set-up.

Noise generation in active suspensions

Generally the vendors of isolation systems are specifying performance through the transfer function for floor to payload vibrations. To measure such curves, excitation of the floors or internal excitation with the actuators is used. Then the signal levels are large enough to obtain clear curves showing large vibration suppression at specific frequencies.
For the actual system no external excitation is present and signals are generally very small. In such cases the noise from sensors and actuators may well become dominant. Figure 7 shows the PSD of acceleration of an isolated platform placed on the floor presented above, both for a 3 Hz and a 1 Hz passive system.

For the system on the 3 Hz suspension, the RMS value for acceleration is predicted to be about 0.5 mm/s^2, only a slight improvement compared to the level in the floor. For the 1 Hz system this RMS value reduces to about 0.02 mm/s^2, a significant improvement.
For active systems, noise generated in different elements will lead to forces exerted on the platform. Such forces will lead to accelerations that must be added to the acceleration due to floor vibrations. It has been indicated [6] that the noise of the sensors used for active suspension systems is between 0.05 mm/s^2 and 0.1 mm/s^2 RMS. As these levels are above the level obtained with a 1 Hz passive system, active systems would not improve matters. For the 3 Hz system an integral improvement of a factor of 5 to 10 would be the limit. Unless better sensors turn up.

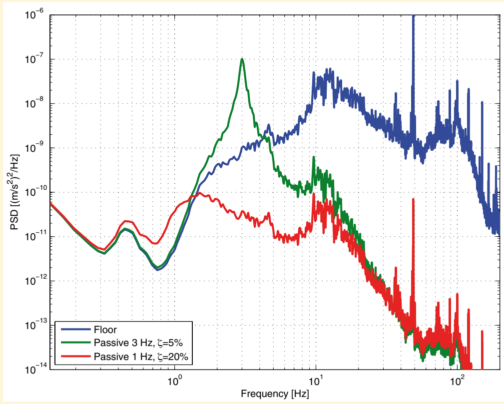


Figure 7. PSD for acceleration of a passively supported platform.

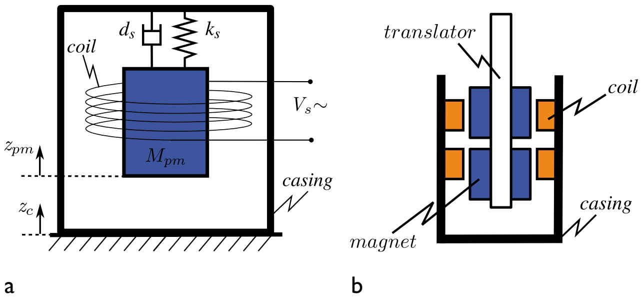


Figure 8. Schematics of sensor and actuator.
(a) Geophone.
(b) Moving magnet actuator.

In practice, the sensor and actuator are integrated in one module (collocation), so mutual inductance can occur. If this effect is large enough it can cause stability problems in the control loop. In the present case these problems were ruled out, but with very large payloads sensor-actuator crosstalk may arise.
To achieve vibration isolation for all six degrees of freedom (DoFs), a total of six sensor-actuator modules is required, which in a practical arrangement are divided into three horizontal and three vertical units.

Elimination of horizontal-tilt crosstalk

The drawback of geophones lays in the fact that a horizontal geophone measures displacements caused by tilt of the payload as horizontal displacements, introducing measurement errors and the need for complex control actions to compensate. Independently, two parties (TNO [1] and MI-Partners) each developed a patent-pending solution to eleminate crosstalk and hence the need for control complexity [8]. Figure 9 shows a schematic of MI-Partners' solution. It encompasses a mechanical guide which constrains tilting motions of the geophone by connecting these to the fixed world using elastic elements; while the Stinger transfers the horizontal motion of the payload to the geophone assembly.

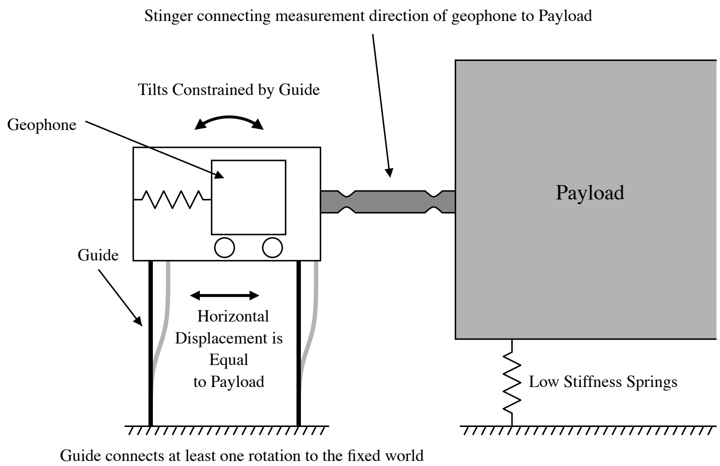
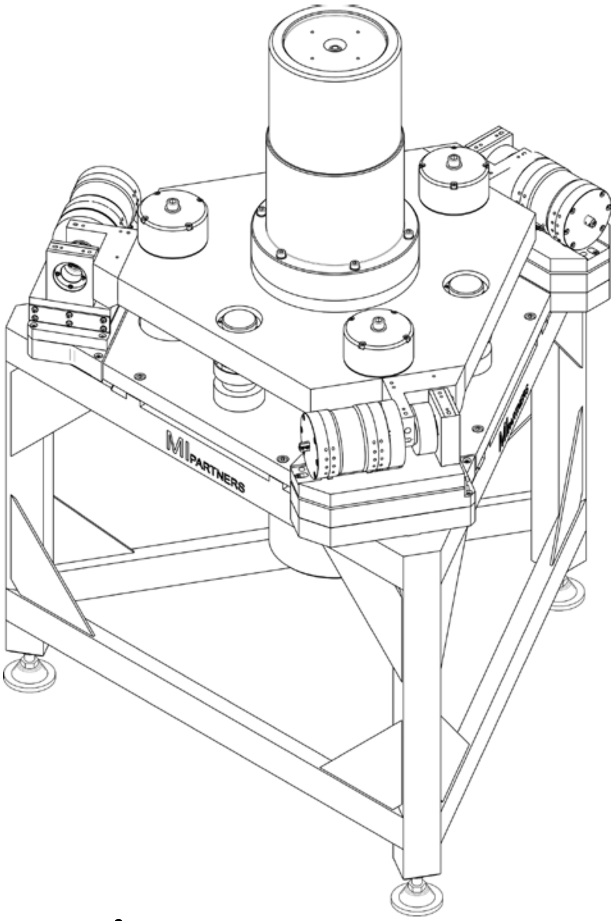


Figure 9. The guide stiffness allows a horizontal motion of the geophone, while constraining tilting motions. In the measurement direction the geophone is stiffly connected to the payload.



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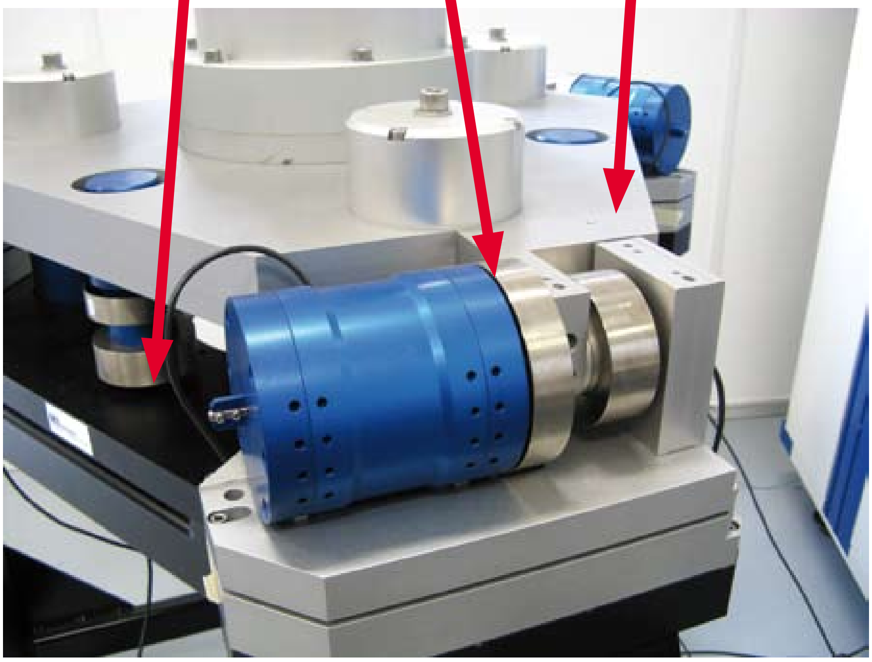

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Figure 10. The MI-Partners Active Vibration Isolation system.
(a) Schematic of the design, showing the payload, the frame, the six sensor/actuator modules (not all visible) and three isolator mounts.
(b) Close-up showing the positioning of horizontal and vertical geophones (the blue elements).

Design

Based on the above considerations, an Active Vibration Isolation system was designed; Figure 10 shows the mechanical design.

From the DEB analysis it was concluded that the signal-to-noise ratio of the geophone signal to the noise level of the AD-convertor that generates the control input was insufficient. This limits the effective bandwidth for the active control and hence the performance of the system. Several solutions were considered and the most cost-effective one turned out to be incorporating a preamplifier in the loop, between the geophone and the AD-converter.

Assessment of the "ultimate performance"

In the final analysis all contributions were considered, including floor, sensor, preamplifier and AD-converter. Figure 11a shows the respective PSDs.

A closed-loop vibration suppression in 6 DoFs has been achieved between 0.3 Hz and 30 Hz, with > 40 dB reduction at 3 Hz, see Figure 11b. The measured cumulative noise level in the frequency range of 0..100 Hz in horizontal direction is 37 $\mu m/s^2$ RMS, while in the vertical direction 24 $\mu m/s^2$ RMS has been achieved. Floor vibrations with a cumulative content of 1.2 mm/s^2 RMS (in the order of magnitude of a VC-D floor) in the frequency range of 0..100 Hz can be reduced with a factor 50 in this frequency band.

Conclusion

An active vibration isolation system (AVI) with a high bandwidth and low-noise components was developed. Using DEB (Dynamic Error Budgeting) methodology all components of the active control loop were analysed in terms of PSDs (Power Spectral Densities). Based on the specifications derived from this analysis, improved sensor

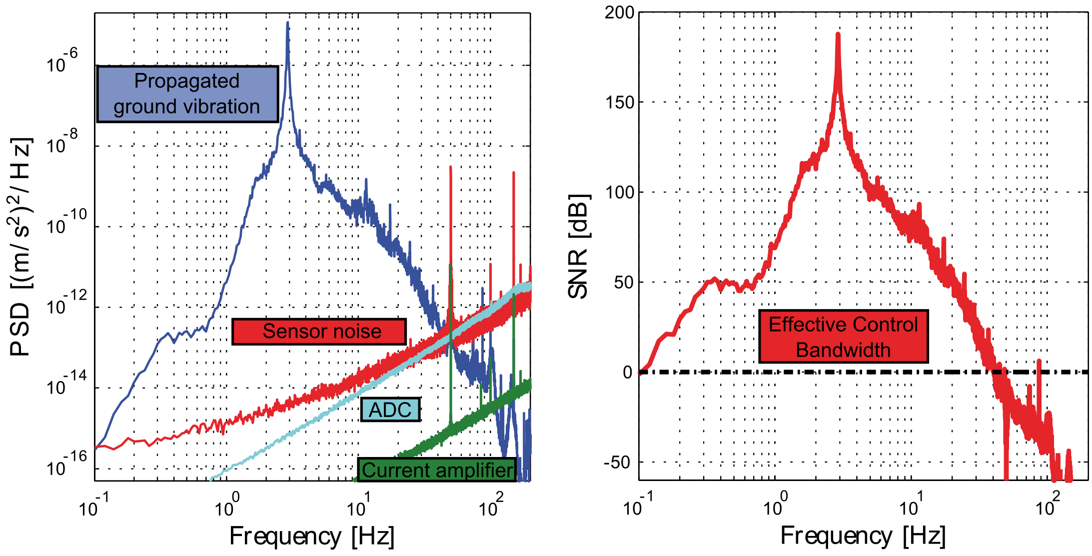

Figure 11. PSDs of all contributions to system performance and resulting signal-to-noise ratio in vertical direction.

Vibrations induced by driving forces

In many cases, active systems yield only limited improvement of floor vibration isolation. For precision systems with mechanical resonances at about 100 Hz one can estimate the position errors due to the accelerations. The 0.5 mm/s^2 RMS for the 3 Hz passive suspension (see the previous box) will lead to deformations of about 1.5 nm. With the 1 Hz suspension only 70 pm will result. These levels are quite low for most applications and hence the passive systems are quite sufficient in most cases and active systems will not be needed.

Still, in the semiconductor industry active systems are widely used in, for example, advanced lithography systems. To understand why, we have to take into account the acceleration caused by the driving forces for the stages. Accelerations of 10 m/s^2 and more are quite normal. After settling, the suspended frame, which typically has a 50 times larger mass, will display an acceleration of 200 mm/s^2. This acceleration is 400 times larger than the acceleration due to floor vibrations. With such acceleration, errors of 600 nm would occur. To eliminate this effect, direct cancellation of the stage forces can be applied. Therefore, an active element is introduced in addition to the passive vibration isolation system. This actuator can also be used to generate damping for the isolation system and hence the total system has all the characteristics of an active vibration isolation system. The main function in this case, however, is the reduction of drive force induced vibrations. An additional advantage of active over passive control is much improved settling behaviour; in the passive case the settling transient is dominated by the (low) suspension frequency.

technology with extremely well small-signal-behaviour was developed and integrated in the new high-end vibration isolation platform. As a result, a platform performance below 40 µm/s^2 was achieved on a standard production floor.

In this article, the focus was on reducing the transmission of floor vibrations. However, internal disturbances can have a large impact on system performance as well. Here active vibration control can also help; see the box on vibrations induced by driving forces.

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