The Mystery of Active Vibration Isolation

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Abstract

Precision Systems have always been threatened by disturbances originating from vibrations. Active Vibration Isolation Systems have promised significant improvements of precision. Broad application however has not yet occurred. Actual benefits appear not sufficient to accept the added cost. Many aspects contribute to this situation. Three of these aspects will be presented to explain the "mysteries" still surrounding Active Vibration Isolation. The following elements will be discussed;

1. Specification of floor induced vibrations
2. Noise Generation in active systems
3. Vibrations induced by driving forces

1 Specifications of Floors.

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

In the design of a precision system the integral influence of vibrations can be predicted using the Dynamic Error Budgeting procedure, Lit 2. Since floor vibration are of stochastic nature the BBN curves can be converted to Power Spectral Density functions, PSD. Using the acceleration as the SI-unit is recommended. Figure 1 shows such curves together with an actual PSD of measured floor-vibrations.

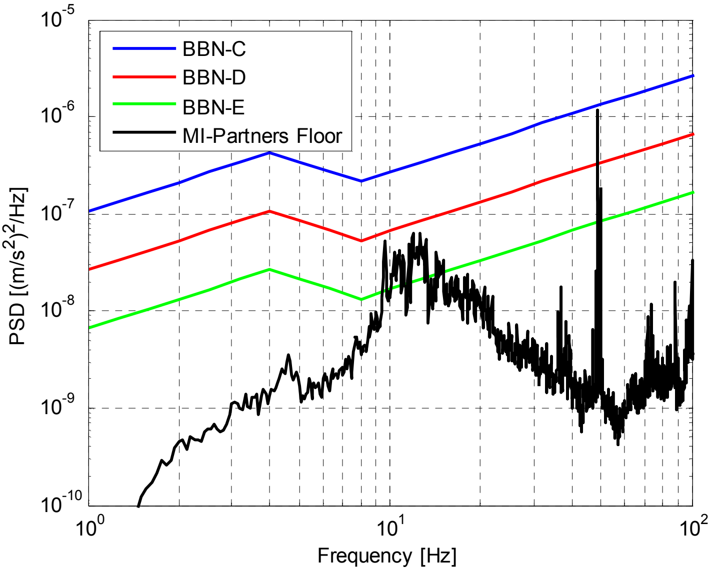

Figure 1. Power Spectral Density presentation of BBN-curves in acceleration units, combined with an actual PSD of measured floor vibrations.

Based upon these graphs it may be decided that the measured floor complies as a "BBN-D" floor. During the design this disturbance level, adding up to about 6 mm/s^2 RMS for frequencies from 0 to 100 Hz, will be used for the prediction of performance. The actual level of accelerations in the measured floor are however adding up to only 0.8 mm/s^2 and during design the influence of floor vibrations will be overestimated by a factor of 8!

Therefore another method of classification of floor vibrations is proposed. For a site the PSD in accelerations, which indicates the specific frequency content, and the accumulated RMS level of acceleration for frequencies from 0 to 100 Hz must be provided. Thus a fair prediction of the impact of floor vibrations can be made.

2. Noise Generation in Active Suspensions.

Generally the vendors of Isolation Systems are specifying the performance of the system using the transfer function from floor vibrations to pay-load vibrations. To measure such curves excitation of the floors or internal excitation with the actuators is used. Thus the signal levels are large enough to obtain clear curves showing large suppression of vibration 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. In figure 2 the PSD of accelerations on isolated platforms placed on the floor presented above are shown. Both the PSD for a 3 Hz and a 1 Hz passive system is shown.

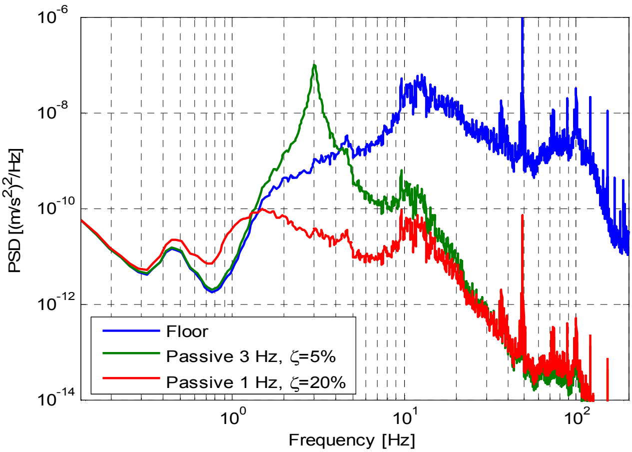

Figure 2. PSD function for acceleration on passively supported platforms.

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 from 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. In lit. 1 van der Poel has indicated that the noise of the sensors used for active suspension systems are 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 will not improve the 1 Hz system. For the 3 Hz system an integral improvement of a factor of 5 to 10 will be the limit. Better sensors must be developed to obtain significant improvements.

3. Vibrations induced by driving forces.

Active systems yield only limited improvements. 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 will lead to deformations of about 1.5 nanometer! With the 1 Hz suspension only 70 picometer will result. These levels are quite low for most applications and thus the passive systems are quite sufficient in most cases and active systems would not be needed.

Still active systems are widely used in the 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 above are quite normal. After settling the suspended frame, typically 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 thus the total system has all the characteristics of an “Active Vibration Isolation” system. The main function however is the reduction of drive force induced vibrations.

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