Suppression of Micro Vibration for Steel Frames using a Proof Mass Actuator

Author: ***Dr. Uwe Stöbener, W.A.V.E. - World of Anti Vibration Engineering, www.w-ave.de***
Eebereschenring 45, 34346 Hann.Münden, Germany

Vibration sensitive equipment, e.g. Scanning Electron Microscopes (SEM) or Nano Lithography Systems, has to be isolated from the noisy environment. Active vibration isolation platforms guarantee a very small remaining vibration amplitude on their isolated top plate. These platforms can be positioned directly on the concrete floor but sometimes they have to be installed on a support frame. Due to the compensation forces of the active isolation platform the support frame has to be as rigid as possible. Welded steel frames are best suited for this issue but even these frames exhibit resonances. The resonance behavior reduces the isolation performance of the platform and results to an increased remaining vibration amplitude. In this paper active vibration dampers are introduced to minimize the vibration of rigid body modes of a steel frame.

Introduction

Modern instruments for Scanning Probe Microscopy (SPM) and other analytical systems provide topography plots and other physical information with a resolution of several nanometers or even less than one nanometer. For this reason these instruments are extremely sensitive against mechanical vibration and the microscopy manufacturers are specifying the maximum allowed vibration limit for their systems. The specified limit is given in terms of acceleration, velocity or displacement. Typical values are 10 to 100 µg rms, 1 to 10 µm/s rms or 10 to 100 nanometer rms. This type of very small vibration is called "micro vibration". A classification of micro vibration is given in Ref.1 and 2.

In most buildings these limits are not fulfilled and vibration isolation systems have to be used. For most sophisticated isolation performance active vibration isolation units are best suited. These systems have been described in Ref.3. They are equipped with vibration sensors, control electronics and actuators. The sensors are detecting the current vibration and the control electronics are generating a control signal to drive the actuators. Finally the actuators are producing forces to counteract the movement of the isolated device. Depending on the size and weight of the isolated instrument the active vibration isolation system is positioned on a table or directly on the floor. Midsize and large instruments, e.g. Scanning Electron Microscopes (SEM) or Scanning Tunneling Microscopes (STM), are commonly installed directly on the concrete floor. In some cases, like clean rooms with double floor, it is necessary to put the active vibration isolation platform and the large instrument on top of a steel frame. A configuration with steel frame, active vibration isolation platform and SEM is depicted in Fig.1.

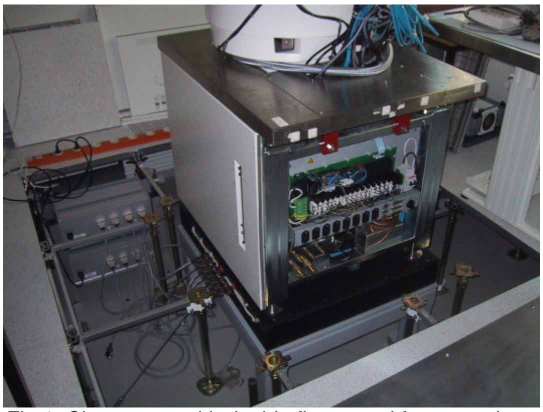

Fig.1: Clean room with double floor, steel frame, active vibration isolation platform and Scanning Electron Microscope

In order to adapt the height of the steel frame to the level of the second floor the frame is equipped with height adjustable feet. The feet are consisting of a threaded rod and a circular base (see Fig.2)

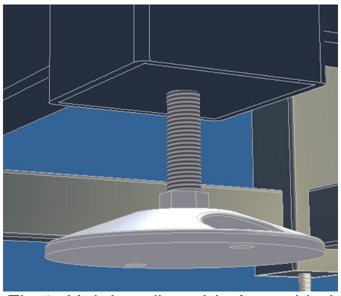

Fig.2: Height adjustable feet with threaded rod and circular base

The feet have an important influence to the structural dynamics of the steel frame. The rods of the feet are acting like elastic springs and the steel frame is moving in kind of rigid body modes (see Fig.3). The main translation motion occurs in the two horizontal directions and rotation can be observered with respect to the z-axis. Depending on the size and weight of the frame the rigid body modes occur between 40Hz and 150Hz.

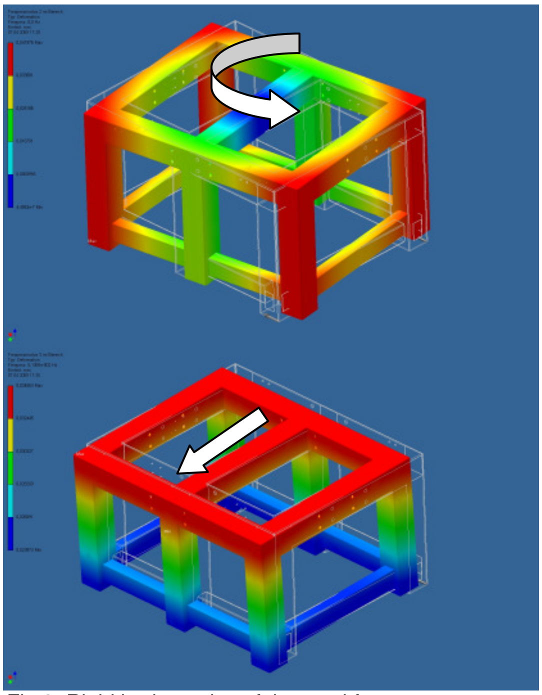

Fig.3: Rigid body modes of the steel frame

As already mentioned, the active vibration isolation platform on top of the steel frame operates with compensation forces. These compensation forces have to be balanced by reaction forces according to Newton´s third axiom (action equals reaction). As a result of the reaction forces the rigid body modes of the steel frame will be excited and the isolation performance of the active vibration isolation platform decreases.

Since the described dynamic behavior is related to rigid body modes damping treatments, like viscous coatings or dampers, would not solve the problem. Much better suited are proof mass actuators. These actuator are using a force of inertia to counteract the motion of the steel frame.

Active Damper / Proof Mass Actuator

Configuration

The proof mass actuator is part of an active damper encapsulated in an aluminum housing. The force generated by the proof mass actuator is coaxial to the housing. In Fig.4 a configuration of six active dampers attached to the steel frame is depicted. The arrows are indicating the directions of the actuator forces. It is obvious that this configuration enable the motion control of the two horizontal translation modes and the rotation with respect to the z-axis. Nevertheless, each active damper operates as standalone unit using its own control circuit.

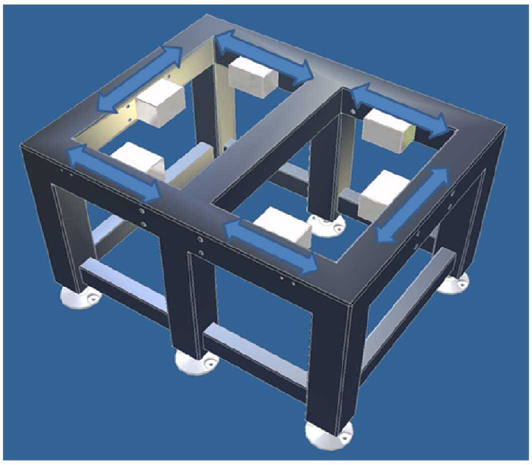

Fig.4: Steel frame with six active dampers

Components of the active damper

The main components of the active damper are:

(1) aluminum housing (not shown in Fig.5)

(2) chassis

(3) ferro magnet

(4) voice coil

(5) suspension of the ferro magnet

(6) accelerometer (inside chassis)

(7) analog control circuit

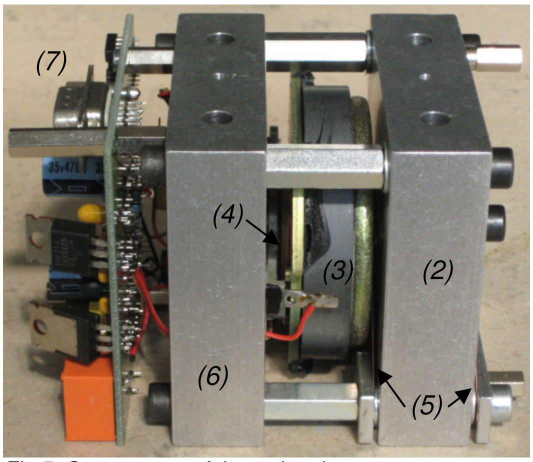

Fig.5: Components of the active damper

The chassis consists of two aluminum blocks connected by four screws. These blocks and the housing are fixed to the steel frame. A ferro magnet and a voice coil are positioned between the two blocks. The ferro magnet is elastically suspended to the block on the right hand side by two copper sheet metals in order to get a parallel motion. The voice coil is fixed to the block on the left hand side. The ferro magnet, the suspension and the voice coil together are representing the proof mass actuator. The sensor is integrated in the block on the left hand side and the PCB with the analog control circuit is attached to the same aluminum block.

Sensor

The sensor is designed with a piezoelectric disc *(1)*, a brass cylinder *(2)*, a plastic ball *(3)* and a copper sheet metal *(4)*.

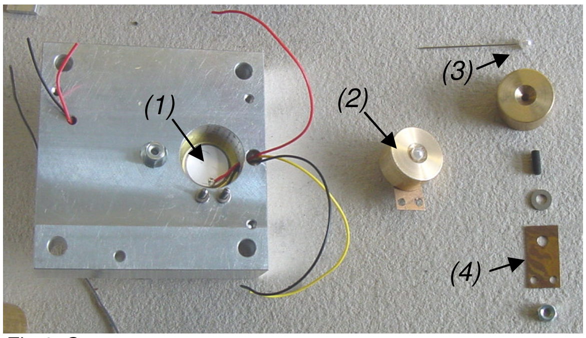

Fig.6: Sensor components

The piezoelectric disc is sitting on an o-seal at the bottom of a drill hole in the aluminum block. The brass cylinder is resting on the piezoelectric disc and the plastic ball is used to guarantee a pure normal contact between the brass cylinder and the piezoelectric disc. The copper sheet metal centers the brass cylinder in the drill hole and generates a pre-stress for the contact between the cylinder and the disc.

By moving the aluminum block, or the complete active damper, the brass cylinder generates a force of inertia normal to the disc which is proportional to its absolute acceleration. Due to the force the piezoelectric disc is getting bended and an electric charge is produced.

In Fig.7 the Frequency Response Function (FRF) of the sensor is shown. Depending on the pre-stress the sensor resonance occurs between 350Hz and 450Hz. A stronger pre-stress shifts the resonance to higher frequencies. To achieve control stability for the active damper the sensor should be used within the linear range of the FRF only. This range is marked with the blue area in the diagrams below and it corresponds to the frequency range of the rigid body modes of the steel frame.

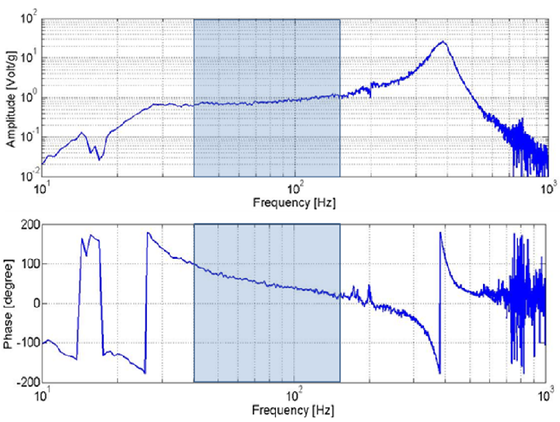

Fig.7: Frequency Response Function of the sensor

Proof Mass Actuator

The proof mass actuator consists of a ferro magnet *(1)*, a voice coil, two U-form copper sheet metals *(2)* and an aluminum cylinder *(3)*.

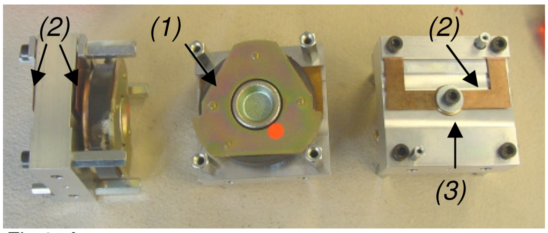

Fig.8: Actuator components

The ferro magnet is used as a mass of inertia. It is elastically suspended by two U-form copper sheet metals. The U-form sheet metals are fixed to the chassis at their left and right arms. At their centers they are connected to each other by an aluminum cylinder which is positioned in a drill hole of the chassis. This setup guarantees a parallel motion of the ferro magnet free of backlash. The maximum stroke of 3mm is limited by the depth of the circular gap inside the ferro magnet and the height of the voice coil.

In Ref.4 it has been shown that the dynamics of the proof mass actuator have to be considered to achieve control stability. Using absolute velocity feedback, which is also known as skyhook control, stability is achieved if the eigenfrequency of the actuator is smaller than the eigenfrequency of the structure to be controlled. In Fig.9 it can be seen that the actuator eigenfrequency is approximately 20Hz. Since the rigid body modes of the steel frame are related to frequencies between 40 and 150Hz the stability condition has been satisfied.

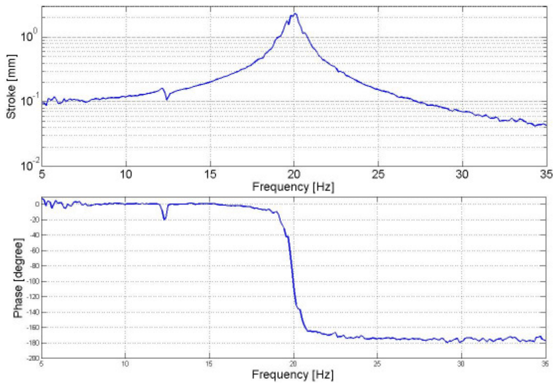

Fig.9: Frequency Response Function of the actuator

As described in Ref.4 the maximum attenuation using a proof mass actuator is:

$$Attn(dB) = -20\log_{10}\left(\frac{\varsigma_m \omega_a}{\varsigma_m \omega_a + \varsigma_a \omega_m}\right),$$

with ζ_m , ω_m as damping and circular eigenfrequency of the steel frame and ζ_a , ω_a as damping and circular eigenfrequency of the actuator.

In order to get a high maximum attenuation for a given value of ζ_m and ω_m , it is clear that the actuator should be as well damped as possible, so that ζ_a is large, and ω_a / ω_m must be as small as possible, i.e., the natural frequency of the actuator must be well below that of the mount.

Analog Control Circuit

The analog control circuit has been designed in kind of a single Printed Circuit Board (PCB). The PCB contains the signal conditioning of the sensor signal and the Pulse Width Modulated (PWM) power amplifier to drive the actuator.

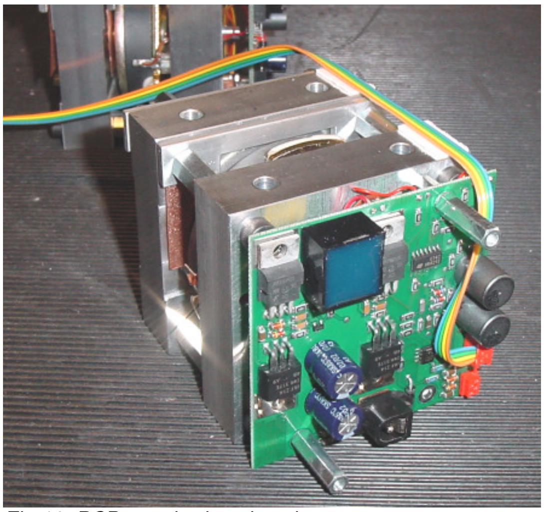

Fig.10: PCB attached to chassis

In Fig.11 the FRF of the control circuit is depicted. The first blue line indicates the eigenfrequency of the actuator and the second blue line indicates the eigenfrequency of the sensor. Between those two lines the blue colored area shows the frequency range of the rigid body modes of the steel frame which shall be controlled by the active damper. Within this area the phase angle of the control circuit is -90°. Therefore the absolute acceleration signal generated by the sensor has been converted into a signal proportional to the absolute velocity. This converted signal is used to drive the actuator. The resulting feedback loop represents a typical skyhook controller as described in Ref.5, Ref.6 and Ref.7.

The amplitude diagram of the FRF in Fig.11 shows a decreasing gain of -20db per decade within the blue colored area. This attenuation is required to compensate the rising actuator force at higher frequencies. For a constant stroke the actuator force grows quadratically with increasing frequency.

The notch filter at the second blue line is used to suppress the resonance behavior of the sensor.

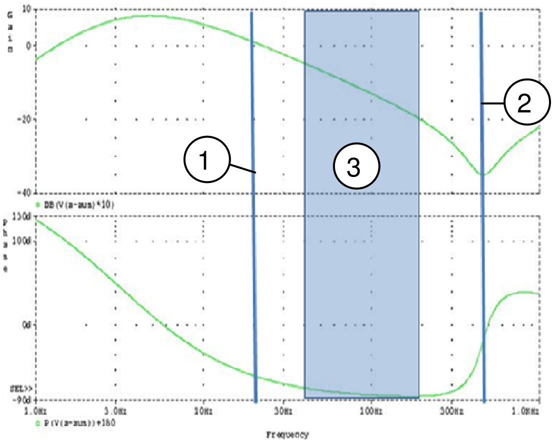

Fig.11: Frequency Resonse Function of the control circuit

Experimental Test

In order to evaluate the performance of the proposed active damper different tests have been carried out.

The first test was made by attaching an electrodynamic shaker to the steel frame. The excitation point was selected on top of the frame at one corner and the direction was 60° out of angle with respect to the x-axis (see Fig.12). White noise was used for excitation.

The response was measured by using a piezoelectric accelerometer positioned on the opposite corner of the frame. Time integration was applied to the response signal in order to get the displacement function of the frame.

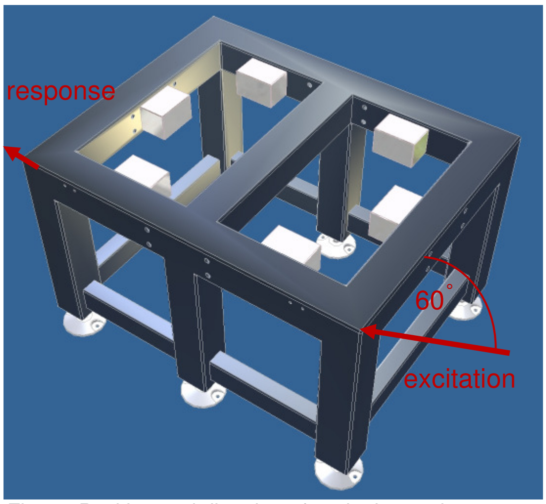

Fig.12: Position and direction of excitation and response

The resulting compliance is depicted in Fig.13. Without active damping three resonances clearly occur in the diagram. The first resonance at 47Hz is related to the rotation of the frame. The second resonance at 64Hz corresponds to the motion in x direction and at 85Hz the rigid body mode in y direction takes place. By turning on the active dampers all resonances have been suppressed. Especially the compliance with respect to the rotational degree of freedom has been reduced from 15µm/N to 0.3µm/N.

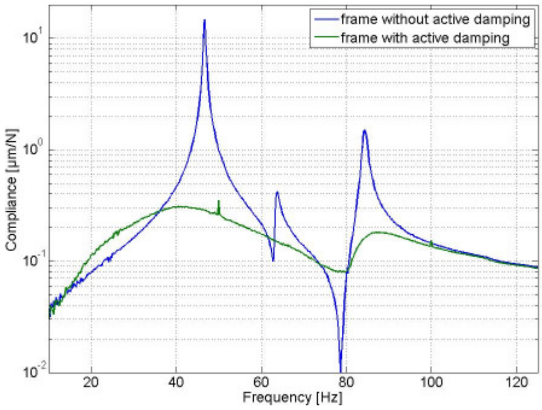

Fig.13: Measured compliance

For a second test the shaker was replaced by an implus hammer and the response was measured within the time domain.

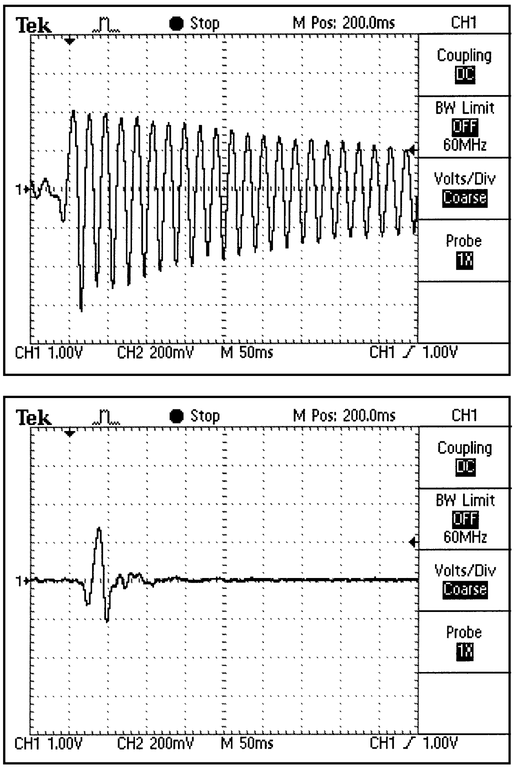

Fig.14: Measured settling time, with (diagram below) and without active dampers (diagram above)

The comparison of both plots in Fig.14 demonstrates the different dynamics with and without active damping. Without active damping it takes more than half a second till the frame comes to rest again. With active dampers the vibration dies out within 50ms.

Summary and Outlook

Active dampers have been used to suppress the rigid body modes of a steel frame. The dampers have been designed to operate in a frequency range between 40 and 150Hz. Their main components are the proof mass actuator, the sensor and the analog control PCB. Certain conditions have to be fulfilled to guarantee stability for the applied skyhook control strategy. Most important is the relation of the eigenfrequencies of the actuator and the frame. The eigenfrequency of the actuator most be below the lowest eigenfrequency of the frame.

The discussed application is just one example to demonstrate the use of active dampers. However, the proposed technology could be applied to numerous different fields, e.g. automotive applications or civil structures.

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