

DYNAMIC MODELING AND VIBRATION ANALYSIS OF A UHV SCANNING TUNNELING MICROSCOPE

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Abstract

Techniques based on scanning probe microscopy (SPM) are used to fabricate surface structures with dimensions ranging from 10 -100 nm. These structures have been fabricated and imaged using a scanning tunneling microscope (STM). The STM requires the tip-sample distance to be controlled with picometer precision. This requirement must be satisfied in the presence of external vibrations with micrometer amplitude, temperature drifts and acoustic perturbations. In this manuscript, the results of dynamic modeling and vibration analysis of a STM that is currently being built and tested at the National Institute of Standards and Technologies (NIST) are presented. The design of an eddy current damping system to isolate the STM unit from environmental vibrations is also discussed.

Keywords: STM, modal analysis, eddy current damping

Introduction

The scanning tunneling microscope (STM) has revolutionized the study of surfaces and nanometer size objects. The STM's principle of operation in a constant current mode is based on maintaining the tunneling current in the gap between a conductor/semiconductor sample surface and a sharp metallic tip, when they are brought in close proximity (< 1nm) to each other. By controlling the current with an electronic feedback circuit while the sample surface is scanned by means of piezoelectric elements, the topography of the surface and/or some electronic properties are obtained [1]. Atomic scale resolution with the STM requires the detection of structures to be on the order of 10 pm - 100 pm [2]. This is accomplished by designing STM's with high mechanical stability and isolating the STM unit from external perturbations. The general form of an equation for tunneling current is an exponential function of tip-sample separation, d , and is given by Eq. (1). This exponential dependence of the current on the tip-sample distance makes vibration isolation a critical part in the STM design.

$$I \propto \left(\frac{V}{d}\right) e^{(-AF^{1/2}d)} \quad (1)$$

This article discusses some experiments conducted to determine the low frequency vibration in the STM structure and methods of isolating them. Results of finite element analyses performed on the STM structure to determine the modes that involve the motion of the tip relative to the sample, which influence the tunneling current and imaging capability are presented. The design of an eddy current damping system to minimize external perturbations is also presented. The complete design and working principle of the STM is explained in detail by Gilsinn et.al [3]. The STM system consists of a PZT tube to drive the STM tip in two directions. The fast scan direction of the STM is replaced with a PZT driven flexure stage, and an interferometer system is used to monitor this axis of motion. The system also incorporates a separate millimeter-scale motion system for navigating and imaging large-scale samples.

General Consideration of Vibration Isolation

External perturbations coming from the building floor have different frequencies, vibration modes, and amplitudes, and these change with time. The acceleration magnitude of the laboratory floor as a function of frequency is measured with a seismic accelerometer (sensitivity of 10 V/g) and a built-in low-pass filter of 100 Hz. The vibration displacement

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magnitude, a , is then computed as a function of the acceleration magnitude, M , and the oscillating frequency, f , with Eq (2).

$$a = \frac{M}{(2\pi f)^2} \quad (2)$$

The magnitude spectra of the acceleration measured on the laboratory floor and the STM frame is shown in Figure 1. The maximum acceleration magnitude on the laboratory floor is $50 \times 10^{-6} \text{ g}^\ddagger$ at 16 Hz^\S , corresponding to amplitude of 50 nm , which is far greater than the atomic scale. The level of the vibration amplitude at the tunneling assembly should be on the picometer scale, and this is achieved by mounting the entire vacuum chamber on a concrete slab supported on air springs. As a result of this arrangement, the acceleration magnitude as measured on the instrument's supporting frame is only $2.5 \times 10^{-8} \text{ g}^\ddagger$, at 68 Hz^\S which corresponds to a displacement amplitude of approximately 71 pm . This represents a reasonable amount of excitation to the STM, and further isolation and damping is necessary to reduce this amplitude to only a few picometers.

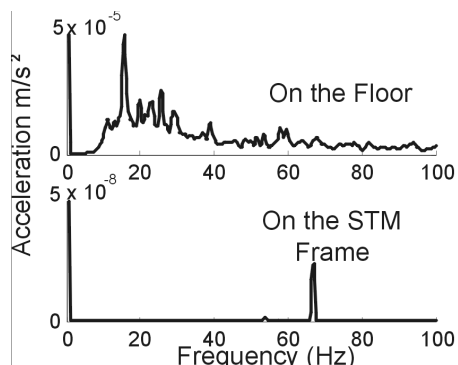


Figure 1: Acceleration Magnitude on the Lab Floor and the STM Frame

Modal Analysis of the STM Structure

The STM structure is an elastic system and can be characterized by its modes of mechanical excitation and their resonance. Pohl recommends that resonant frequencies in the direction of the tip-sample gap exceed 1 kHz [4]. Estimates of the STM's structural resonant frequencies and mode shapes were calculated by performing a finite element modal analysis. These FEA results are also useful when conducting experimental studies by exciting the STM's structure with an impact hammer or vibrations from a mechanical shaker. Experimental studies were performed on the STM structure consisting of the top plate, bottom plate and four columns (as shown in

[‡] Uncertainty in amplitude measurement is $\pm 500 \text{ ng}$, $k = 2$

[§] Uncertainty in frequency measurement is $\pm 0.25 \text{ Hz}$, $k = 2$

Figure 2). This structure will enable to determine the frequency of the mode that causes bending of the columns which causes fluctuations in the tip-sample gap. Different modes of the STM assembly were detected by mounting the accelerometer at anti-nodes on the structure that were predicted by the FEA results. The structure is excited by an instrumented impact hammer. The theoretical and the experimental results are compared in Table 1, and the agreement is better than 12% for each of the first six modes. Both the FEA and experimental results demonstrate that the first resonant frequency is well above the 1 kHz recommendation of Pohl.

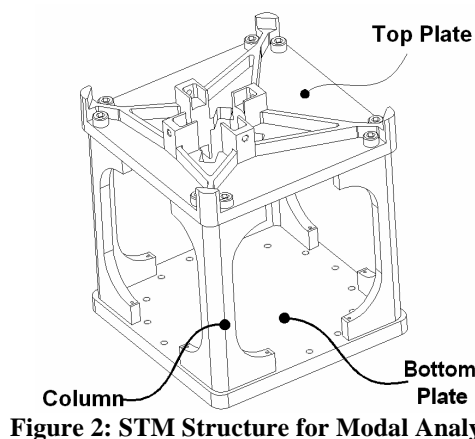


Figure 2: STM Structure for Modal Analysis

Table 1: Comparison of FEA and Experimental Modal Analysis Results

Modes	FEA (Hz)	Experimental ^{**} (Hz)	Difference (%)
1	1618	1456	10.01
2	1619	1460	9.82
3	1728	1796	3.93
4	1730	1804	4.27
5	2550	2670	4.70
6	2937	2860	2.62

Eddy Current Damping

The influence on the tip-to-sample distance from the external perturbations can be reduced by addition of vibration isolation mechanisms and by designing the STM structure to have sufficient stiffness. Spring suspension is widely used for scanning tunneling microscopes that operate in ultra high vacuum (UHV). Metal springs with low spring constants yield low resonant frequencies but have high quality factor (low internal damping) [5]. Hence additional damping elements such as magnetic eddy current damping are necessary to reduce amplitudes at

^{**} Uncertainty in frequency measurement is $\pm 8 \text{ Hz}$, $k = 2$

resonance. Though viscous fluid damping is used in many vibration isolation schemes, magnetic eddy current damping is better alternative for vacuum compatibility and the ability to vary the damping coefficient. Eddy currents are generated as a result of a conductor moving through a magnetic field. These currents dissipate energy as they flow through the resistance of the conductor. The resulting drag force on the conductor is proportional to its velocity relative to the field and thus functions as a viscous damping element. The eddy current damper operates at the base of the floating assembly as shown in Figure 3.

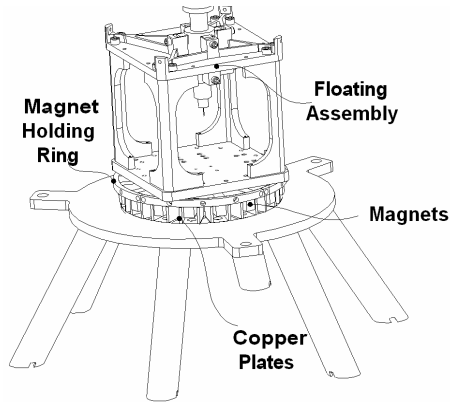


Figure 3: Eddy Current Damping Setup

The eddy current damper consists of a circular plate with rectangular cutouts for holding the permanent magnets and four rings are used for rigidly clamping the magnets in position. A circular plate holding the copper conductor is welded to the bottom flange of the vacuum chamber. The permanent magnets are arranged such that the opposite poles are facing each other. This kind of arrangement ensures that the magnetic flux lines are perpendicular to the direction of the moving conductor, which results in a more uniform damping effect. The field lines for this arrangement are simulated using commercially available electromagnetic field simulation software and illustrated in Figure 4.

The undamped natural frequency, f_n , of the floating module is a function of the equivalent stiffness of the extension spring, k , and the mass of the module, m , and is estimated with Eq.(3)

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (3)$$

The undamped natural frequency of the structure, with a spring stiffness of 33 N/m¹ and a mass of 1.817 kg (calculated from FEA) is 1.918 Hz. The

¹ Based on manufacturer's published specifications

critical viscous damping coefficient, c_c , for the system is expressed in terms of mass, m , of the system and the circular natural frequency, ω_n , and is given by Eq (4). It is calculated to be 4.5 Ns/m.

$$c_c = 2m\omega_n \quad (4)$$

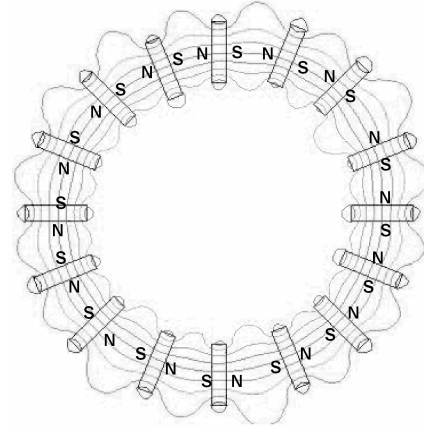


Figure 4: Magnetic Field Lines for the Magnet Configuration (without copper plates)

The damping coefficient, c (Ns/m), for an eddy current damper is a function of the magnetic flux density, B (Tesla), length of the conductor, L , in meters, cross-section area of the conductor, A , in m², and resistivity of the conductor material, r , in Ω m and is given by Eq.(5) [6].

$$c = K \frac{B^2 h A}{r} \left[\frac{1}{2 \left(1 + \frac{L}{h} \right)} \right] \quad (5)$$

K is a factor that takes into account losses from the damper configuration and other potential losses due to imperfections. It generally varies between 0.25 and 0.35 [7]. The damping coefficient for the current setup with an estimated flux density, $B = 1.19$ T and resistivity, $r = 16.8 \times 10^{-91}$ is calculated to be 0.15 Ns/m. The damping ratio ζ is expressed, as the ratio of c/c_c and for an individual set of magnets and plates is approximately 0.0343. Therefore the combined damping ratio for 16 such assemblies is approximately 0.55.

Displacement Amplitude Spectra of System

Experiments were conducted to measure the displacement spectra of the system. One accelerometer is mounted on the suspended structure and another is mounted on the frame. A dynamic signal analyzer records the frequency responses of the suspended structure and the reference frame due

to the impulse from the hammer. In general, the transfer function for spring suspended systems at frequencies much greater than the resonant frequency is proportional to $1/f^2$ without external damping. Since eddy current damping stiffens the coupling as a function of frequency, it decreases by $1/f$ yielding a system sensitivity increasing with frequency [8].

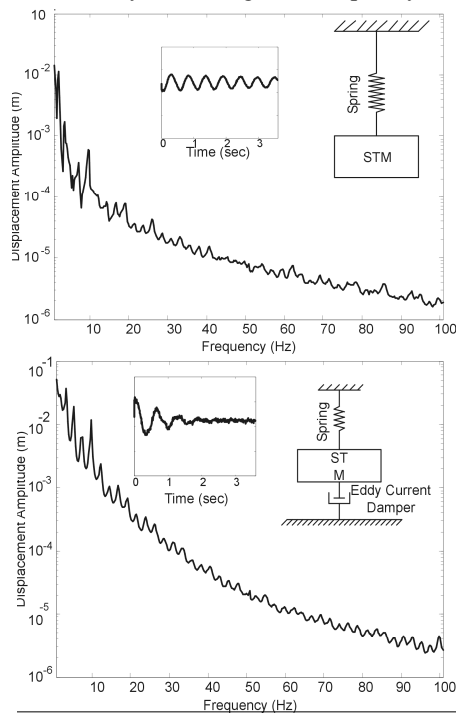


Figure 5: (a) Displacement Amplitude Spectra of the STM Floating Module without Dampers (b) with dampers

The isolation characteristics of the STM with and without the eddy current dampers are shown in Figure 5(a) and Figure 5(b) respectively. The natural frequency of the system is identified from Figure 5(a) as approximately 1.75 Hz. The inset of Figure 5(a) shows the waveform of the free oscillation at its natural frequency after a light tap from the hammer. The oscillations continued for about 30 sec – 40 sec indicating insufficient damping. The inset of Figure 5(b) shows the oscillations after the eddy current dampers were installed. The oscillation of the assembly attenuated within a few seconds.

Conclusions

This paper discussed some issues in characterizing the dynamics of the structure of a scanning tunneling microscope. Some of the key factors such as vibration affecting the performance of the instrument have been identified and necessary precautions are taken to minimize their effects.

Theoretical and experimental modal analysis is performed on the STM structure and the results are compared. An eddy current damper system is designed to enhance the vibration isolation. The amplitude transfer function of the system is measured to evaluate the efficiency of the damping setup.

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