Mechanical Analysis and Measurement of Scanning Probe Microscope Performance

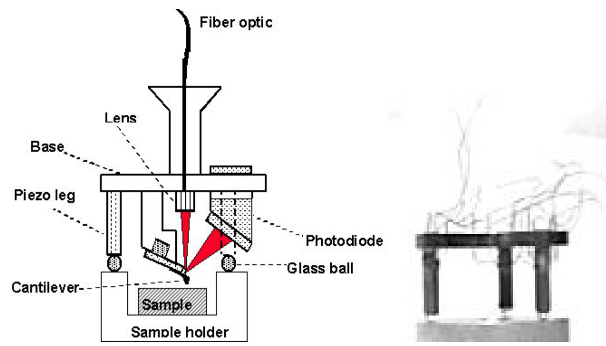
Matthew J. Brukman and Robert W. Carpick[1,2],
Engineering Mechanics Program, Engineering Physics Dept. and Mechanics and Materials Research Group,
[1]Materials Science Program, [2]Rheology Research Center,
University of Wisconsin – Madison, 1500 Engineering Dr. Madison, WI, 53706

ABSTRACT

Euler beam theory and Lagrangian mechanical analysis have been applied to the beetle-style scanning probe microscope (SPM) system to predict the natural frequencies for two significant vibrational modes. In the first mode, the three piezoelectric legs vibrate transversely and the scan head moves from side to side, and in the second, the legs bend tangentially and the scan head twists about its center. The closed-form solutions presented allow the designer to make quantitative comparisons when choosing the materials and dimensions used in the SPM design. The two modes have comparable natural frequencies in the 1-2 kHz range. The frequencies found experimentally are in good agreement with the predictions although other modes are also present. In addition, we find that different materials used in the leg-ramp system may substantially reduce vibrations.

BACKGROUND

The beetle-style SPM [1] is a popular design that may be applied to atomic force microscopy (AFM), scanning tunneling microscopy, or near field optical microscopy in both home-built and commercial systems. In this configuration, the probe equipment is attached to a small disk, to which three piezoelectric legs are soldered [2]. The legs in turn are each attached to glass balls that rest in contact on the sample holder, which is typically a helical ramp [3]. The legs are nickel plated on four outer sectors and a common interior sector, whereby voltages applied to these five sectors induce bending via the piezoelectric effect, allowing the SPM head to raster-scan the sample. In this form of motion, the balls remain fixed on the ramp. Alternately, the entire head can be “walked” by inertial motion where fast voltage pulses cause the balls to slide along the ramp. This enables coarse positioning of the head for acquiring data over a range of positions. As well, by walking the scan head clockwise or counter-clockwise on the helical ramp, the SPM is brought gently into or out of contact with the sample. Another advantage of the beetle design is that the sample is completely decoupled from any piezoelectric scanning elements, allowing thermal, mechanical, or other stimuli to be easily applied to the sample. A schematic of the AFM is shown in Figure 1(a).



(a) (b)

Figure 1. (a) Schematic drawing of beetle-type AFM. (b) The actual scan head as tested.

Since SPMs are used to image samples with sub-nanometer scale resolution, even very small amounts of mechanical noise may severely impair data collection [4]. To attenuate ambient mechanical noise, SPMs are typically placed in basements, on air-tables, in soundproof chambers, and on suspension systems, sometimes all simultaneously. Even when precautions are taken, some vibrations are passed from the environment to the SPM. This may be modeled as driven oscillator with natural frequency ω_0 [5]. The transfer function |T(ω)| governing this interaction provides the amplitude ratio of SPM vibration to noise input from the environment and is found, in the case of a one-stage vibration isolation system, to be proportional to $(\omega - \omega_0)^{-2}$ for $\omega > \omega_0$. To minimize mechanical noise then, SPMs should be constructed with natural frequencies as high as possible.

Of course, SPMs have many degrees of freedom and accordingly have many natural modes of vibration, each with its own frequency. Since the input noise spectrum is the same for all modes, it is the natural mode with the lowest natural frequency that will have the greatest transfer function at a given frequency and the designer's attention should be focused on the one or two lowest frequency modes so that their frequencies may be increased.

Feedback gains provide another reason to consider the resonant frequencies of the SPM during the design process. Feedback in the SPM system regulates the interaction between the sample and probe by comparing a measured value of surface-probe interaction with a user-defined set point. This difference is called the error signal. Voltages on the piezo legs' sectors are then changed in proportion to the error signal, its time derivative, or its time integral, with the proportionality constants called the gains. Higher gains allow faster scanning, but as the gains are increased, the frequency bandwidth over which the electronics respond increases. If the gains are set high enough, then the feedback system will drive the legs at resonance, introducing significant errors in the SPM signal. With increased natural frequencies of the legs, the gains may be set higher and the user will be able to image samples faster.

ANALYSIS

In the beetle design, the probe equipment (*i.e.* lens, cantilever, and photodiode in the AFM case) is attached to a disk, to which the legs are soldered. The legs are attached to glass or sapphire balls that contact the ramp. In this configuration, the legs may be modeled as beams cantilevered and displaced at the disk end and tip loaded at the ball end. Running x along a leg from the ball (x = 0) to the disk (x = L) the boundary conditions for displacement $u(x)$ are thus (with subscripts indicating differentiation in x):

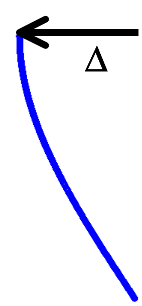


Figure 2. Leg bending model.

(1) $u(L)=\Delta,\ u_x(L)=0,\ u(0)=0,\ u_{xx}(0)=0.$

Using Euler beam theory, the deflection of the leg may be written as a cubic polynomial:

(2) $u(x) = \Delta\cdot(3x^2/L^2 - x^3/L^3).$

A model of one leg, translated with no bending at the disk and fixed but free to bend at the base, is shown in Figure 2. The two vibrational modes considered have different definitions of Δ. In the translational case, all three legs deflect the same amount Δ_t and in the same direction. For rotation, the three legs deflect tangentially by R·Θ, R being the distance from the legs to the center of the disk and Θ being the angle of rotation (Figure 3). Assuming that the SPM oscillates harmonically, the motion is described by:

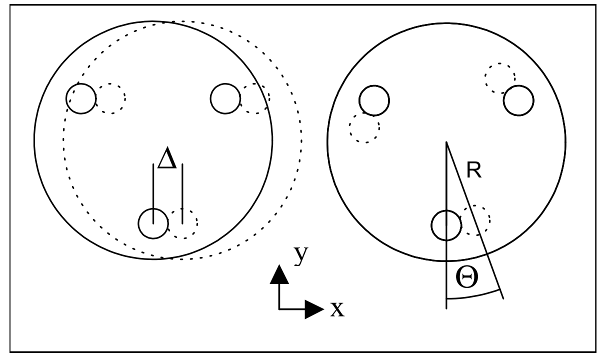


Figure 3. Top views of the scan head's motion. Left: translation by distance Δ. Right: rotation by angle Θ.

(3) $u(x) = \Delta(t)\cdot(3x^2/L^2 - x^3/L^3)$
$= \Delta_0\sin(\omega\cdot t)\cdot(3x^2/L^2 - x^3/L^3).$

The kinetic energy of translation, T_t, is equal to $3\cdot T_{leg} + \frac{1}{2}M(d\Delta/dt)^2$, with T_{leg} the energy of an individual leg and M the mass of the disk and attached probe equipment. With ρ and A defined as the density and cross sectional area of a leg, T_{leg} is known. Superscript dots denote partial differentiation with respect to time t. T_{leg} is simply given by:

(4) $T_{leg} = \frac{1}{2}\int \rho\cdot A\cdot(\dot{u}(x,t))^2 dx$.

The kinetic energy of rotation is:

(5) $T_r = 3T_{leg} + \frac{1}{2} J\cdot\dot{\Theta}^2$

where J is the polar moment of inertial of the disk and probe equipment about the disk's central axis. The potential energy of the three legs' bending is:

(6) $V = \frac{3}{2}\int EI(u_{xx}(x,t))^2 dx$,

where E and I are the Young's modulus and area moment of inertia for the leg, respectively.

The next step in finding the natural frequencies is to apply Lagrange's equations to the energies:

(7) $L_t = T_t - V$ (8) $\partial L_t/\partial\Delta_t = d/dt\,(\partial L_t/\partial\dot{\Delta}_t)$

(9) $L_r = T_r - V$ (10) $\partial L_r/\partial\Theta = d/dt\,(\partial L_r/\partial\dot{\Theta})$

After the harmonic assumption is applied, eqns (8) and (10) yield formulas for ω_t and ω_r:

(11) $-\frac{\ddot{\Delta}}{\Delta} = \omega_t^2 = \frac{9EI}{L^3} \div \left(\frac{51}{35}\rho AL + M\right)$

(12) $-\frac{\ddot{\Theta}}{\Theta} = \omega_r^2 = \frac{9EIR^2}{L^3} \div \left(\frac{51}{35}\rho ALR^2 + J\right)$

or in units of Hertz:

(13) $f_t = \frac{3}{2\pi}\sqrt{\frac{EI}{L^3}} \times \sqrt{\frac{1}{(\frac{51}{35}\rho AL + M)}}$

(14) $f_r = \frac{3}{2\pi}\sqrt{\frac{EI}{L^3}} \times \sqrt{\frac{R^2}{(\frac{51}{35}\rho ALR^2 + J)}}$.

These explicit formulas for two low-frequency, high-amplitude modes should aid the designer of a SPM. In particular, the following conclusions may be drawn:

1) When deciding on the length of the piezo leg, the designer must compare vibrational response and piezo sensitivity (displacement per applied volt, *i.e.* the scan range), which increases as L^2. Resonant frequency falls, however, as $L^{-1.5}$.

2) It is advantageous to increase the area moment I of the legs. For a thin walled tube of mean radius r and thickness τ, I may be approximated as $\pi r^3\tau/4$, so f should increase as $r^{1.5}$. The cross sectional area also increases with r, however, equaling $2\pi r\tau$ for a thin tube. If the product $r\cdot\tau$ is kept constant while r increases, then f will increase linearly with tube mean radius. Following this rule and making the tubes thinner has the secondary benefit of increasing scan range without loss of stability, for the piezo sensitivity varies as $1/\tau$.

3) The material used for the disk and lens housing is important. Aluminum is used commonly, but there are superior alternatives. Beryllium, for example, is less dense (ρ = 1.8 g/cm^3), reducing M and J, and stronger (E = 303 GPa) than Al (ρ = 2.7 g/cm^3, E = 70 GPa). Since it is stiffer, the disk and lens housing may be made thinner than with Al, further reducing the system mass. Unfortunately, beryllium is difficult to obtain and machine because of health risks.

4) If the rotational frequency is calculated to be less than the translational frequency, then the designer should consider increasing the leg separation parameter R until the two frequencies are equal. The combined masses of the legs are generally much smaller than total mass of the disk and probe equipment, so the variation is almost linear. If f_r is predicted to be greater than f_t, then increasing R would decrease performance, since it is the lower of the two frequencies that most limits microscope performance and increasing R may require a larger disk, increasing the mass of the system.

Behler and co-workers have previously modeled the beetle system [6] using a simplified resonance analysis. Our approach differs in that we take into account both the mass of the legs and the rotational moment of inertia of the disk.

EXPERIMENTAL TESTING

The scan head tested consisted only of the disk and three legs so that the polar moments of inertia of the probe equipment, which are not readily calculated, would not hinder comparison with the predicted frequencies. The scan head was attached to control electronics via fifteen 0.003-inch kapton-coated copper wires and placed on a vibration isolation system in a 2 m cube lined with anechoic foam. In two series of tests, the scan head rested on either steel or a PTFE (known commercially as Teflon) block. Steel was chosen to replicate the stainless steel ramp used during imaging and the use of PTFE allowed us to observe the effects of boundary conditions on the resonant frequency.

The method used to drive and sense the head motions is nearly identical to that of Behler *et al.*[6] For translation mode testing, two legs were driven by a function generator in the same direction (labeled y, see Figure. 3). As the scan head vibrated, deformations were induced in the y sectors of the third leg, further inducing electric fields in those sectors due to the inverse piezoelectric effect. The voltages in the two y sectors were connected to the differential input of a lock-in amplifier referenced to the function generator.

For rotation testing, five of the twelve outer sectors were driven tangentially (using a series of voltage dividers to supply the appropriate voltages to the sectors) and the voltage response from five opposite sectors were summed to calculate an average response relative to the inner sectors and, as above, connected to the lock-in amplifier. The two remaining sectors were unused.

The data were taken as two frequency sweeps, 10-250 Hz and 250-600 Hz. There is some uncertainty in the synchronization between the function generator and the lock-in. The uncertainty increases exponentially with frequency from 0.1 Hz at 0 Hz to 30 Hz at 6000 Hz.

DISCUSSION

The selected data presented here (Figures 4 and 5) are representative of the spectra taken in each configuration. The data are scaled such that the peak amplitude in each figure has a value of 1 with the scales for the two figures being different.

The scan head was in slightly different positions on the surfaces for the scans and the spectra differed from position to position. Peak splitting was observed in the steel spectra and peak shifting in both steel (by about 40 Hz) and PTFE (50 Hz) scans. The steel spectrum shown exhibits peaks at 1770, 1863, 2083 and 2170 Hz. The second is believed to be the predicted translation mode and the first is thought to show coupling between the rotation and translation modes. The third and fourth peaks are not understood. They may be additional modes or they may be the actual first modes of translation and rotation and represent errors in the model.

The rotation on steel spectrum has large peaks at 1600 and 1827 Hz. Again, these are believed to be the first rotation and translation modes. The two peaks centered on 1200 Hz are not understood. It is suspected that they correspond to a mode that has two legs pivoting about the third.

The wires that connect the control electronics to the piezo legs were not considered in the model but they do add a small amount of stiffness, increasing the natural frequencies of vibration. They also transmit vibrations from the environment to the scan head.

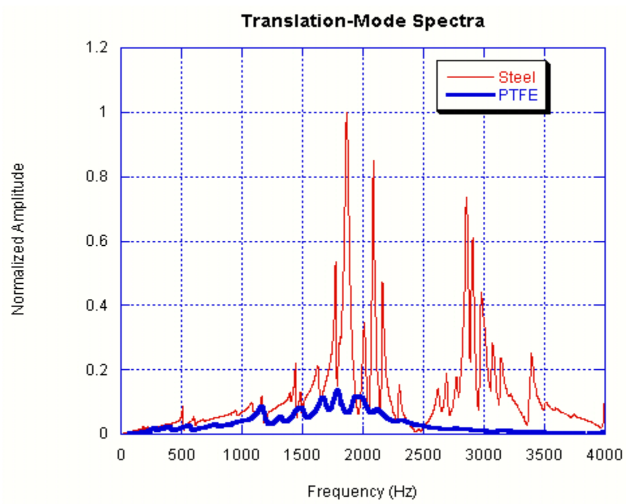


Figure 4. Translation mode characterization.

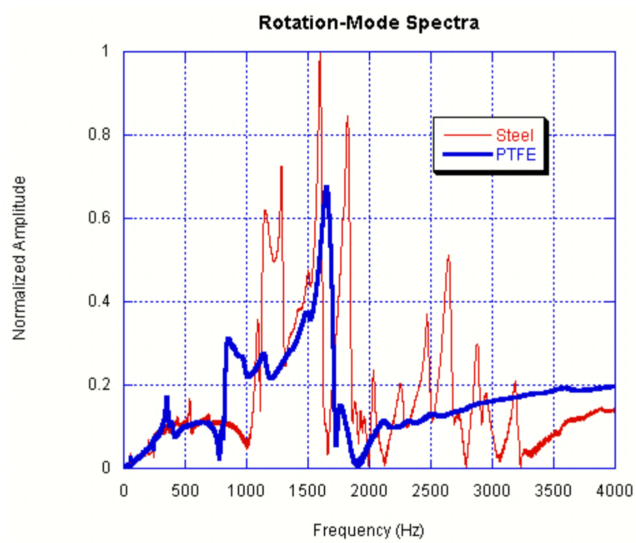


Figure 5. Rotation mode characterization.

The difference in response between steel and PTFE surfaces is striking. The polymer substantially attenuates the resonance peaks below 2500 Hz and almost totally eliminates them above 2500 Hz, especially in the case of the translation mode. As SPM users continually seek to find ways to reduce vibration in their systems, this effect may be of great utility in future microscope design.

The different behaviors may derive from the materials or the geometry, since the pieces of steel and PTFE were of different dimensions. It remains to be seen if PTFE (or any other) polymer is still an effective damper as a thin film coated on the ramp. Such a configuration is needed since polymers, including PTFE, have large thermal expansion coefficients. Alternatively, polymers may replace the glass and sapphire that are typically used in the balls.

CONCLUSION

Lagrangian mechanics and Euler beam bending theory have been applied to the scan head of the beetle-style scanning probe microscope and two major modes of vibration have been found. The closed-form solution for the natural frequencies of scan head translation and rotation presented allow the SPM designer to quantitatively determine how changes in dimension and material will impact the response of the microscope when subjected to mechanical noise. Testing of a home-built atomic force microscope validates the solutions and testing on multiple surfaces has raised the possibilities of new materials choices in SPM systems.

ACKNOWLEDGEMENTS

MJB acknowledges the National Science Foundation for support via a Graduate Student Research Fellowship.

REFERENCES

(1) K. Besocke, "An easily operable scanning tunneling microscope", *Surf. Sci.*, **181**, 145 (1987).

(2) C.J. Chen, "Electromechanical deflections of piezoelectric tubes with quartered electrodes", *Appl. Phys. Lett. (USA),* **60**, 132 (1992).

(3) J. Frohn, J.F. Wolf, K. Besocke, M. Teske, "Coarse tip distance adjustment and positioner for a scanning tunneling microscope", *Rev. Sci. Instrum. (USA),* **60**, 1200 (1989).

(4) D.W. Pohl, "Some design criteria in scanning tunneling microscopy", *IBM J. Res. Dev. (USA),* **30**, 417 (1986).

(5) S.-I. Park, R.C. Barrett, "Design Considerations for an STM System", *Methods of Experimental Physics,* **27**, 31 (1993).

(6) S. Behler, M.K. Rose, D.F. Ogletree, M. Salmeron, "Method to characterize the vibrational response of a beetle type scanning tunneling microscope", *Rev. Sci. Instrum. (USA),* **68**, 124 (1997).