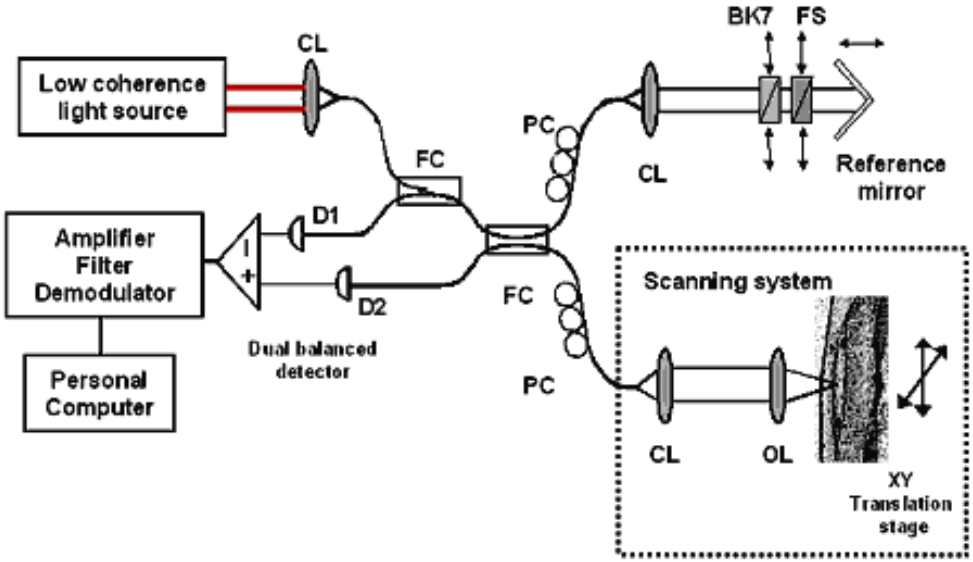
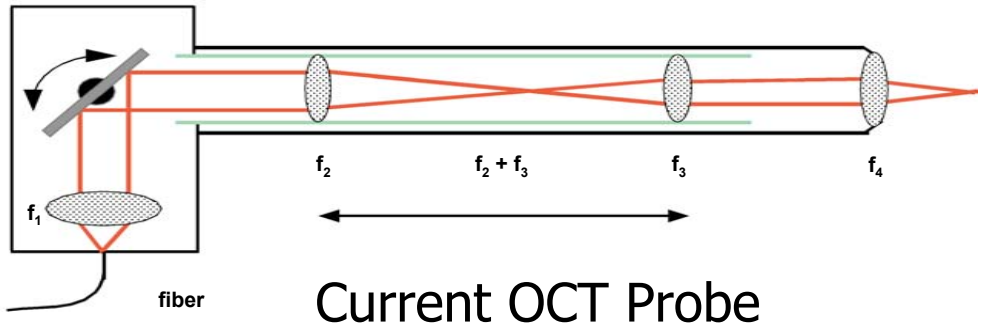
Optical MEMS Scanner for Bio-Medical Imaging

Group F2

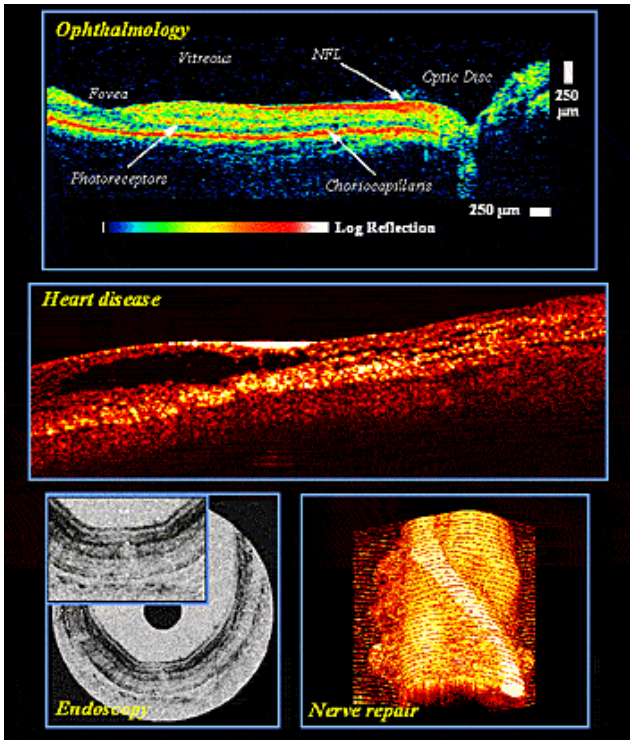
Shorya Awtar, David Reinharth, Klint Rose,
Wenyang Sun, Hoe Phong Tham

Optical Coherence Tomography





Current OCT Probe

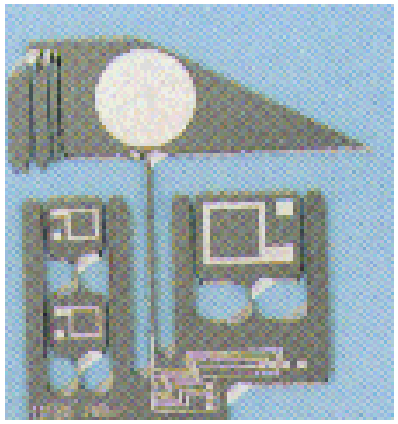


OCT Images

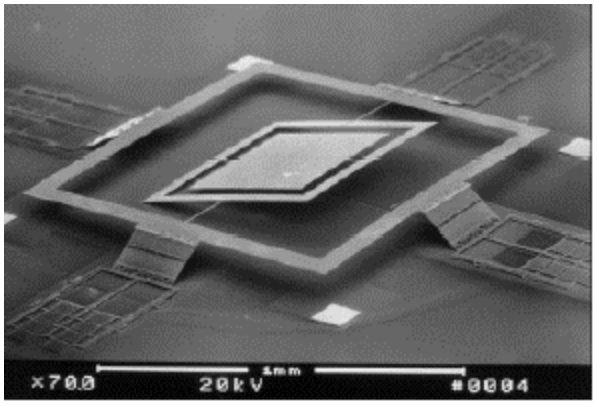
Design Requirements

- Optical probe to scan an optical beam across tissue
  - Handheld, pen-like package
  - Fit within 9 mm inner diameter
- Two-dimensional scan
  - 2-4 kHz for each direction
  - +/- 15 degrees optical (+/- 7.5 mechanical)
- Reflective scanner surface
- 2-10 μm spot size at 1-2 mm working distance
- Power: 1 mA at 50-100 volts

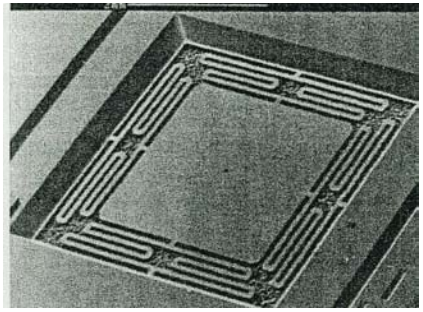
Typical 2D Mirror Designs



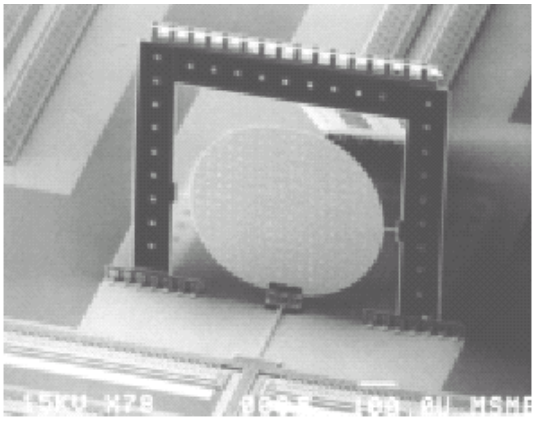
Single beam design (OMRON)



Double gimbals

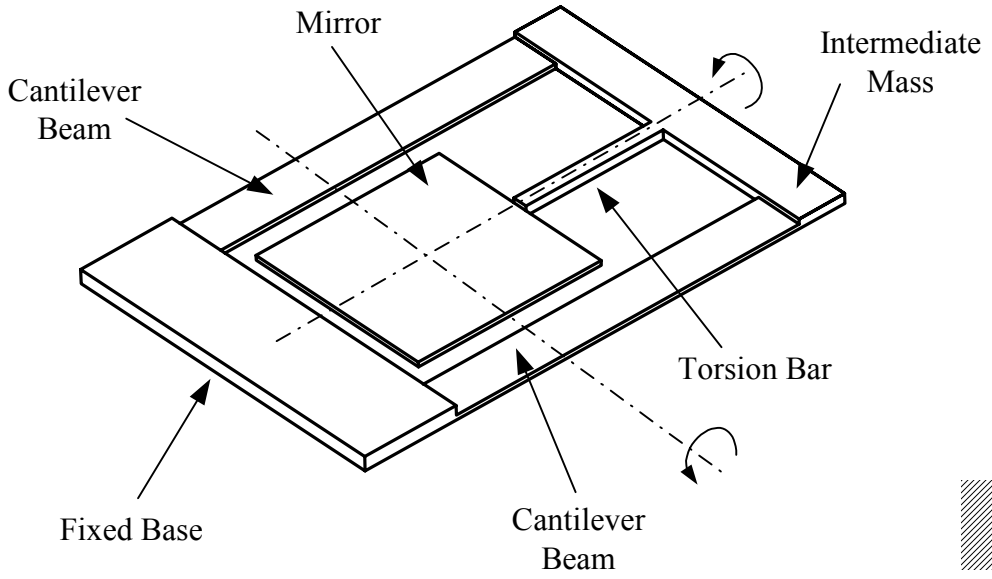


Folded beams



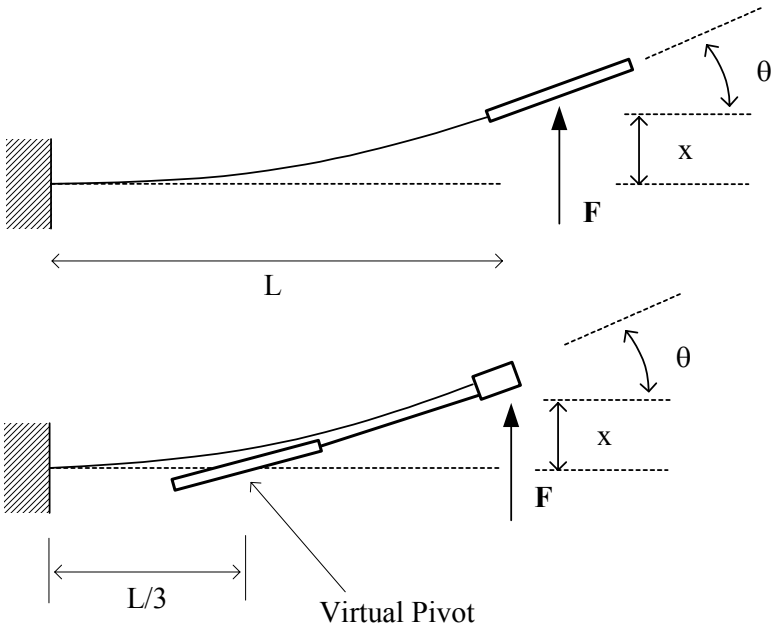
3-D Self-assembly

Device Concept

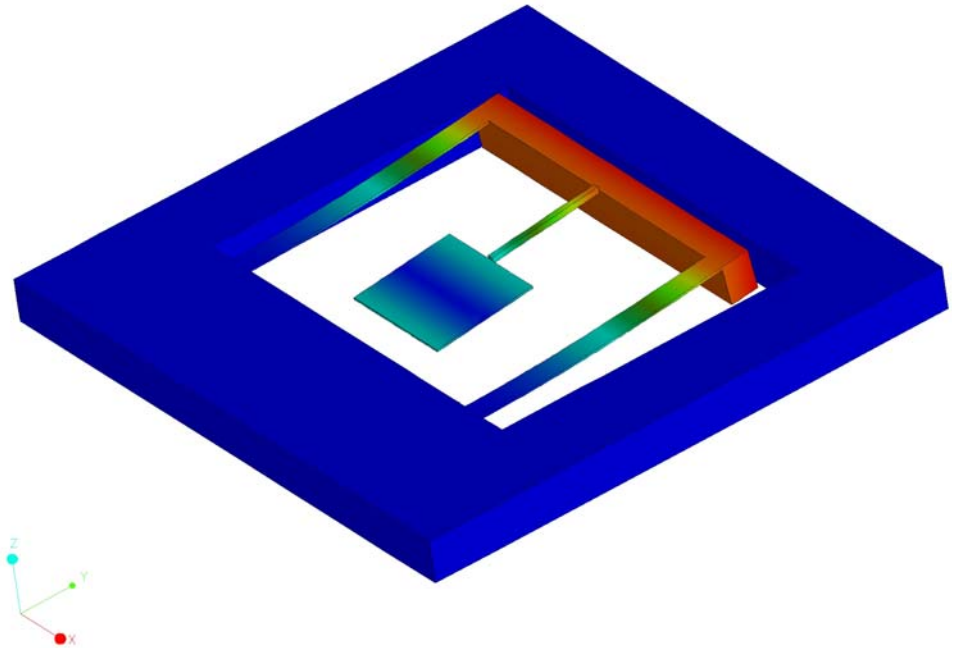


- No mirror translation
  - Principle of Virtual Pivot
  - Valid for zero moment at the cantilever tip

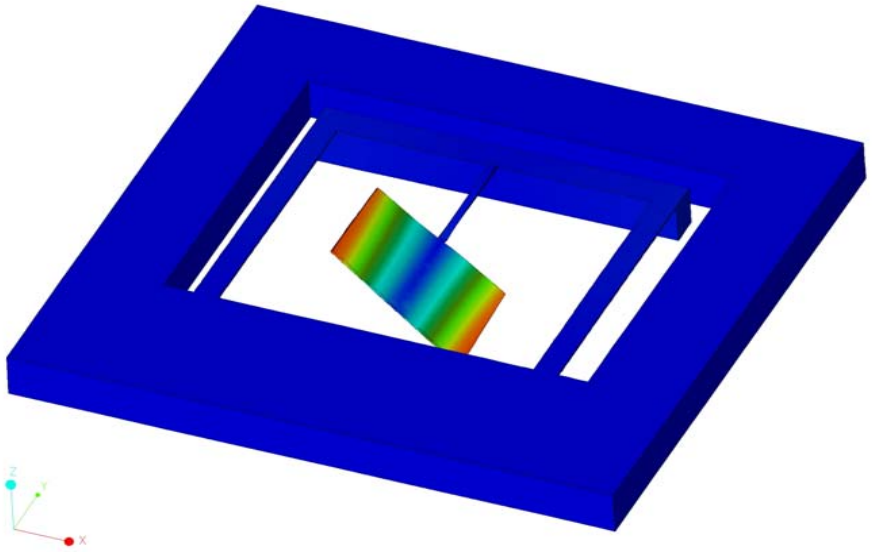
- Decoupled DOFs
  - Cantilever beams produce bending
  - Torsion bar produces twisting



Finite Element Based Modal Analysis



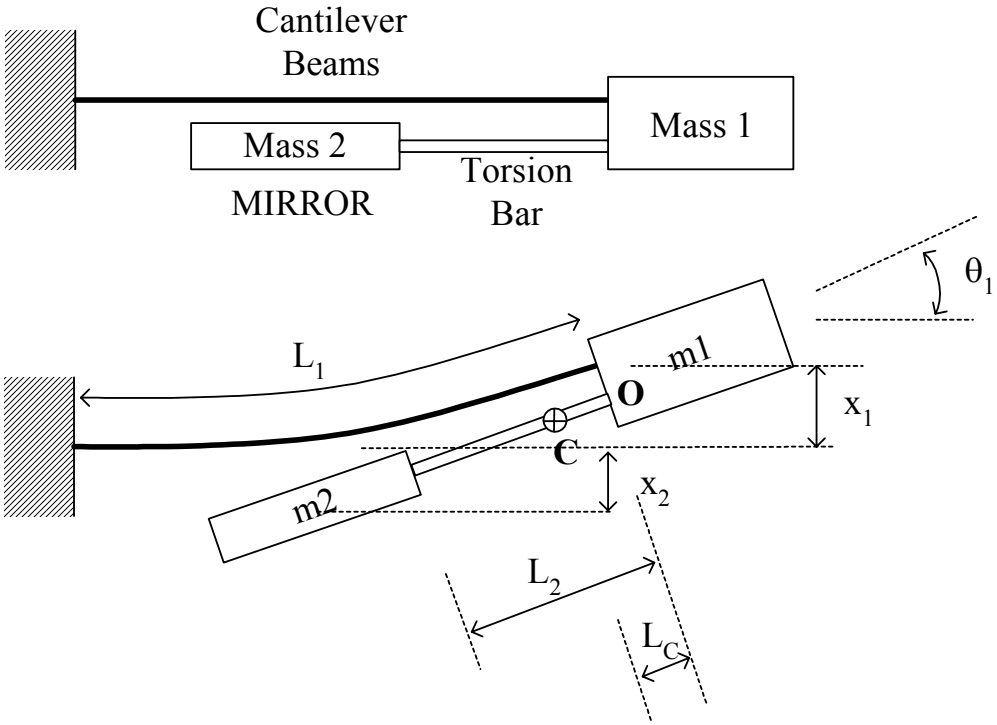
Bending Mode
560 Hz



Torsion Mode
15.5 kHz

System Modeling: Dynamics

- Bending Mode Analysis
- Assume torsion bar is rigid in bending
- Generalized coordinates : x_1 and θ_1



$$(ms^2 + b_1 s + k_1)\, X_1 = (mL_c s^2 + b_1 L_c s + \frac{k_1 L_1}{2})\, \Theta_1 + \frac{k_1 L_1 c_1}{2}\, V_{in}$$

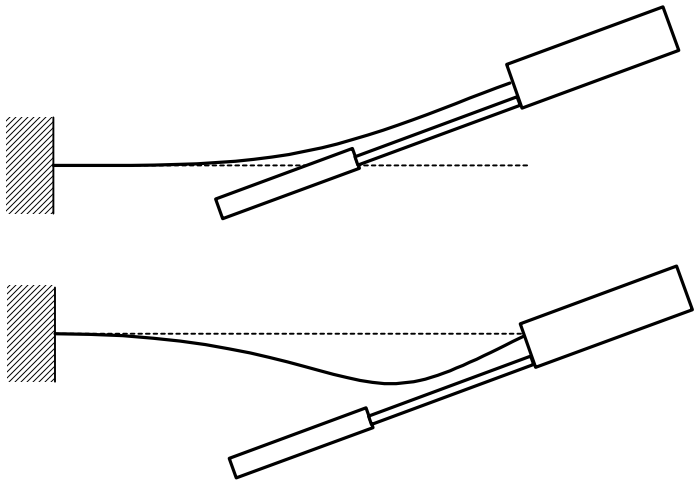
$$[I_1^{\,o} s^2 + (b_2 + b_1 L_c^{\,2}) s + \frac{k_1 L_1^{\,2}}{3}]\, \Theta_1 = (mL_c s^2 + b_1 L_c s + \frac{k_1 L_1}{2})\, X_1 - \frac{k_1 L_1^{\,2} c_1}{6}\, V_{in}$$

System Modeling: Dynamics

$$\left[mI_1^{\,c}s^4 + (mb_2 + I_1^{\,c}b_1)s^3 + \left\{ mk_1L_1\left(\frac{L_1}{3} - L_c\right) + k_1I_1^{\,o} + b_1b_2 \right\} s^2 .....\right.$$

$$\left. .... \left\{ b_1k_1L_1\left(\frac{L_1}{3} - L_c\right) + b_1k_1L_c^{\,2} + b_2k_1 \right\} s + \frac{k_1^{\,2}L_1^{\,2}}{12} \right] \Theta_1$$

$$= \left[m\left(L_c - \frac{L_1}{3}\right)s^2 + b_1\left(L_c - \frac{L_1}{3}\right)s + \frac{k_1L_1}{6} \right] \frac{k_1L_1c_1}{2} V_{in}$$

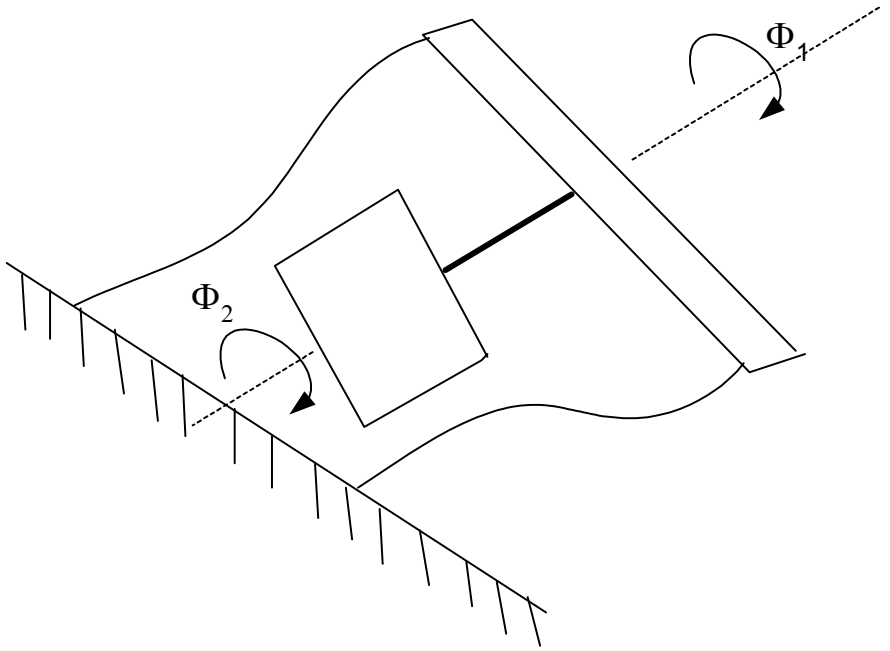


First Bending Mode

Second Bending Mode

System Modeling: Dynamics

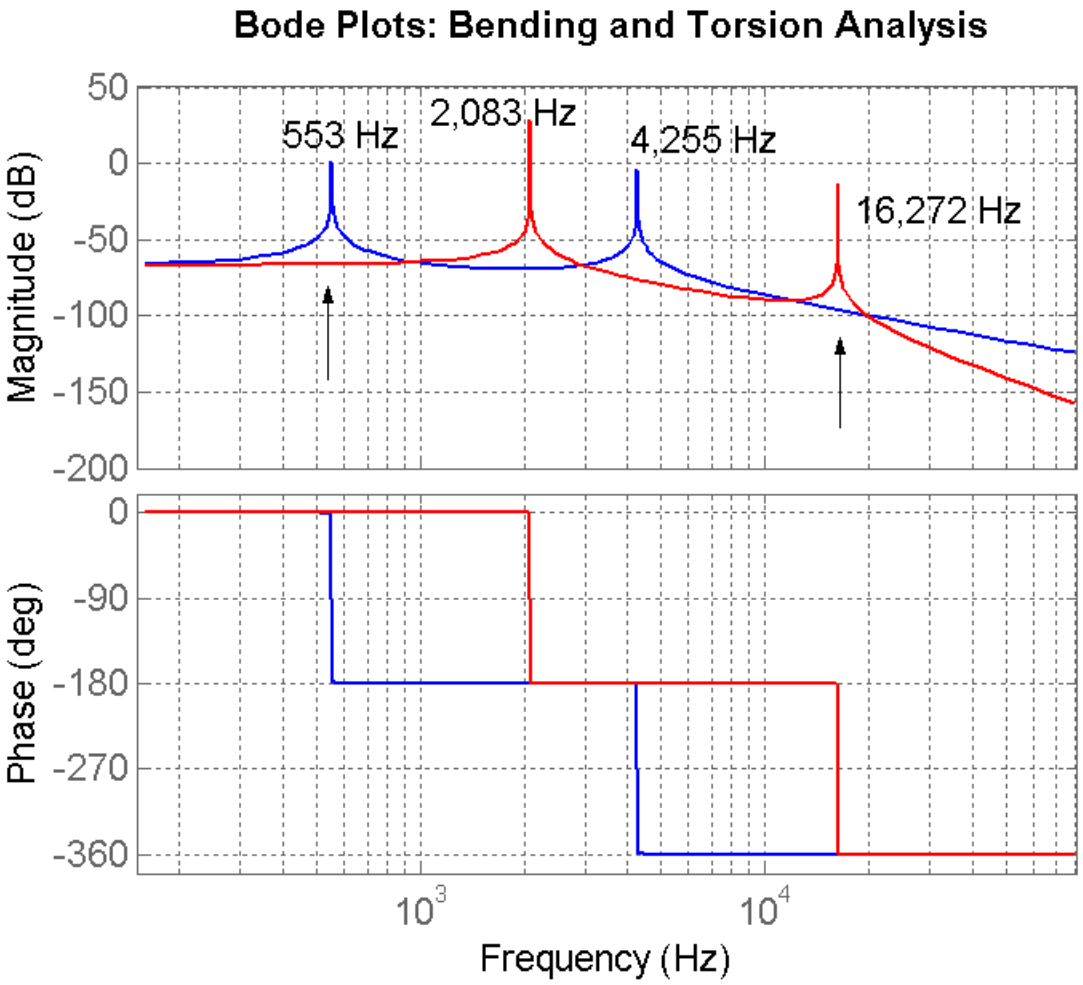
- Torsion Mode Analysis
- Generalized coordinates : ϕ_1 and ϕ_2



$$\left[I_1{}^t I_2{}^t s^4 + (I_1{}^t b_4 + I_2{}^t b_3) s^3 + \left\{ I_1{}^t K_2 + I_2{}^t (K_1 + K_2) + b_3 b_4 \right\} s^2 ..... \right.$$

$$\left. .... \left\{ K_2 (b_3 + b_4) + K_1 b_4 \right\} s + K_1 K_2 \right] \Phi_2 = K_2 c_2 V_{in}$$

System Modeling: MATLAB Results



- Blue: Bending Analysis
  - W_1=553 Hz Q_1=1740
  - W_2=4225 Hz Q_2=2700
- Red: Torsion Analysis
  - W_3=2083 Hz Q_3=56000
  - W_4=16.3 kHz Q_4=26190

FEM vs. Analytical Model

Vibration Mode	Description	ProE Results	MATLAB Results
1	**Simple Bending of Cantilevers**	**562 Hz**	**553 Hz**
2	Simple Torsion of the Cantilevers	2190 Hz	2083 Hz
3	Compound Bending of Cantilever	3025 Hz	4225 Hz
4	In-plane Bending of Torsion Bar	5623 Hz	
5	In-plane Bending of Cantilever Beams and Torsion Bar	11.67 kHz	
6	Out-of-plane Bending of Torsion Bar	12.63 kHz	
7	**Pure Torsion of Torsion Bar**	**15.54 kHz**	**16.27 kHz**

Range of Motion and Input Voltage

- Analytical Estimate (MATLAB)
  - Θ_1/V_{in} is 0 dB @ 553 Hz
    => 5.7deg/V for Q=1740
  - Φ_2/V_{in} is –15 dB @ 16.27 kHz
    => 1.8 deg/V for Q=26190
- FEA Estimate (Pro/Mechanica)
  - Apply dynamic load boundary condition
  - Specify a quality factor of Q=1500
  - 15 deg/V
- Experimental measurement of damping shall produce more accurate estimates

System Modeling: Damping

- Effect of air damping directly affects performance of scanning system
- A simple model was used to estimate 4 different damping coefficients of the structure

Fluid Mechanics Considerations

- Stokes' Law $\Rightarrow Re < 1$
  $\Rightarrow$ linear relationship between force and velocity
- For our model, $Re = 23.1$ (> 1)

What to do? ☹

- Work in the transition region ($10 < Re < 1000$)
  - Messy calculations
  - Fluidic FEM analysis
- Make simplifying assumptions and work with Stokes' Law

System Modeling: Damping

Working With Stokes' Law

- Assuming Stokes' Law for *Re* > 1 results in huge underestimation of actual damping present in system
- Solution: Use the constant value calculated from Stokes' Law and multiply it by a factor of 10
  - Compensates for underestimation
  - takes into account material damping inherent in system

System Modeling: Damping

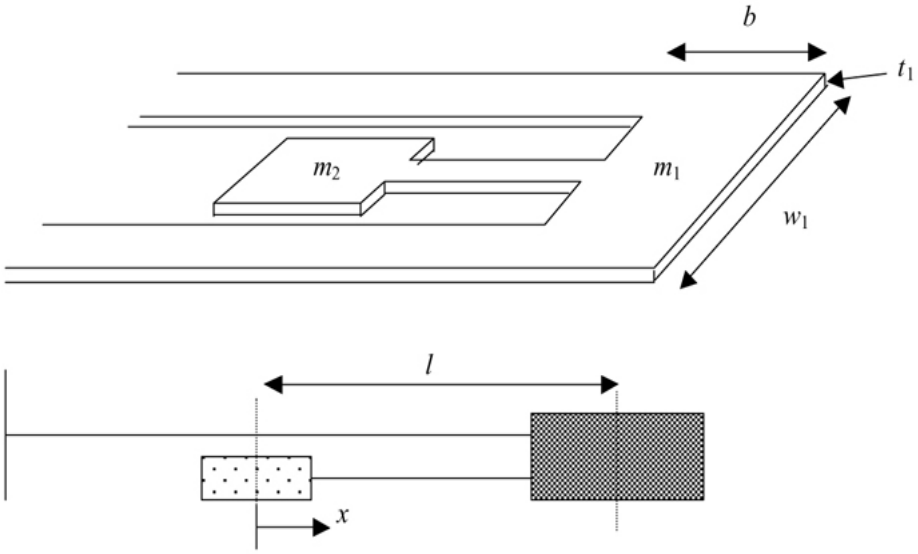
Translational Damping in Bending (b_1)

- Assuming Stokes' Law:

$$F = C_D \eta w_1 V = b_1 V$$

$$\Rightarrow \quad \boxed{b_1 = C_D \eta w_1}$$

$$\Rightarrow \boldsymbol{b_1 = 4.8 \times 10^{-6}\ \text{Ns/m}}$$

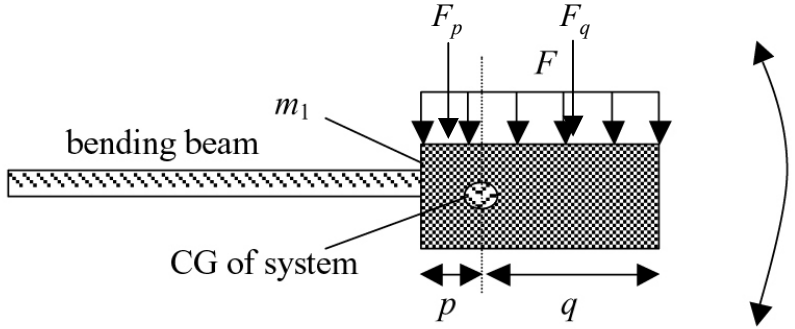


System Modeling: Damping

Rotational Damping in Bending (b_2)

- Assume that force F acting on m_1 as it rotates is constant

$$F_p = \frac{Fp}{p+q} \qquad F_q = \frac{Fq}{p+q}$$



- Moment, M_1, acting on m_1 about CG of the structure due to F is:

$$\Rightarrow M_1 = \frac{b_1 l \theta_{\max} (q-p)}{2} \omega$$

System Modeling: Damping

- Next, we find M_2 acting on m_2 due to bending of the beams
- Pressure, P:

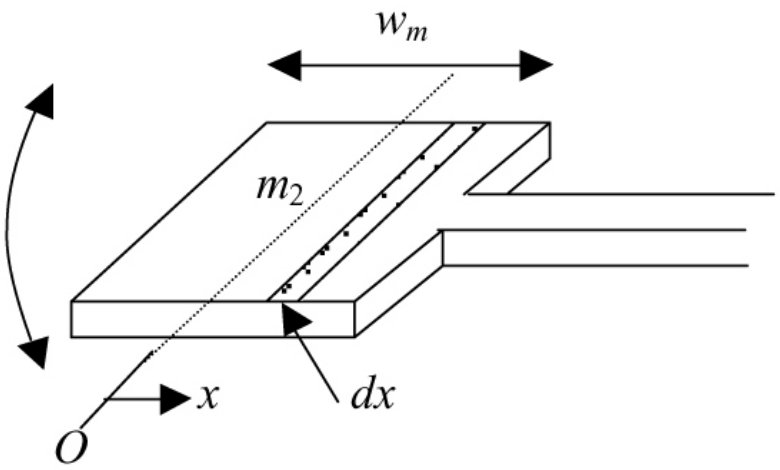
$$P = \frac{F}{w_m^{\,2}} = \frac{b_1 V}{w_m^{\,2}}$$

- Differential force, dF:

$$dF = \frac{b_1 \theta_{\max} \omega}{w_m} x dx$$

- Then M_2 of entire mass m_2 about O is:

$$M_2 = 2\int_0^{\frac{w_m}{2}} \frac{b_1 \theta_{\max} \omega}{w_m} x^2 dx = \frac{1}{12} b_1 \theta_{\max} w_m^{\,2} \omega$$



- Moment of entire structure about its CG:

$$M = M_1 + M_2 = \left[\frac{b_1 \theta_{\max}}{12}\left(6l(q-p) + w_m^{\,2}\right)\right]\omega = b_2 \omega$$

$$\Rightarrow \quad b_2 = \frac{b_1 \theta_{\max}}{12}\left(6l(q-p) + w_m^{\,2}\right)$$

$$\Rightarrow \quad \mathbf{b_2 = 1.68 \times 10^{-12}\ Nsm}$$

System Modeling: Damping

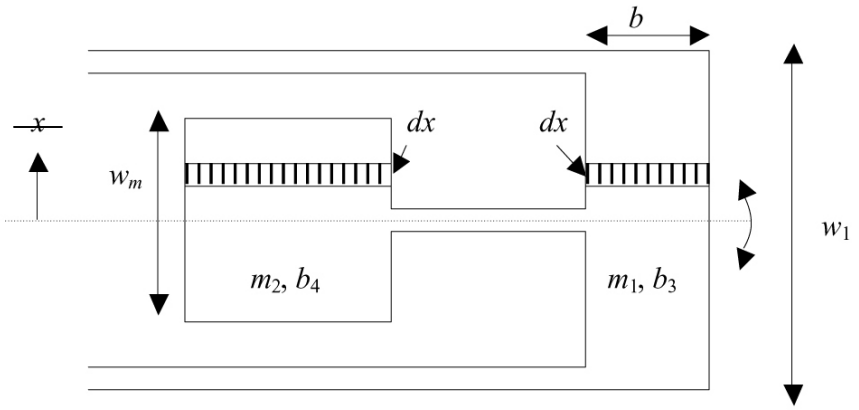
Rotational Damping in Torsion (b_3 and b_4)

- Consider m_1. We know that $F = C_D \eta w_1 V$
- Pressure, P:

$$P = \frac{C_D \eta w_1 V}{w_1 b} = \frac{C_D \eta V}{b}$$

- Differential force, dF:

$$dF = C_D \eta \theta_{\max} \omega x dx$$



- Moment of the entire mass m_1 about centerline:

$$M = 2\int_0^{\frac{w_1}{2}} C_D \eta \theta_{\max} \omega x^2 dx = \frac{1}{12} C_D \eta \theta_{\max} w_1^{\,3} \omega = b_3 \omega$$

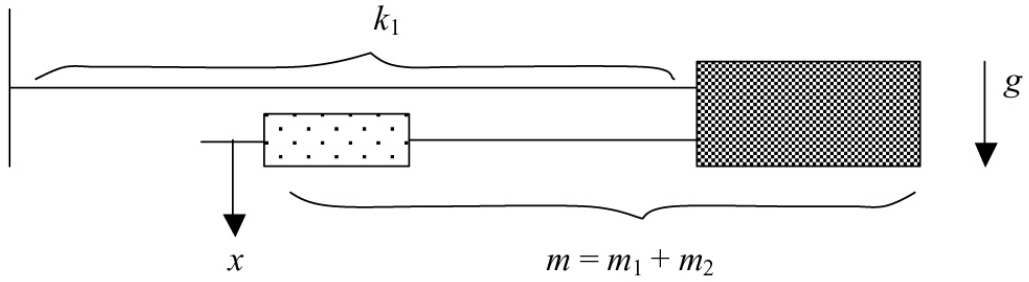
$$\Rightarrow \quad b_3 = \frac{1}{12} C_D \eta \theta_{\max} w_1^{\,3}$$

$$b_4 = \frac{1}{12} C_D \eta \theta_{\max} w_m^{\,3}$$

$$\Rightarrow \quad b_3 = 6.12 \times 10^{-13} \text{ Nsm} \qquad b_4 = 1.65 \times 10^{-14} \text{ Nsm}$$

Modeling: Static Deflection

Deflection Due to Gravity



- Deflection: $x = mg/k_1 =$ **0.183 μm**

Modeling: Mirror Deformation

Dynamic Deformation of Mirror [1]

- Dimensions of scanning mirror optimized to satisfy Rayleigh Limit.
- Rayleigh Limit allows a peak-to-valley surface deformation, δ, of $\lambda/4$.
  $\Rightarrow \delta <$ 212.5 nm

$$\delta = 0.183 \frac{\rho(1-\nu^2)(2\pi f)^2 \theta}{Et^2}\left(\frac{l}{2}\right)^5$$

- δ for our mirror: **212.1 nm**



[1] R.A. Conant, J.T. Nee, K.Y. Lau and R.S. Muller, "*A Flat High-Frequency Scanning Micromirror*", Hilton Head 2000 Conference, Hilton Head, South Carolina, June 4-8, 2000.

Modeling: Fatigue Failure

- Maximum bending stress, σ_{max} = **2.73 MPa**
- Maximum shear stress, τ_{max} = **611 MPa**

} Safety Factor of 2

- Yield strength of silicon: **2,800 – 6,800 MPa**

$\Rightarrow$ Fatigue by cyclical bending and torsion is not a concern

Actuation Schemes

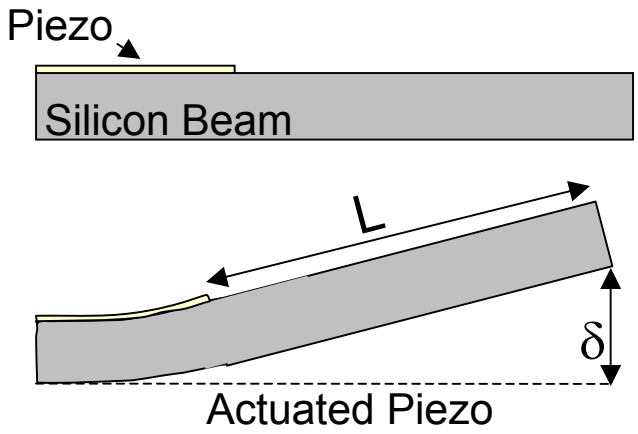
- Electromagnetic:
  - Provides relatively large forces and has high repeatability
  - But poor in high-frequency operations
  - External coils needed to produce magnetic field
- Electrostatic:
  - Versatile and simple
  - But inherent problem of electrostatic instability (i.e. pull-in voltage)
  - Problem of non-linearity – crosstalk is too large to achieve independent control of the angles
- Thermal:
  - Inherent limitations for long-term use – at high temperatures, silicon loses its nearly perfect linear-elastic properties
  - Large power consumption
- Piezoelectric:
  - Very rapid response to alternating voltage input required to drive the MEMS structure at its higher resonant frequency of 15.5 kHz
  - Also highly repeatable and very reliable for MEMS applications

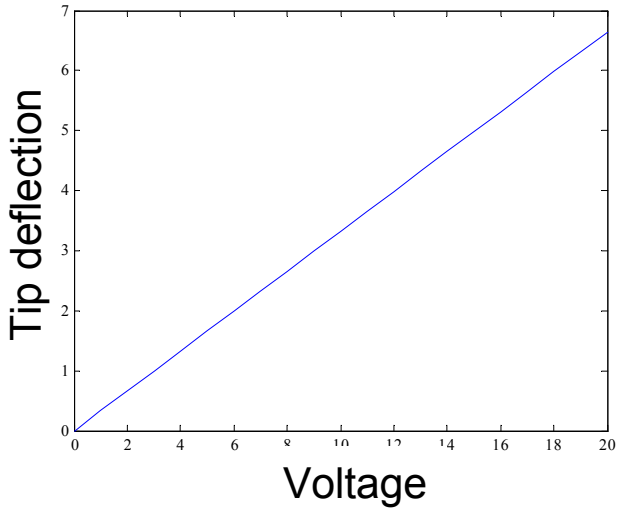
PZT Actuation: Base Deflection

- Voltage causes piezo to constrict
  - Electrical to mechanical coupling
- Traction between piezo and beam causes bending
- Deflection vs. voltage should be linear

$$\delta_{total} = K_1 V + L\sin(K_2 V)$$

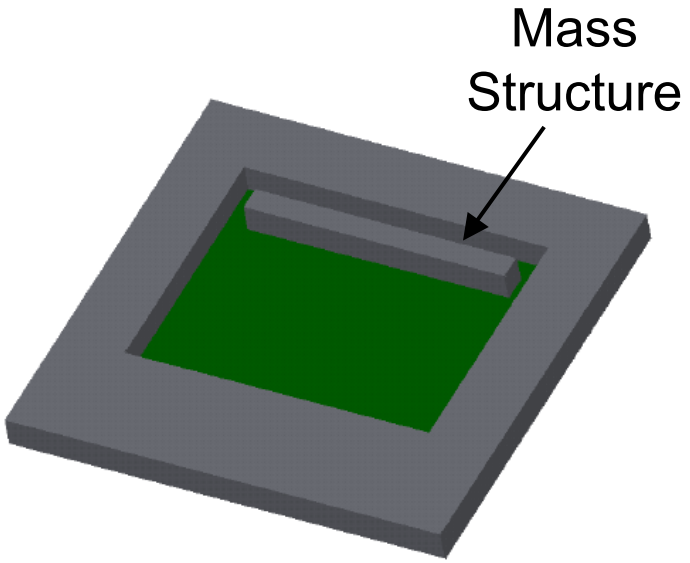
K_1=20.5 K_2=1.85e^{-4}





Fabrication: Backside Processing

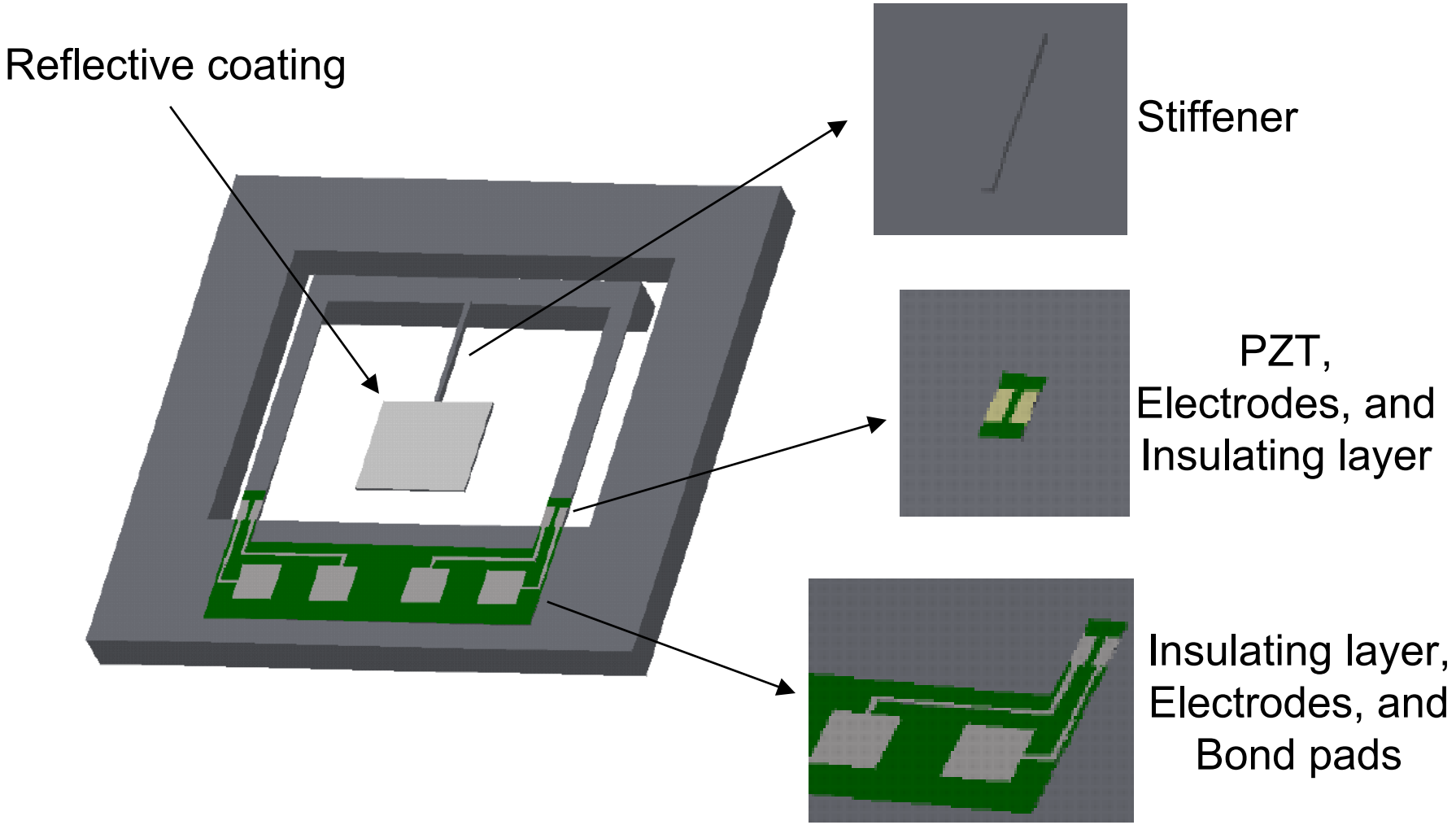
- SOI wafer substrate
  - Defines thickness for front-side features
  - Provides etch stop for backside DRIE





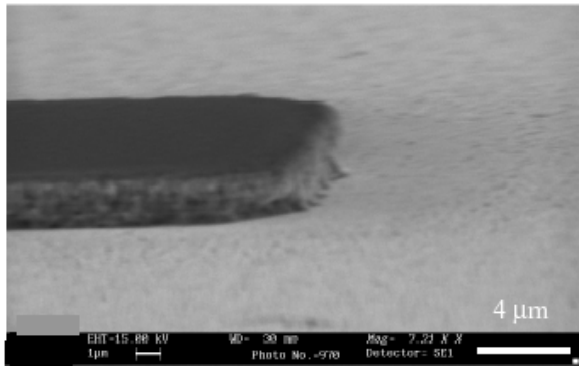


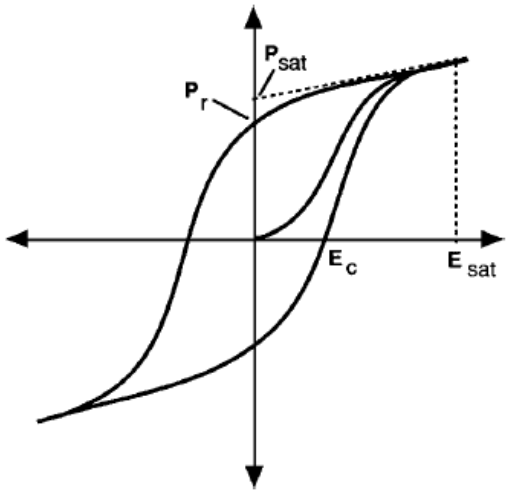
Fabrication: Frontside Processing



Fabrication: PZT Details

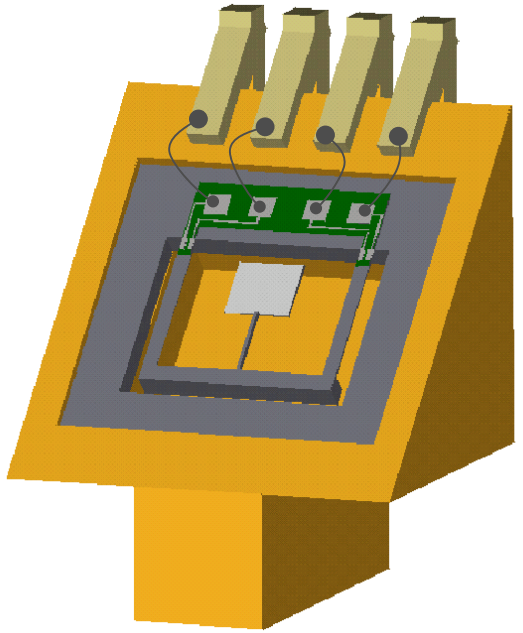
- Sol-gel deposition
  - Two step annealing for texture
- Etched in two step process
  - 10 HN_4F:1HF and 2HCl:H_2O
- Poled at elevated temperature with applied voltage
  - Increases effective d_{33}

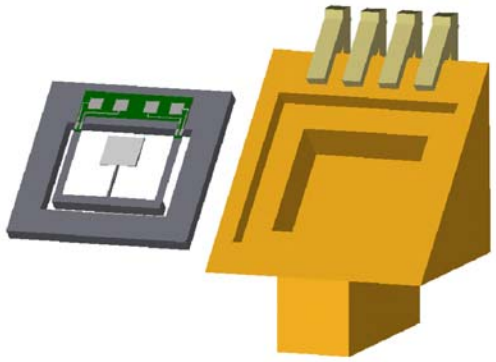




Device Packaging

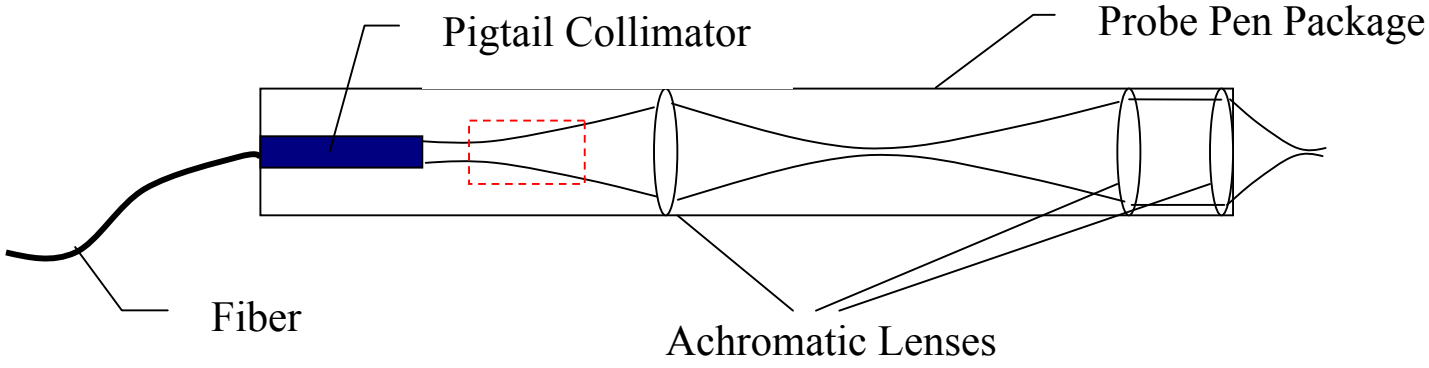
- Packaged in a molded plastic “seat”
- Inset patterned into package aligns device
- Gap below mirror to allow for deflection
- Secured with epoxy
- Contact pads wire bonded to lead frame





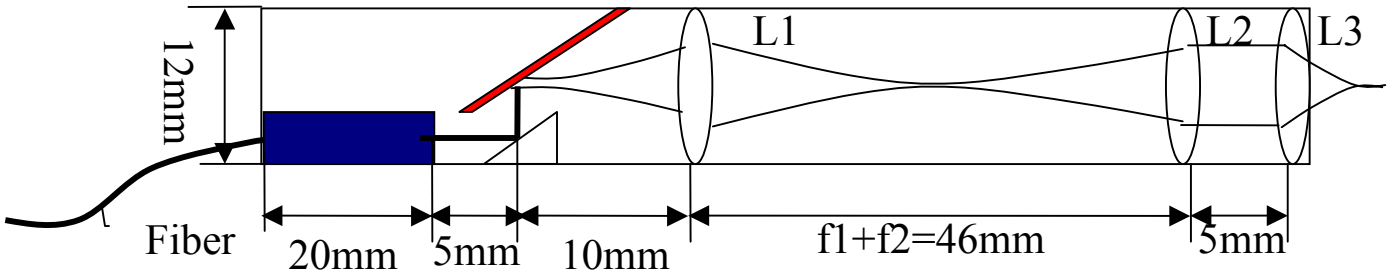
Optical Design Requirements

- Optics designed to achieve a small light spot at the working distance for probing tissue
- Design for optics should interface with the two-dimensional scanning MEMS device
  - Probe pen diameter < 15 mm, length ~ 150 mm
  - Spot size between 2-10 μm
  - Working distance of approximately 2 mm



Optics Analysis

Analysis method: Gaussian Beam modeling and ABCD matrix calculation



$$\frac{1}{q(z)} = \frac{1}{R(z)} - \frac{j\lambda}{\pi\omega^2(z)}$$

$$q_{out} = \frac{[q_{in}A + B]}{[q_{in}C + D]}$$

Waist of the beam from the collimator is w_0=0.2mm, divergence half angle is $\theta = 0.2^0$.

$$\frac{1}{R} = \frac{\theta}{\theta'} = \frac{f_2}{f_1}\sqrt{1+\left(\frac{\lambda l}{\pi w_0^2}\right)^2}$$

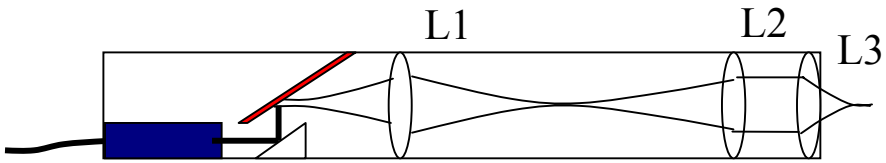
$$w = \frac{f_2}{f_1}(w + l\theta) = 0.526mm$$

$$R = \frac{0.526}{\frac{0.1}{180}\pi} = -301.4\text{mm}$$

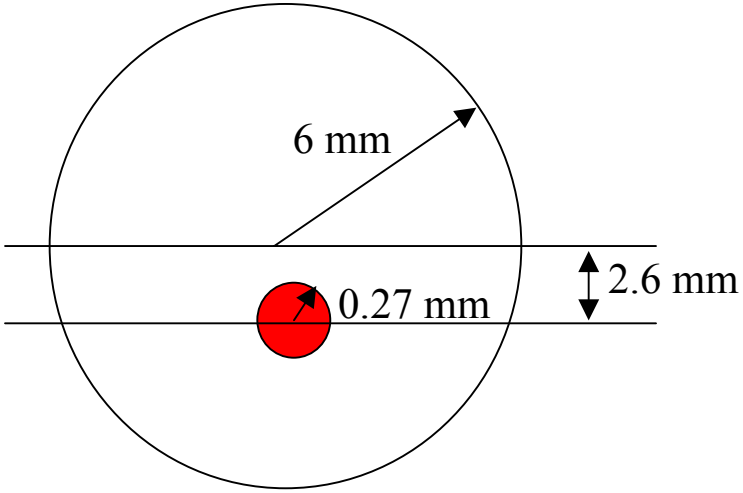
$$\frac{1}{q} = \frac{1}{R} - j\frac{\lambda}{\pi w^2} = 277.3 + 81.7i$$

Optics Analysis

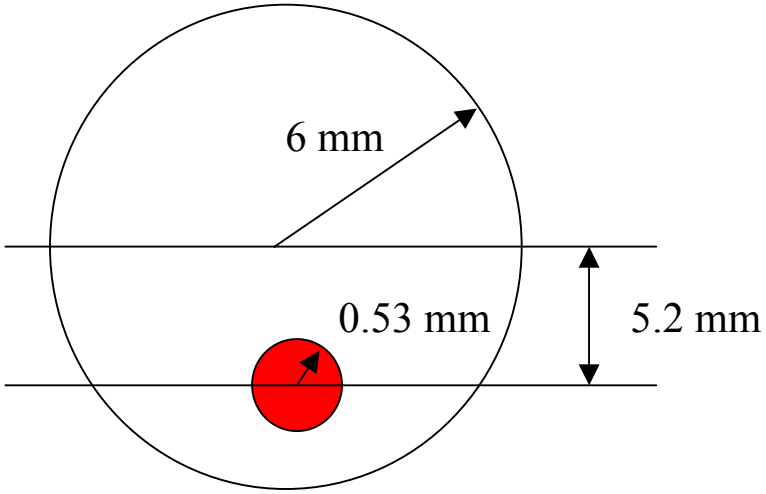
Working distance d=2.013mm
Spot size 2.02um



Ray tracing when beam tilted:



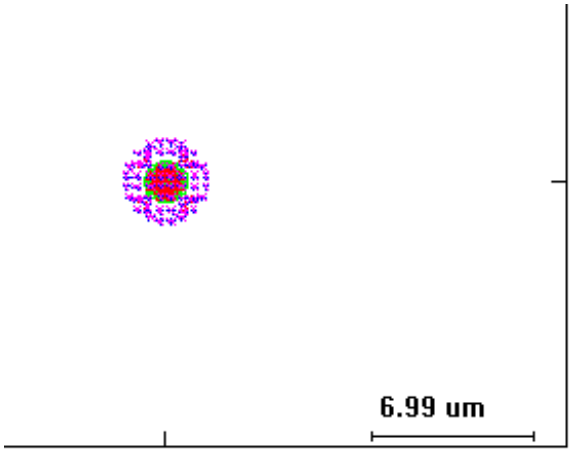
Spot at Lens 1



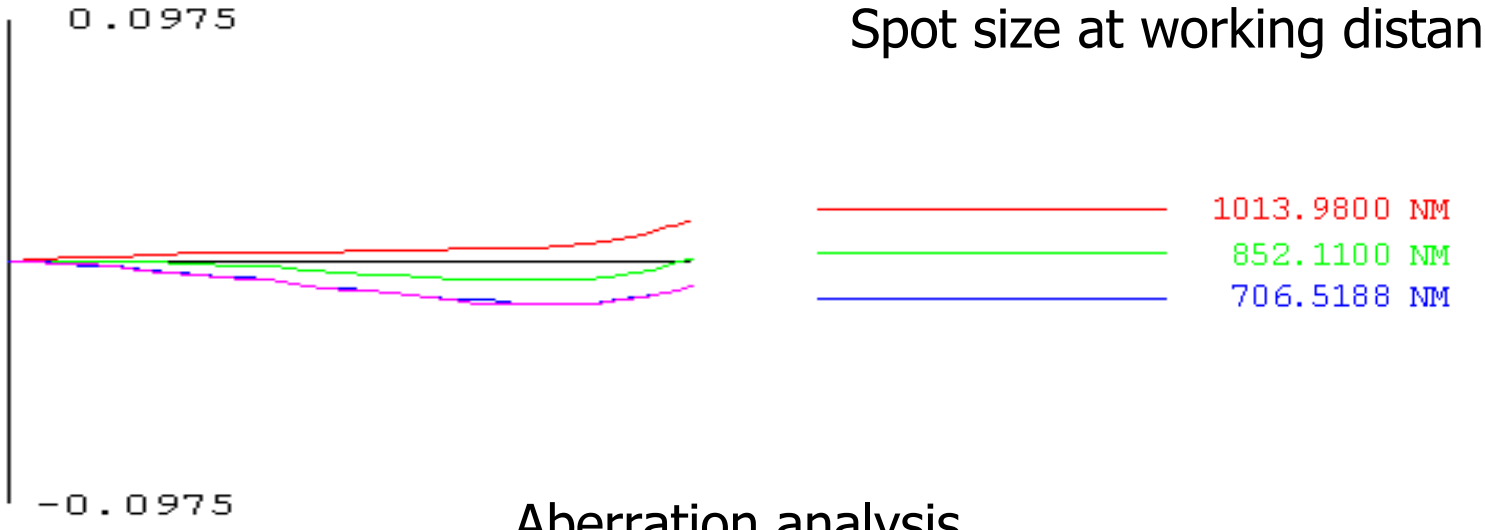
Spot at Lens 2

Optics Simulation

- Simulation Software – Code V
  - spot size & dispersion curve



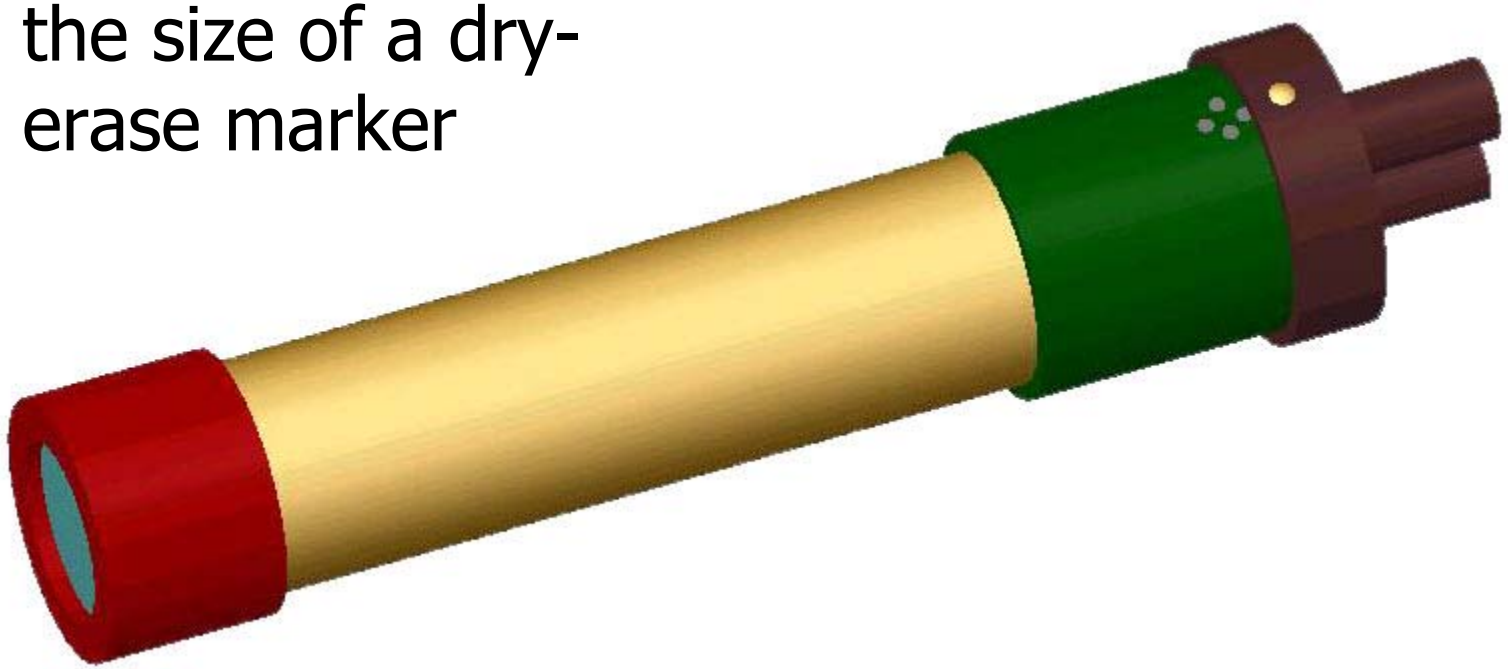
Spot size at working distance



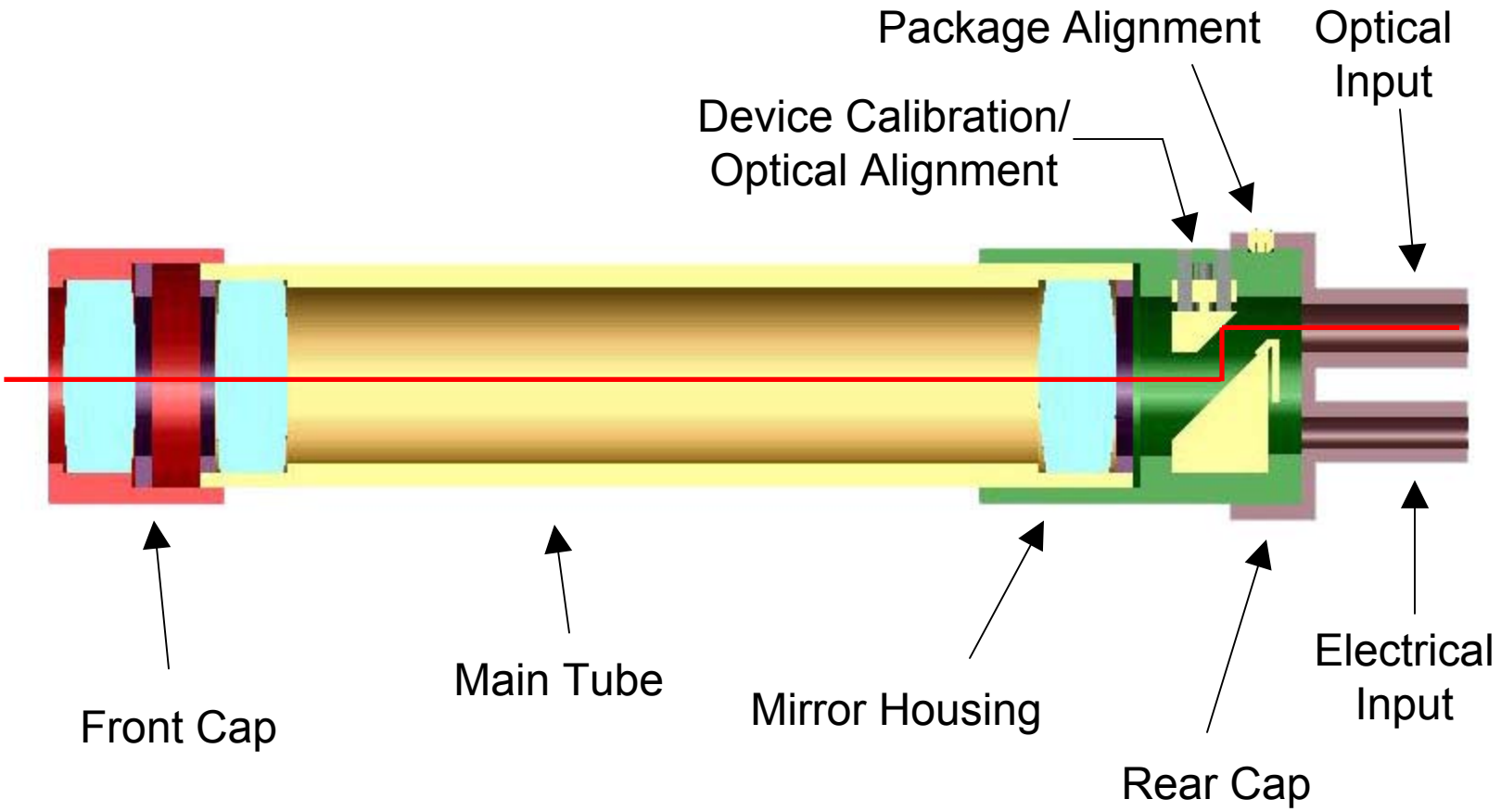
Aberration analysis

Mechanical Design

- Final design is about the size of a dry-erase marker

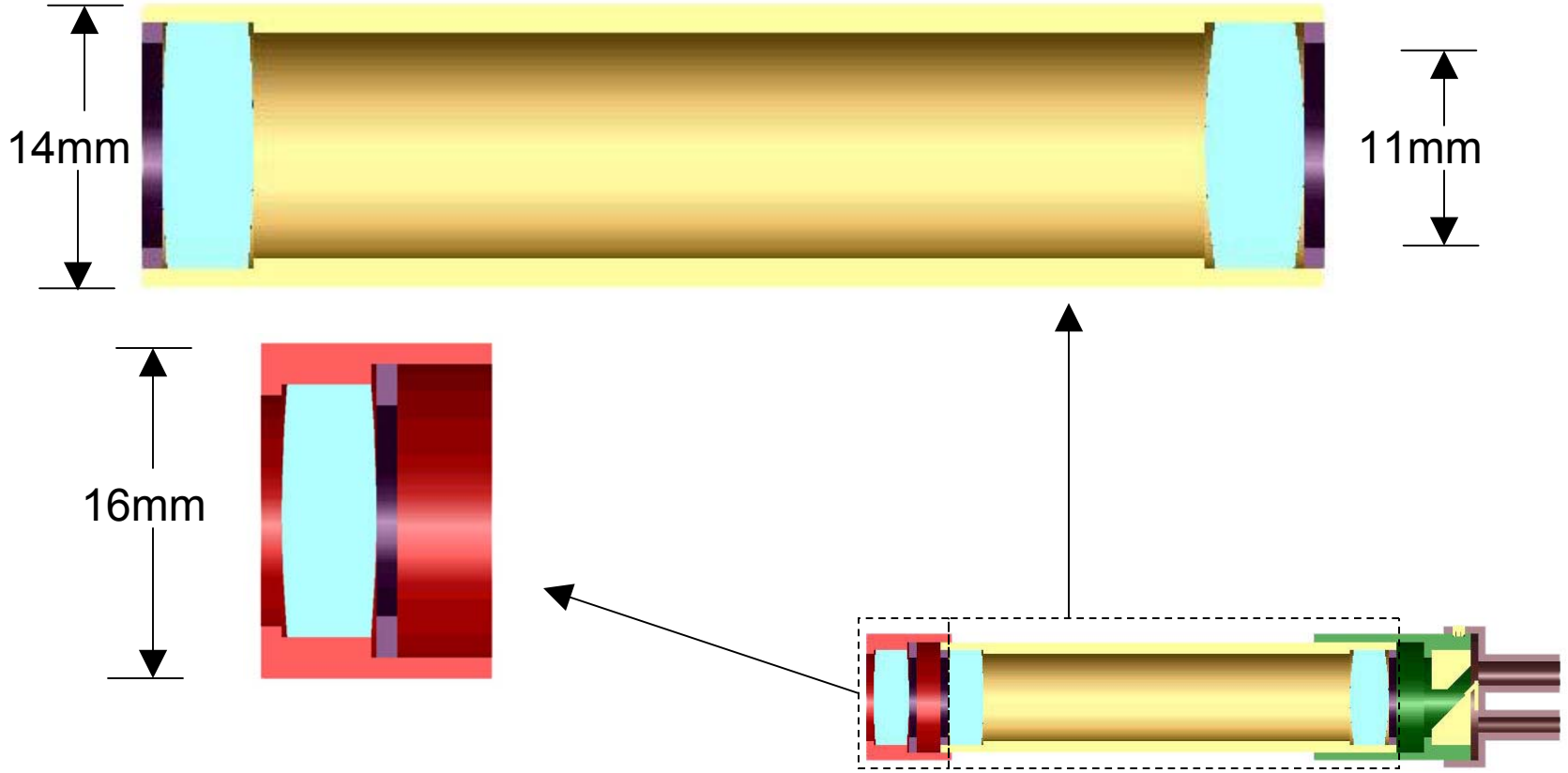


Mechanical Assembly

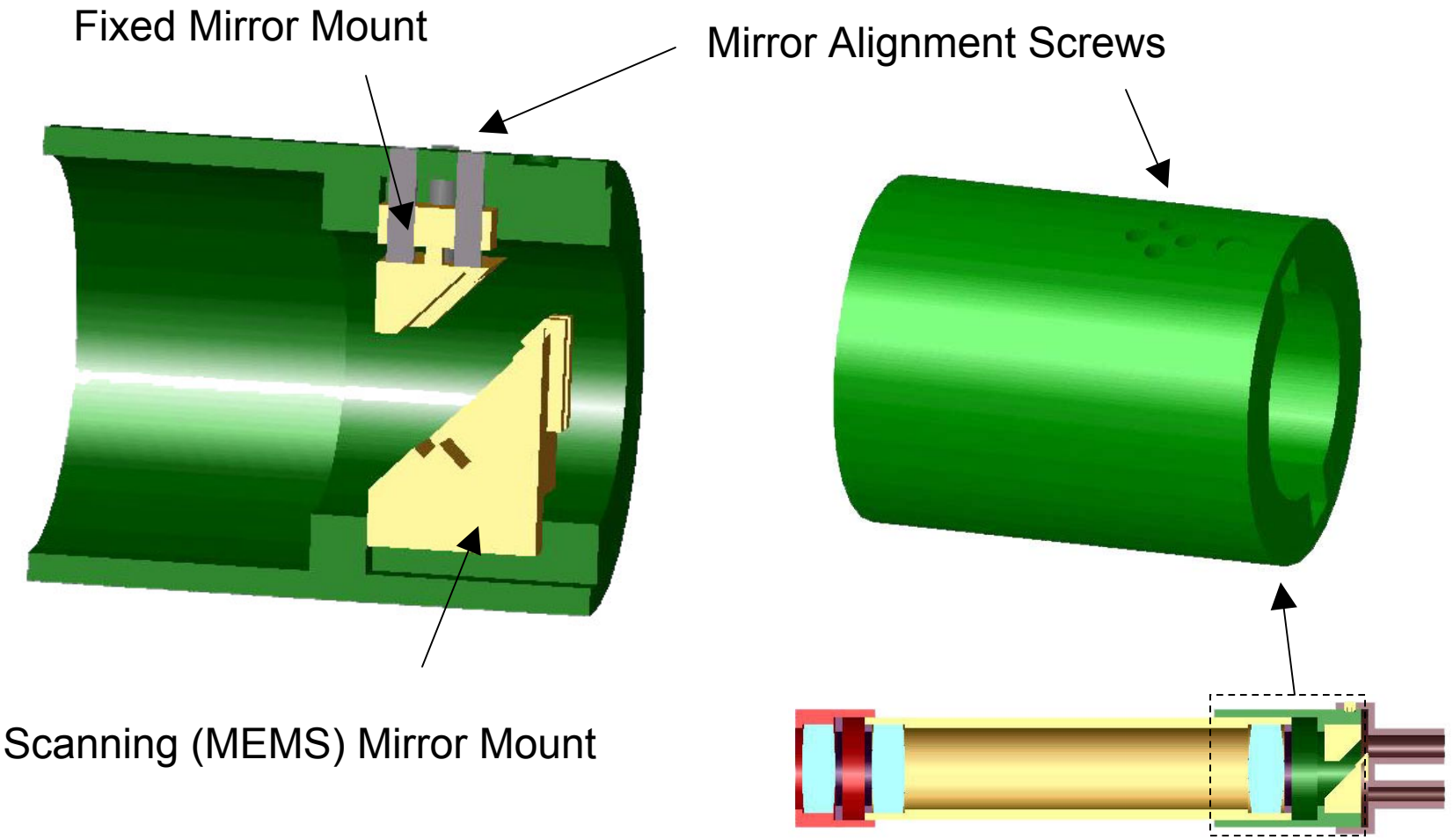


Front Cap, Main Tube

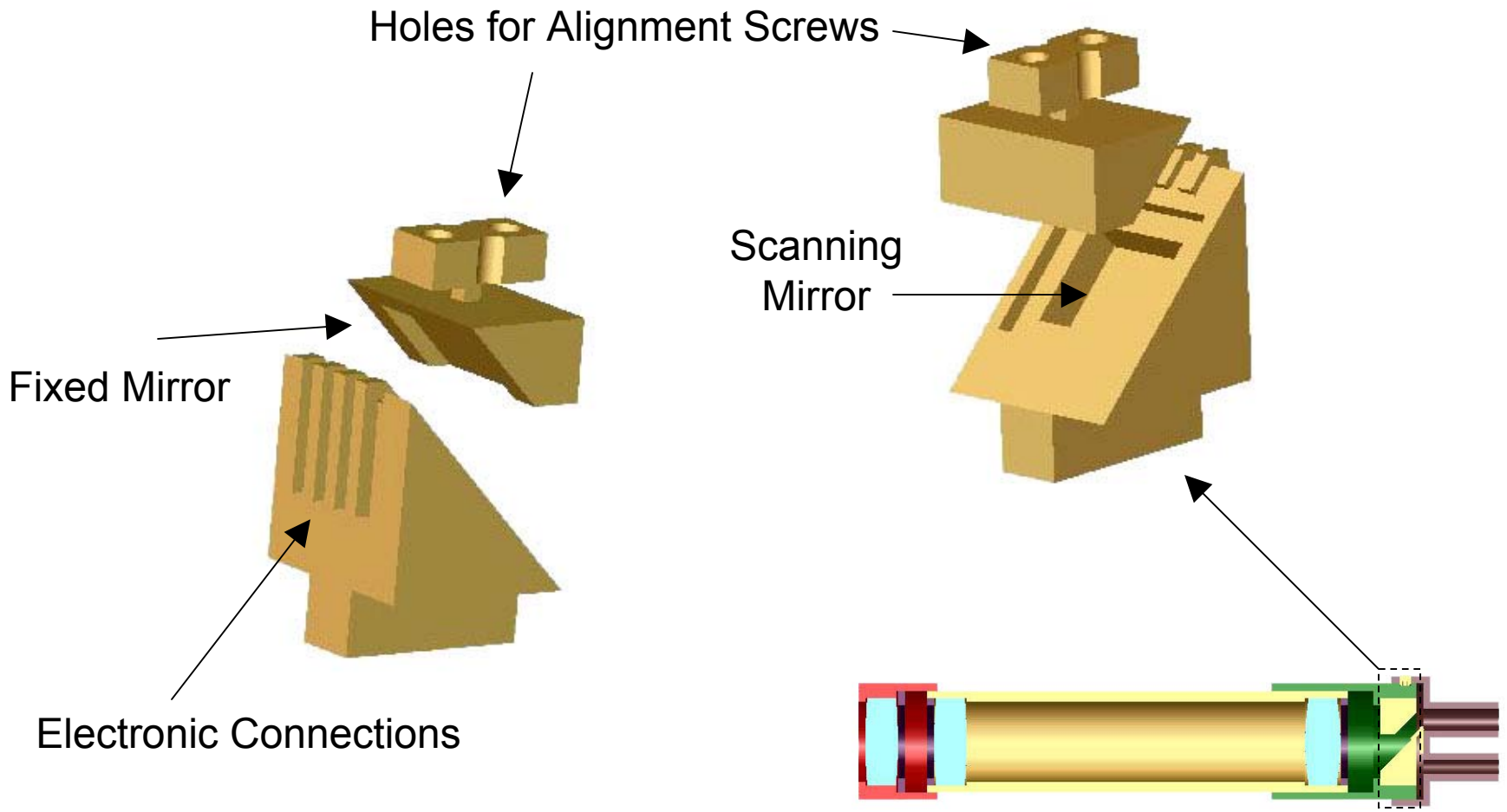
- Front cap and main tube each have threaded sections to secure lenses



Optical Calibration

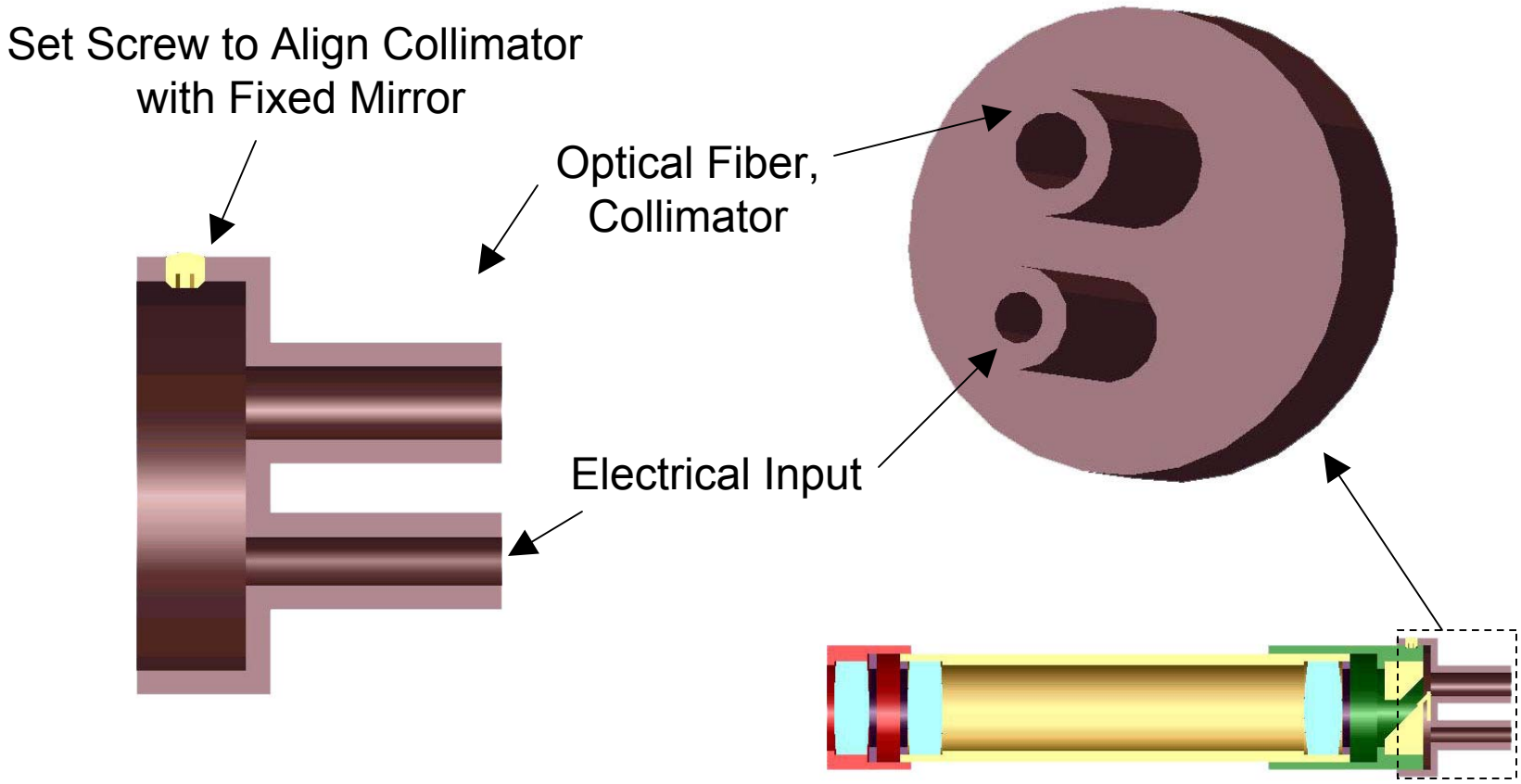


Mirror Mounts



Rear Cap

- Rear cap provides strain relief for the incoming optical fiber and electrical wire



Conclusions

- Probe size
  - OD: 16mm, ID: 11mm, Length: 90mm
- Scanning Mirror
  - Scan frequencies: 550Hz & 15.5 kHz
  - Scan angles: +/- 10 deg (mechanical)
- Aluminum scanner surface
- Spot size: 2.02 μm, Working distance: 2mm
- Power Requirement: 20 Volts, <1mA