Mechanical Mode Damping for Parametric Instability Control

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Motivation for focusing on mechanical mode Qs

- Recall: parametric gain is proportional to mechanical mode Q_m:

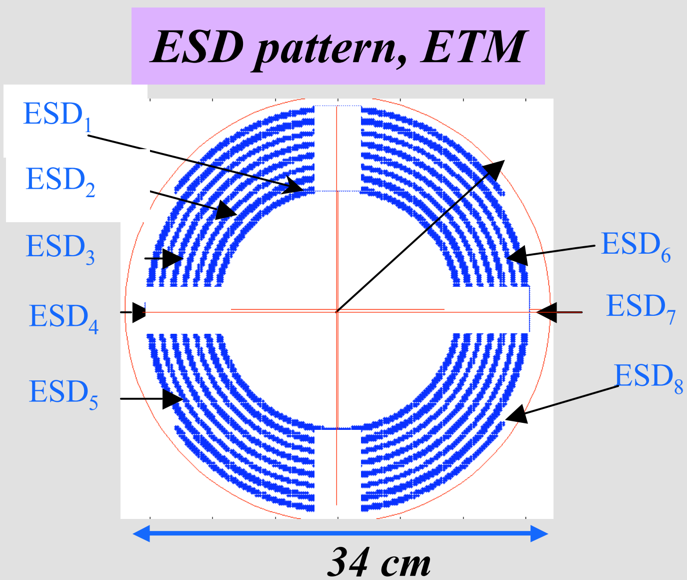
10-100 million

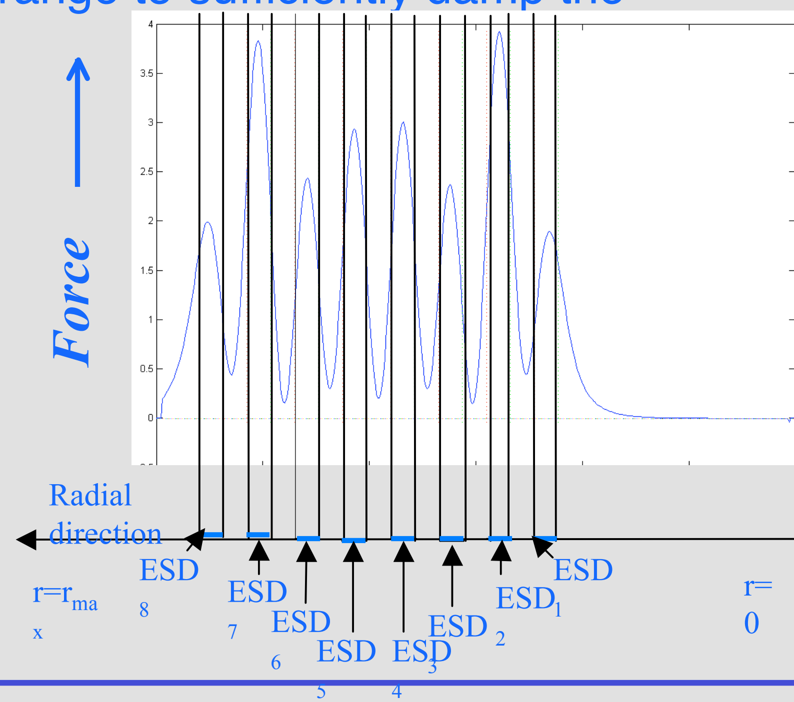
$$R = \frac{2P_{opt}Q_m}{ML\omega_m^2 c} \times \text{optical gain}$$

- Past work looked at 'broadband' dampers on the test mass barrel (Zhao et al., UWA)
  - Uniform barrel coating; localized damping ring around barrel
  - Not very satisfying effectiveness: thermal noise increased by 10% (100 Hz), but mode Qs still several million
- Want a more frequency selective approach
  - Active damping using the test mass actuators (electro-static drive)
  - Passive damping using added tuned mass dampers

Active damping with the electro-static drive (ESD)

- Basic idea:
  - sense the mechanical mode with the interferometer signal, apply a feedback damping force with the ESD
  - MIT ponderomotive experiment has a PI at 28kHz: stabilized with feedback to the mirror or the laser
- First question:
  - Does the ESD have enough range to sufficiently damp the mechanical modes?





ESD mode damping

- Required force:

mode freq. → ω_m^2; *modal mass* → M_m; *rms modal amplitude* → x_m^{rms} → **thermal excitation:**; *overlap w/ ESD* → Γ_m; *damped Q* → Q_m

$$F_{ESD} = \frac{\omega_m^2 \cdot M_m}{\Gamma_m \cdot Q_m} \cdot x_m^{rms} = \frac{\omega_m}{\Gamma_m \cdot Q_m} \cdot \sqrt{M_m k_B T}$$

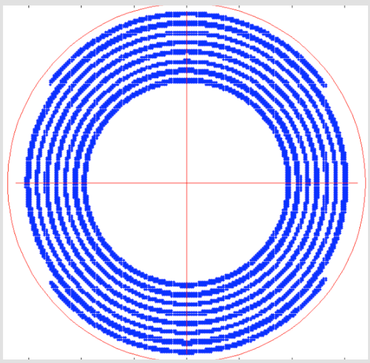
$$F_{ESD} = 400nN \left(\frac{f_m}{30\,\text{kHz}} \right) \left(\frac{10^5}{Q_m} \right) \left(\frac{10^{-3}}{\Gamma_m} \right) \left(\frac{M_m}{10\,\text{kg}} \right)^{1/2}$$

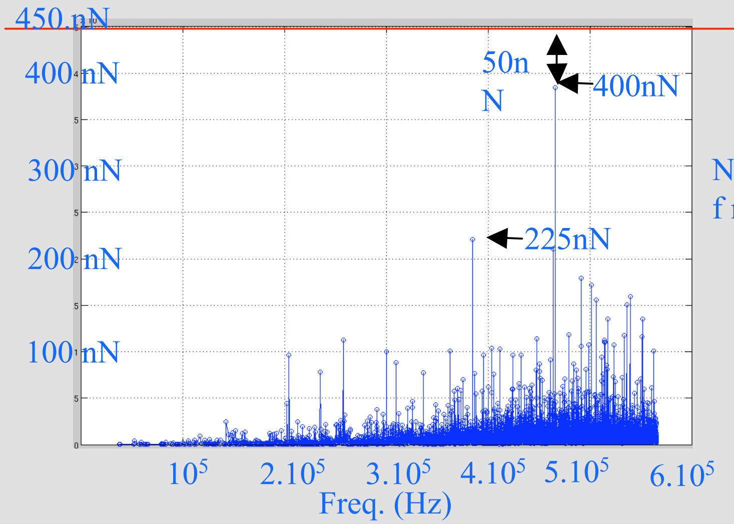
- Available ESD force
  - 200 micro-Newton peak, acquisition mode
  - Few micro-Newtons in low-noise mode

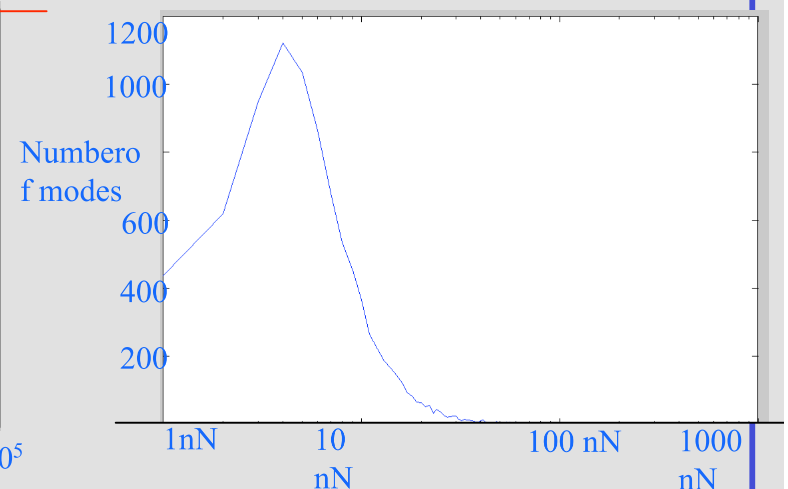
ESD mode damping, cont'd

Calculate overlap with all acoustic modes & force required to reduce Q to 200,000

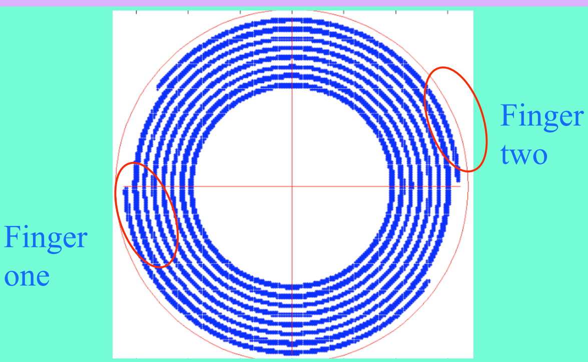
Nominal ESD pattern

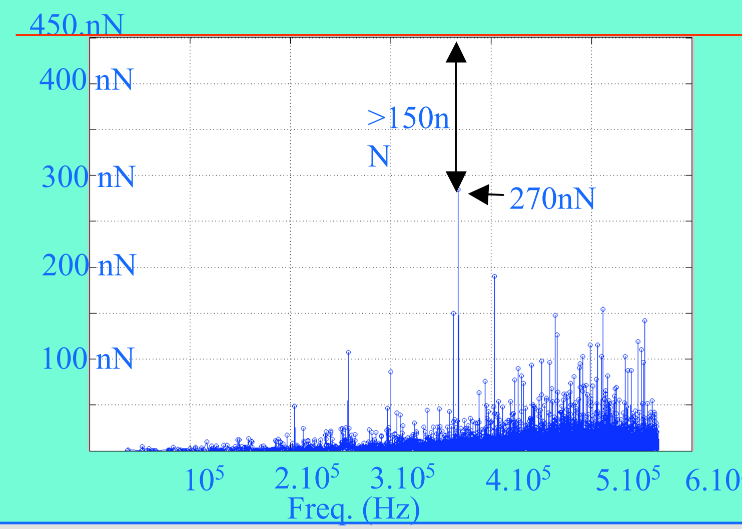




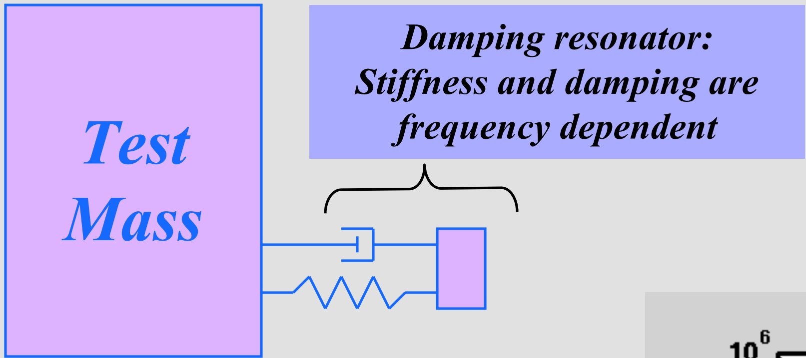


Slightly asymmetric ESD pattern

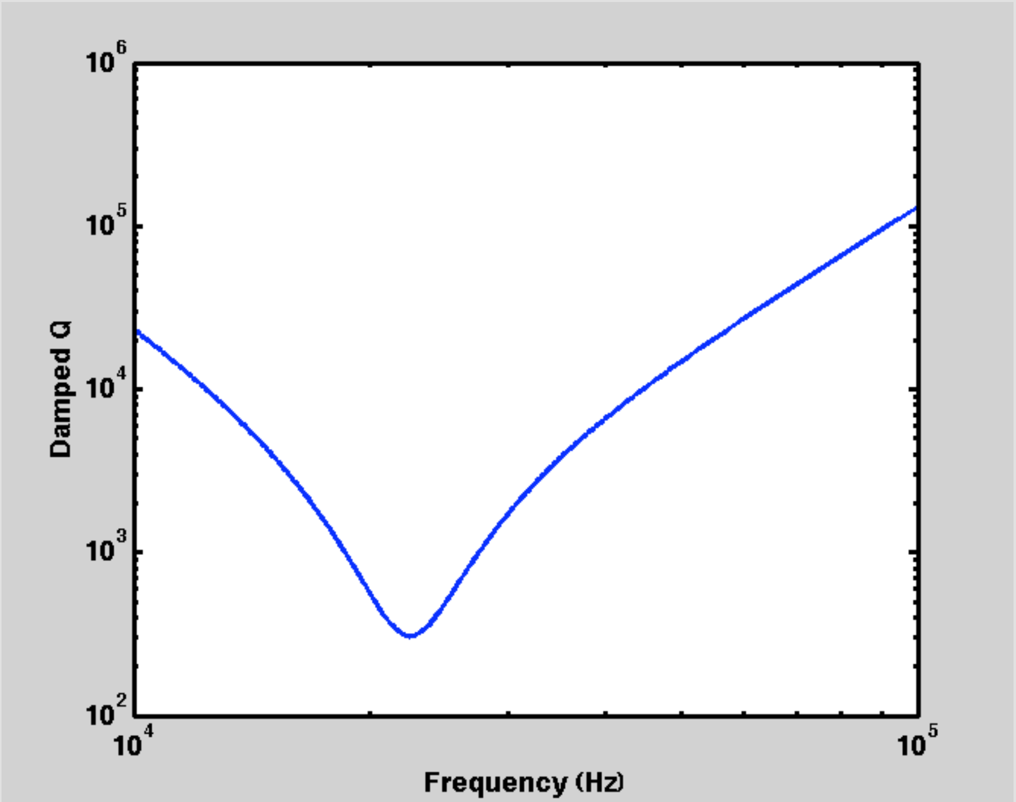




Passive mode damping

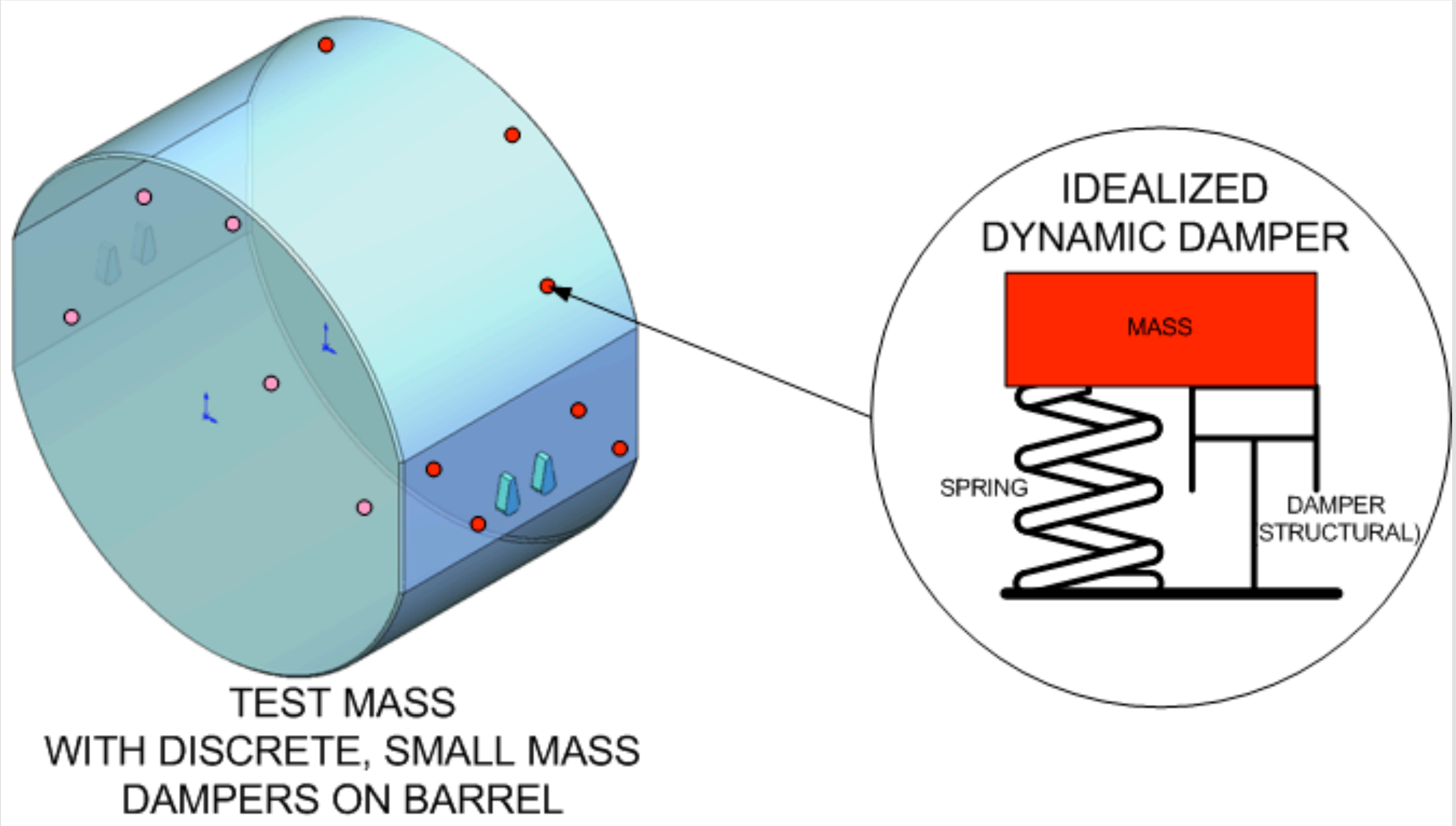


- Toy model to illstrate potential effectiveness
  - 1 DOF 'test mass', 1 kg
  - Damper mass: 1 gm
  - Damper tuned to 25 kHz
  - Damper is coupled 100% to test mass mode
  - Damper properties that of a resistively shunted piezo



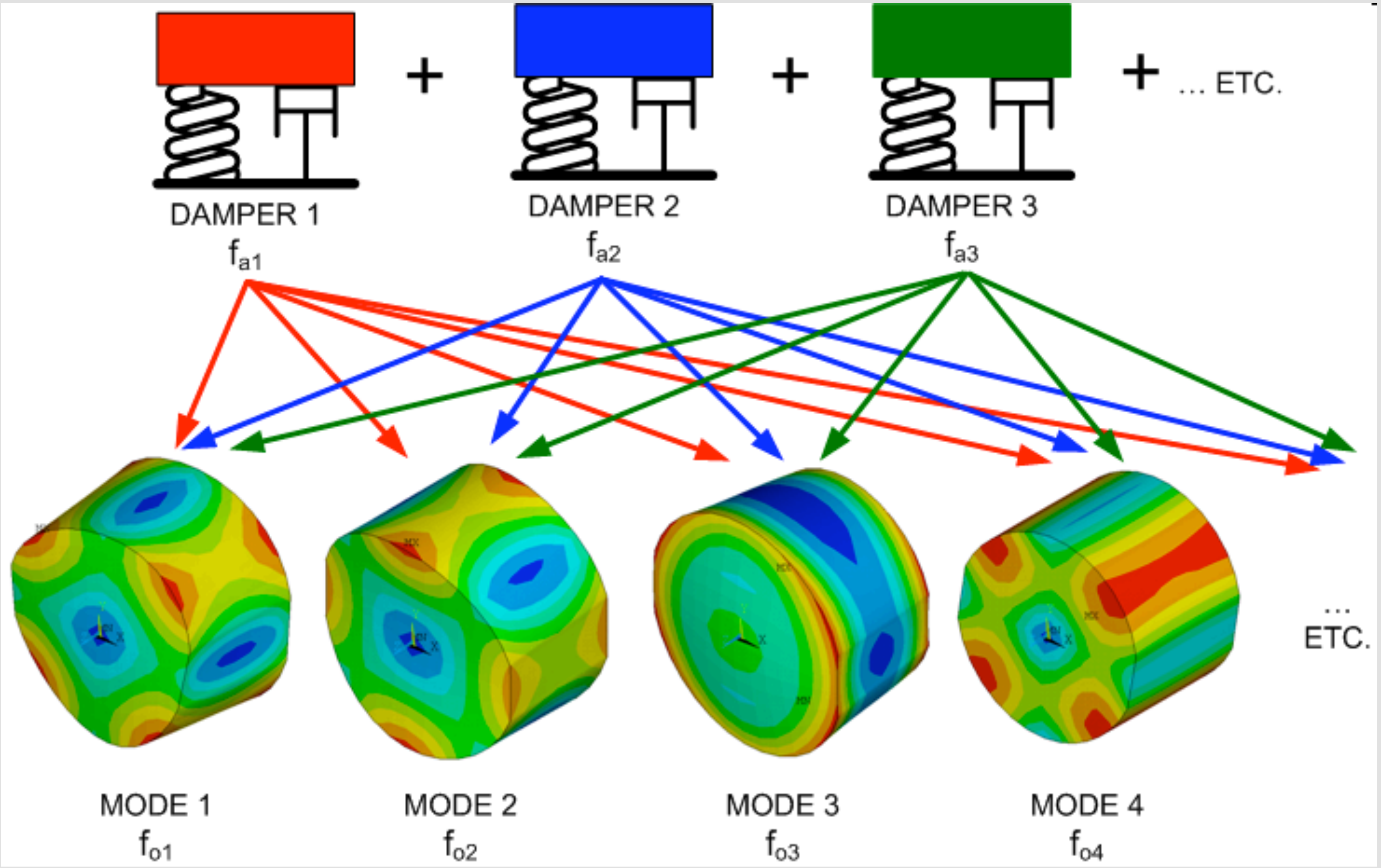
Dynamic Absorbers

- *Consider the addition of a number of discrete, idealized dynamic dampers to the Test Mass*



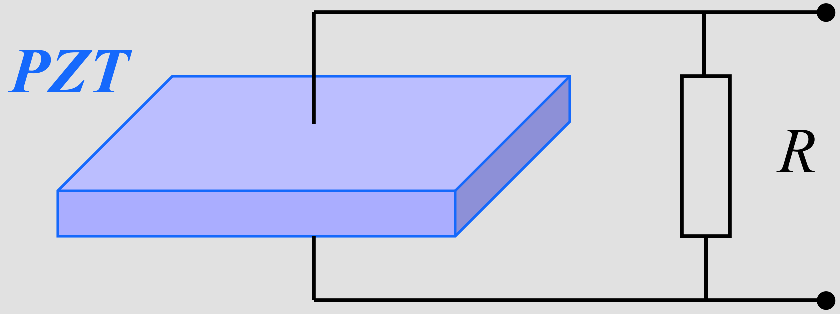
Dynamic Absorbers

- *The effect of the dynamic dampers can be addressed as the pairwise interaction of each damper and each eigenmode of the test mass*

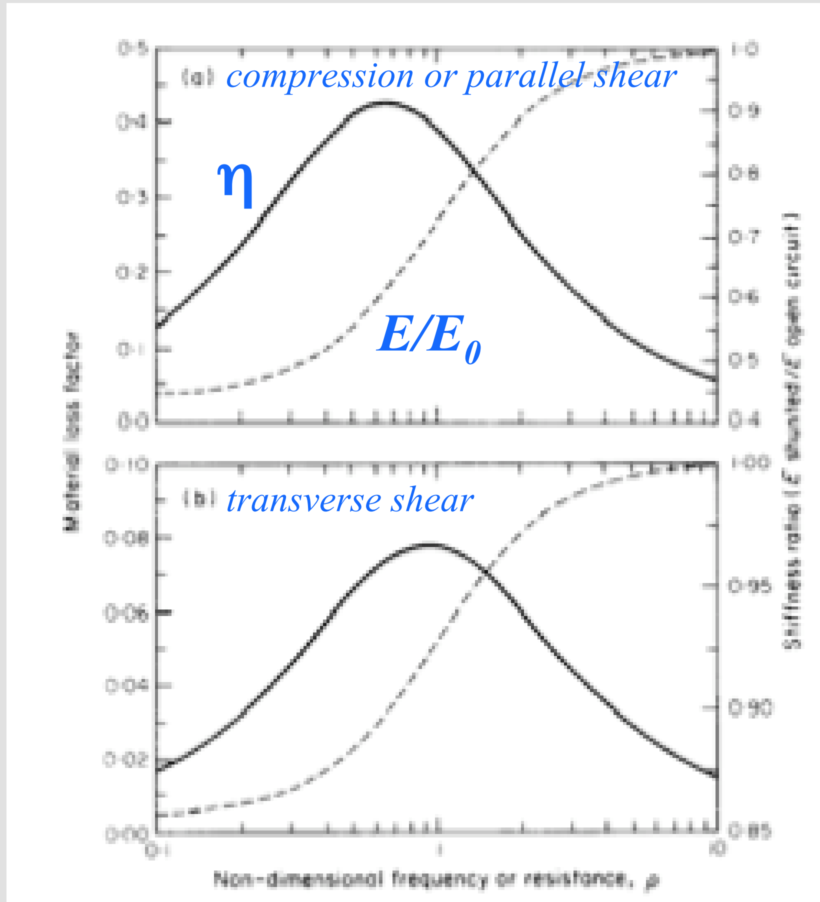


Resistively shunted piezoelectric damper

"Damping of structural vibrations with piezoelectric materials and passive electrical networks", Hagood and von Flotow, J Sound & Vib., 1991.



$$\rho = \omega / RC$$



Material loss factor:

$$\eta = \frac{\rho k_{ij}^2}{(1 - k_{ij}^2) + \rho^2}$$

Material stiffness:

$$\frac{E}{E_0} = 1 - \frac{k_{ij}^2}{1 + \rho^2}$$

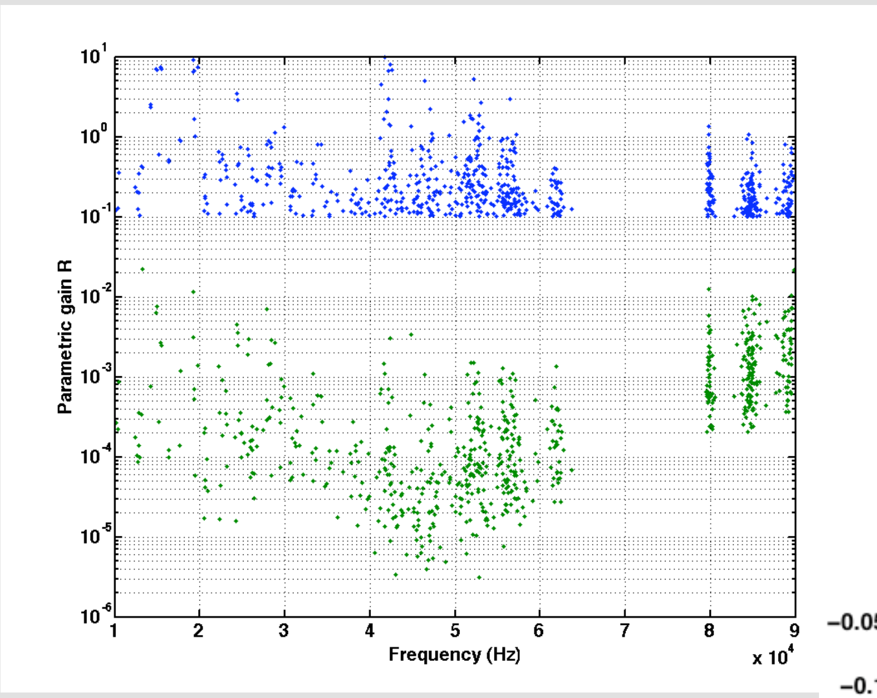
k^2 = (electrical energy stored / mechanical energy stored)

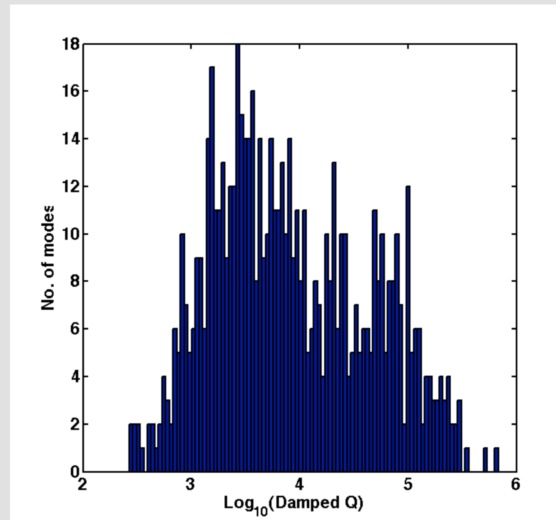
Selection of acoustical modes

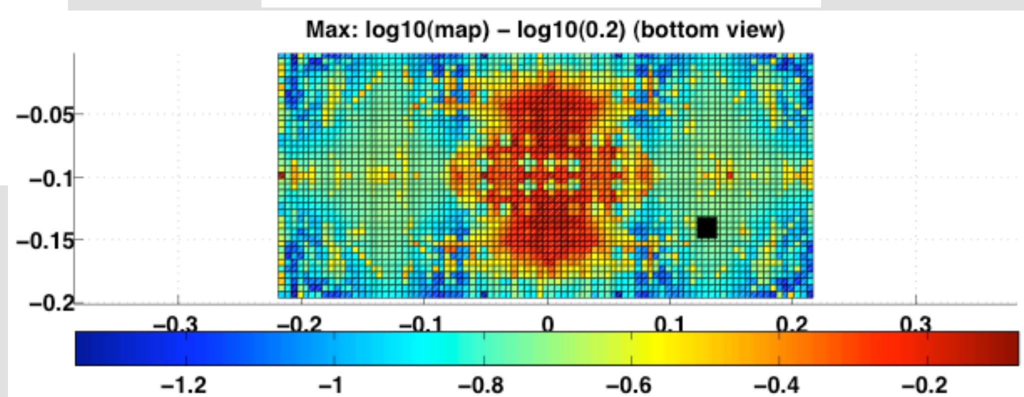
- Determine which acoustical modes might be problematic, so we don't always have to look at all ~10,000 modes between 10-100 kHz
- Calculate parametric gain R for a single arm cavity:
  - Include Hermite-Gauss modes up to order m+n=8
  - Approximate optical mode diffraction loss as 2x clipping loss
  - Artificially widen the cavity optical modes (but don't lower their Q) to account for uncertainty in mirror radii of curvature (used dR = +/- 10m)
  - Take acoustic mode Q = 10 million
  - Accept all modes with R greater than 0.1
  - End up with 675 modes between 10-90 kHz
    - Caveat: higher frequencies need to be redone with higher resolution FEA

Conceptual damper design

- Two piezo-dampers appears to be sufficient
  - Mounted on the barrel of the TM
  - Damper mass = 10 gm; f = 20 kHz & 50 kHz; $k^2 = 0.5$

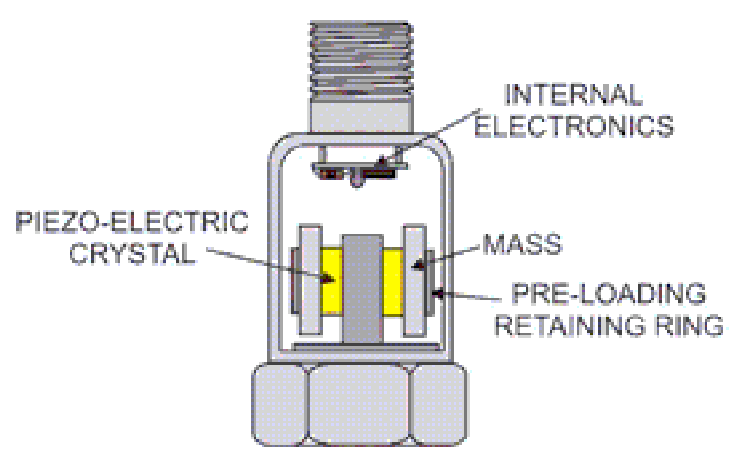






Piezo damper details

- Thermal noise impact
  - Combination of resonant design, loss function of the piezo, and the physical location, leads to negligible thermal noise impact due to piezo damping
  - More important with be TM surface strain energy coupling to damper materials and bonds: this needs to be estimated
- Practical design: it's essentially a piezo-electric accelerometer
  - Tri-axial sensitivity may be important
  - Need a rigid, vacuum-compatible structure



Shear type piezo-electric accelerometer

Plans

- Initial LIGO test mass and suspension test
  - Try damping a couple of ~10 kHz modes from a Q of a million to a ~100,000, using a piezo-damper (with a commercial accelerometer)
- LASTI test mass, suspended in the quad suspension with glass fibers (to be installed)
  - Measure internal mode Qs
  - Try damping Qs using the electro-static drive
  - Piezo-damper ??