EE C245 – ME C218
Introduction to MEMS Design
Fall 2007

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Lecture 19: Resonance Frequency II

Lecture Outline

- Reading: Senturia, Chpt. 10
- Lecture Topics:
  - Energy Methods
    - Virtual Work
    - Energy Formulations
    - Tapered Beam Example
    - Estimating Resonance Frequency

The Raleigh-Ritz Method

- Equate the maximum potential and maximum kinetic energies:

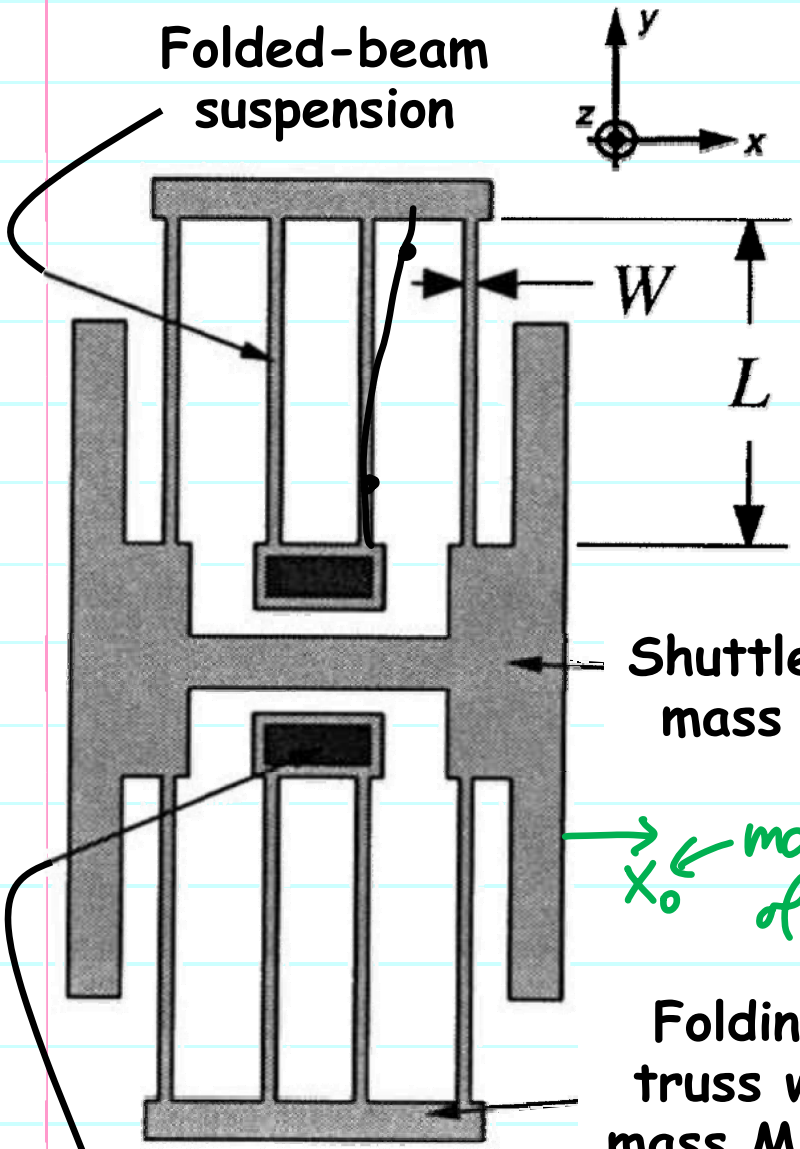
$$K_{\max} = \int_0^L \frac{1}{2}\rho W h \omega^2 \hat{y}^2(x)dx = \mathcal{W}_{\max}$$

- Rearranging yields for resonance frequency:

$$\boxed{\omega = \sqrt{\frac{\mathcal{W}_{\max}}{\int_0^L \frac{1}{2}\rho W h \hat{y}^2(x)dx}}}$$

ω = resonance frequency
$\mathcal{W}_{max}$ = maximum potential energy
ρ = density of the structural material
W = beam width
h = beam thickness
$\hat{y}(x)$ = resonance mode shape

Example: Folded-Beam Resonator



- Derive an expression for the resonance frequency of the folded-beam structure at left.

Use Rayleigh-Ritz method:

$$KE_{max} = PE_{max}$$

Kinetic Energy:

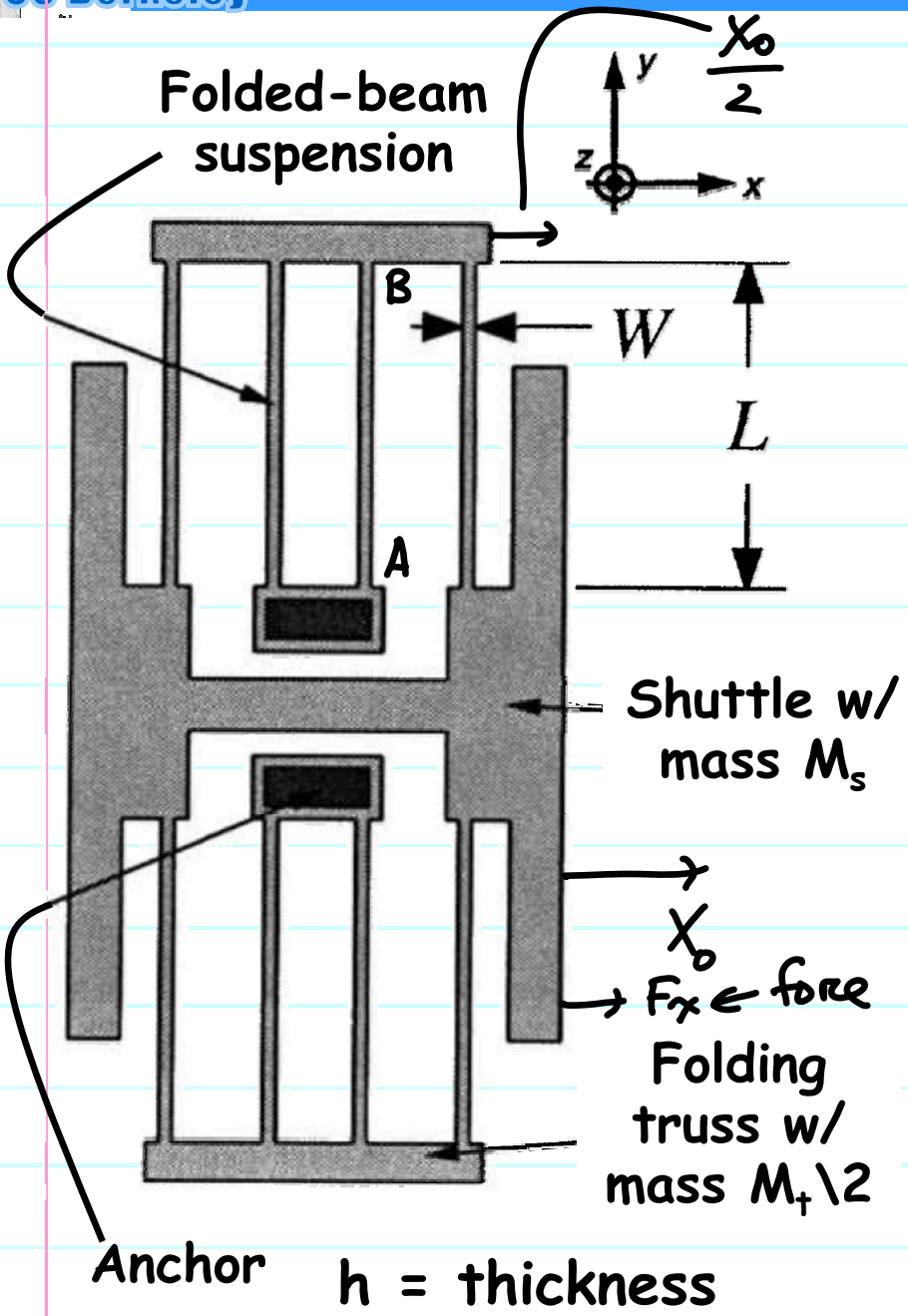
$$KE_{max} = \underbrace{KE_s}_{\text{shuttle}} + \underbrace{KE_t}_{\text{trusses}} + \underbrace{KE_b}_{\text{beams}}$$

$$= \frac{1}{2} v_s^2 M_s + \frac{1}{2} v_t^2 M_t + \underbrace{\frac{1}{2}\int v_b^2 \, dM_b}$$

mass of both trusses

Must integrate since the beam velocity is a function of location y!

Get Kinetic Energies



Velocity of the shuttle: $v_s = \omega_o X_o$

Resonance Freq. → ω_o; Maximum Displacement Amplitude → X_o

ρV

$$\therefore KE_s = \frac{1}{2} v_s^2 M_s = \frac{1}{2}\omega_o^2 X_o^2 M_s$$

Velocity of the truss: $v_t = \frac{1}{2} v_s = \frac{1}{2}\omega_o X_o$

$$\therefore KE_t = \frac{1}{2}\left(\frac{1}{2}\omega_o X_o\right)^2 M_t = \frac{1}{8}\omega_o^2 X_o^2 M_t$$

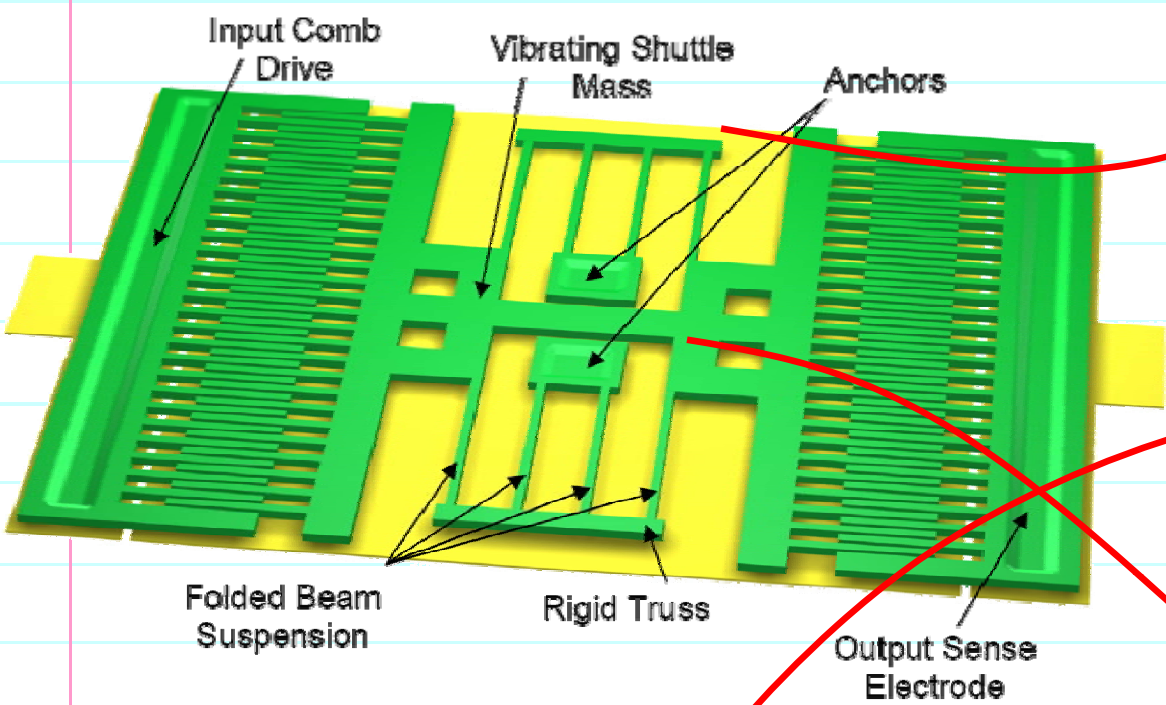
Velocity of the beam segments:

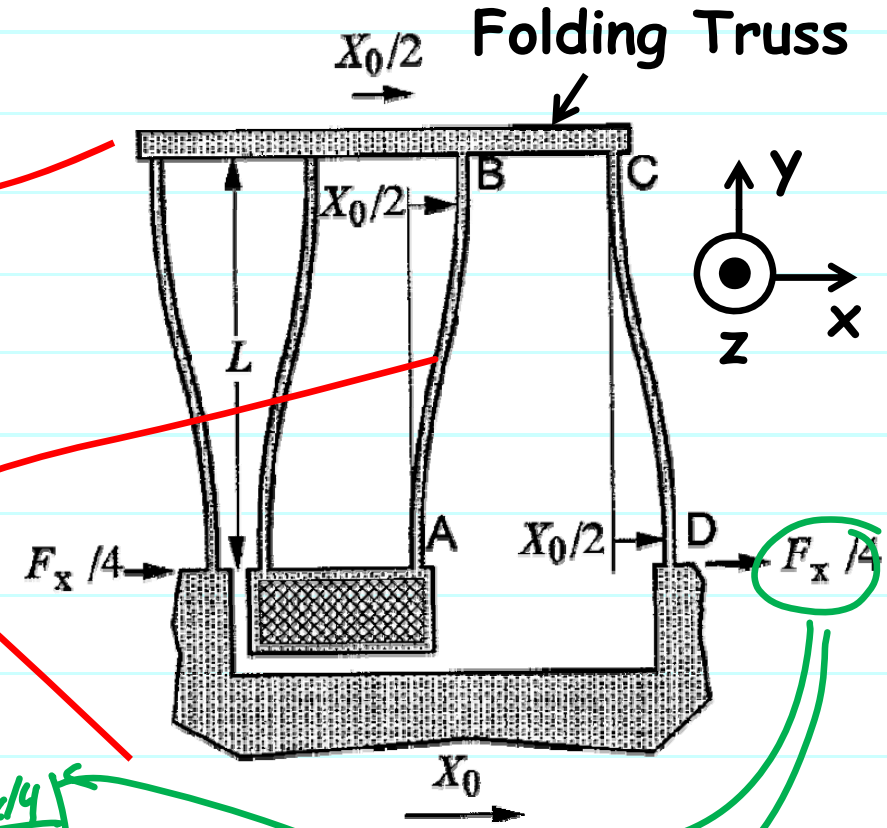
⇒ assume the mode shape is the same as the static displacement shape

⇒ For segment AB

$$\hat{x}(y) = \frac{F_x}{48EI_z}\left(3Ly^2 - 2y^3\right) \quad 0 \le y \le L \qquad (1)$$

Folded-Beam Suspension





Comb-Driven Folded Beam Actuator

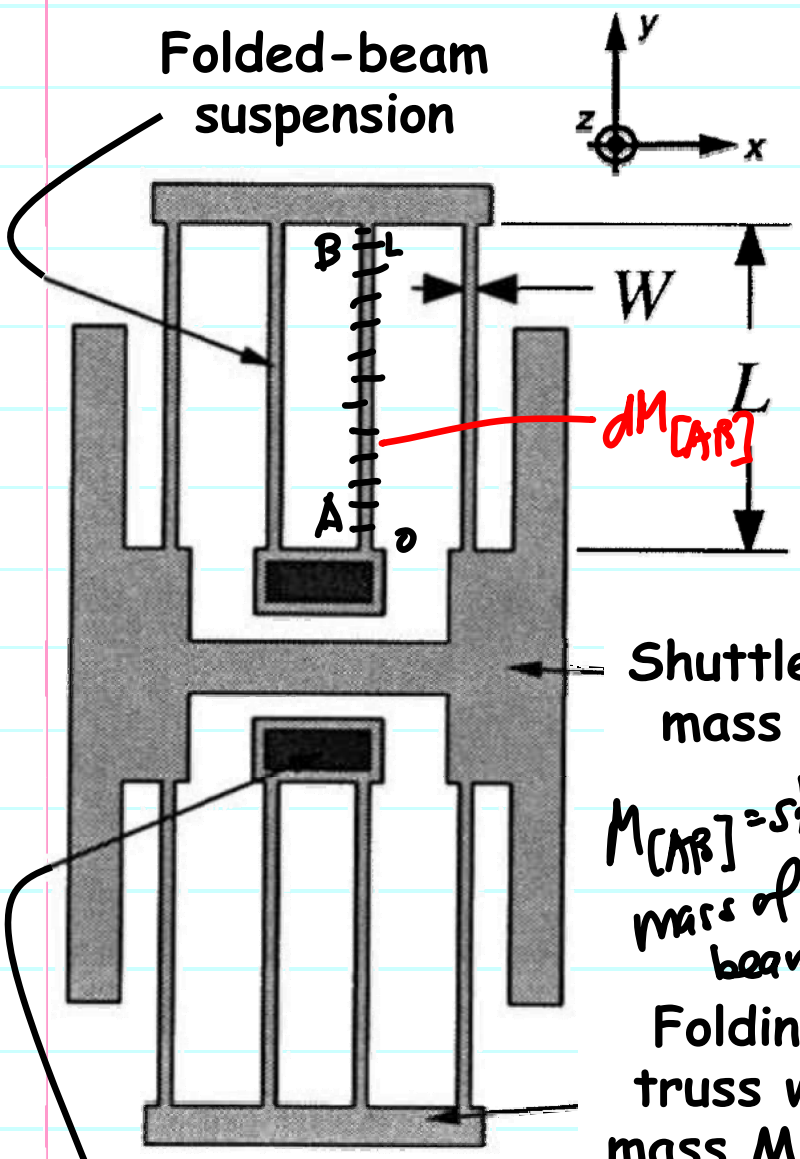
$$\hat{x}(y) = \frac{F_x}{48EI_z}\left(3Ly^2 - 2y^3\right) \qquad 0 \le y \le L$$

$$\left(\frac{F_x/4}{12EI_z}\right)$$

Case: $y=0$ $\quad \hat{x}(y=0) = 0$ ✓

Case: $y=L_c$ $\quad \hat{x}(y=L_c) = \frac{F_x}{48EI_z}L^3 \rightarrow k = \frac{(F_x/4)}{x} = \frac{12EI_z}{L^3} = \frac{k_c}{2}$ ✓

Get Kinetic Energies (cont)



h = thickness

At $y = L$: $x(L) = \frac{X_0}{2} = \frac{F_x L^3}{48 E I_z}$

Substituting into (1):

$$\hat{x}(y) = \frac{X_0}{2}\left[3\left(\frac{y}{L}\right)^2 - 2\left(\frac{y}{L}\right)^3\right]$$

which yields for velocity:

$$v_b(y)\Big|_{[AB]} = \frac{X_0}{2}\left[3\left(\frac{y}{L}\right)^2 - 2\left(\frac{y}{L}\right)^3\right]\omega_0$$

Plugging into the expression for KE_b:

$$KE_{[AB]} = \frac{1}{2}\int_0^L \frac{X_0^2\omega_0^2}{4}\left[3\left(\frac{y}{L}\right)^2 - 2\left(\frac{y}{L}\right)^3\right]^2 dM_{[AB]}$$

$M_{[AB]}$ = static mass of the beam

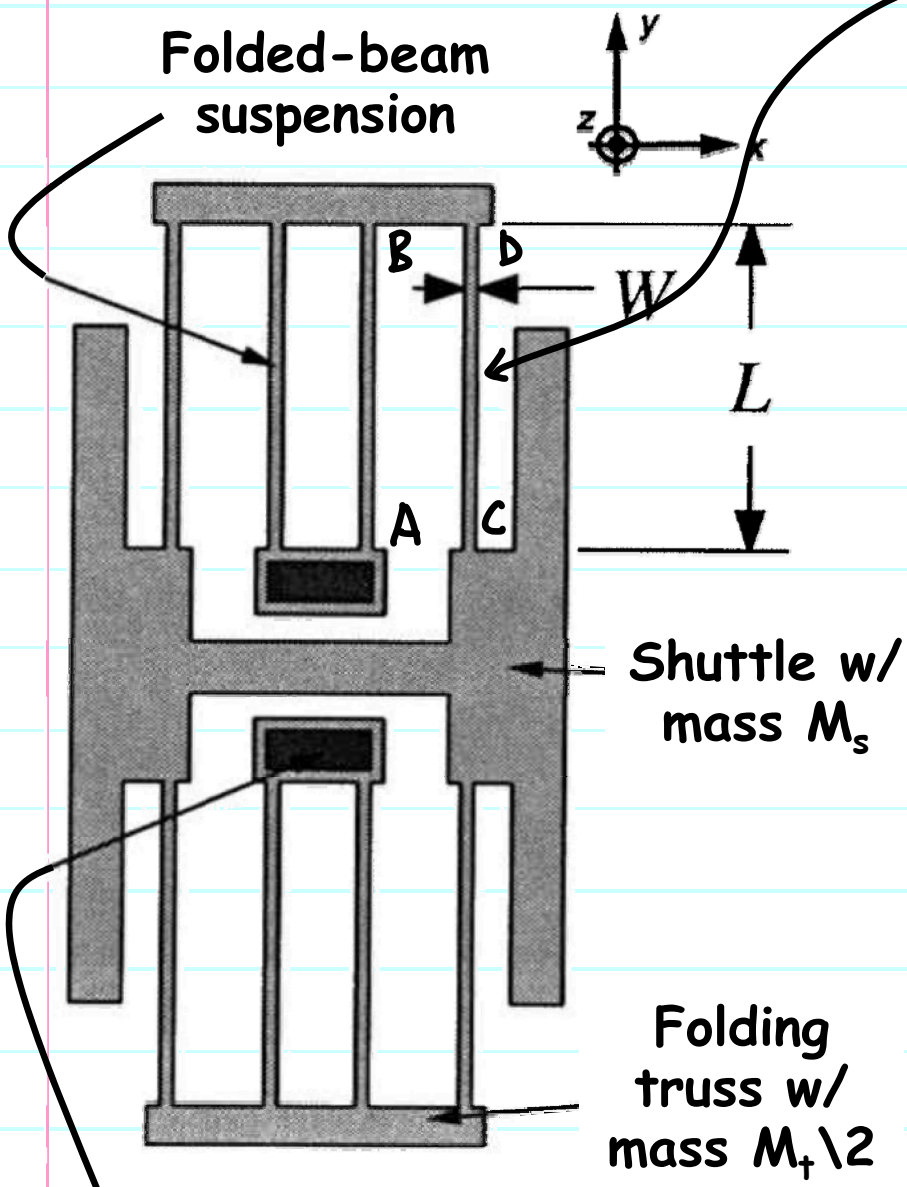
[uniform material] $\Rightarrow dM_{[AB]} = \frac{M_{[AB]}}{L}dy$

ρV = density · volume

$$= \frac{X_0^2\omega_0^2 M_{[AB]}}{8L}\int_0^L \left[3\left(\frac{y}{L}\right)^2 - 2\left(\frac{y}{L}\right)^3\right]^2 dy$$

$$\Rightarrow KE_{[AB]} = \frac{13}{280}X_0^2\omega_0^2 M_{[AB]}$$

Get Kinetic Energies (cont)



For segment CD:

$$v_b(y)\big|_{[CD]} = X_0\left[1 - \frac{3}{2}\left(\frac{y}{L}\right)^2 + \left(\frac{y}{L}\right)^3\right]\omega_0$$

Thus:

$$KE_{[CD]} = \frac{X_0^2\omega_0^2 M_{[CD]}}{2L}\int_0^L \left[1 - \frac{3}{2}\left(\frac{y}{L}\right)^2 + \left(\frac{y}{L}\right)^3\right]dy$$

$$KE_{[CD]} = \frac{83}{280}X_0^2\omega_0^2 M_{[CD]}$$

Static mass of beam [CD]

Let $M_b \triangleq$ total mass of the 8 beams.

Then: $M_{[AB]} = M_{[CD]} = \frac{1}{8}M_b$

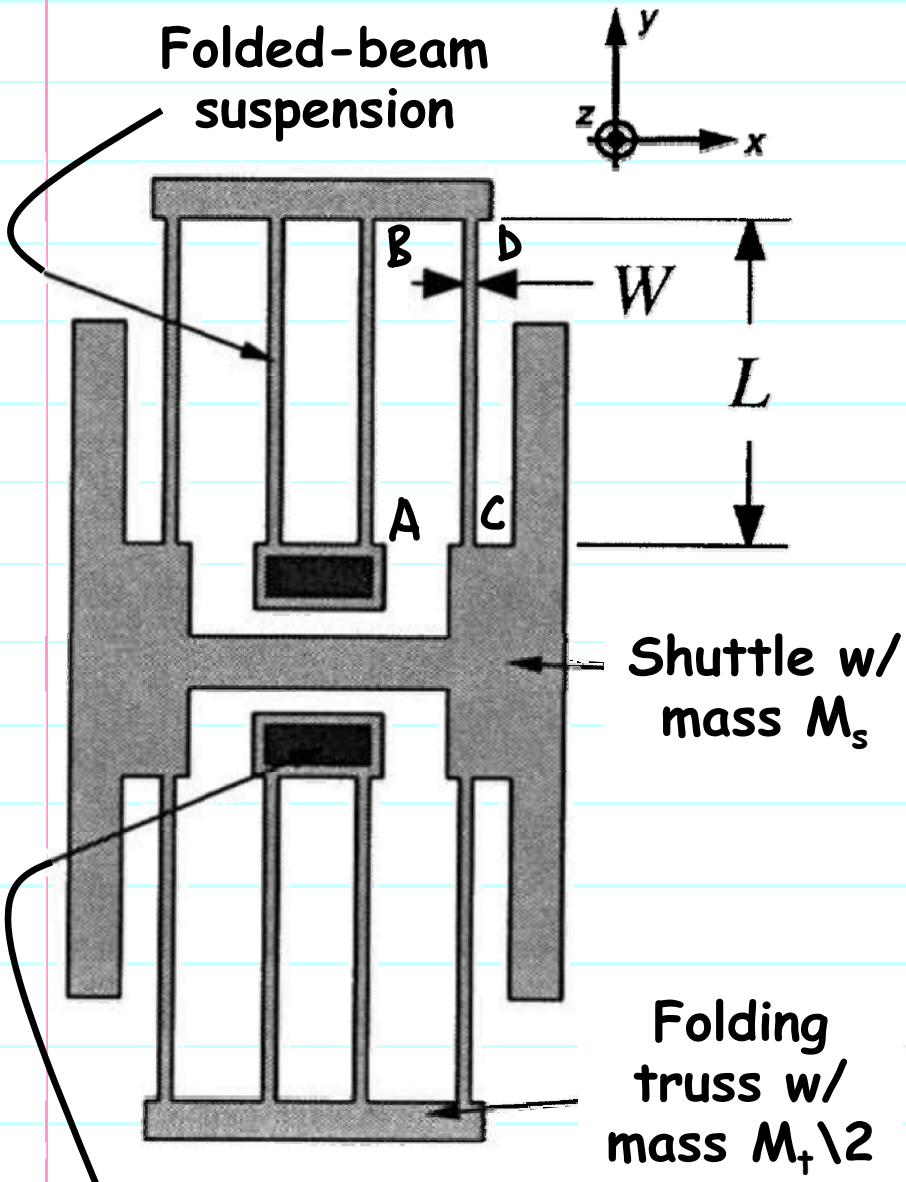
Thus:

$$KE_b = 4KE_{[AB]} + 4K_{[CD]} = \frac{6}{35}X_0^2\omega_0^2 M_b$$

and

$$KE_{max} = X_0^2\omega_0^2\left[\frac{1}{2}M_s + \frac{1}{8}M_t + \frac{6}{35}M_b\right]$$

Get Potential Energy & Frequency



PE_{max} is simply the work done to achieve maximum deflection:

$$PE_{max} = \frac{1}{2} \overset{=k_c}{k_x} X_0^2$$

Thus, using Raleigh-Ritz:

$$KE_{max} = PE_{max}$$

$$\cancel{X_0^2}\omega_0^2\left[\frac{1}{2}M_s + \frac{1}{8}M_t + \frac{6}{35}M_b\right] = \frac{1}{2}\overset{=k_c}{k_x}\cancel{X_0^2}$$

$$\boxed{\omega_0 = \left[\frac{\overset{=k_c}{k_x}}{M_{eq}}\right]^{1/2} \quad \text{where } M_{eq} = M_s + \frac{1}{4}M_t + \frac{12}{35}M_b}$$

(Resonance Frequency of a Folded-Beam Suspended Shuttle)

Brute Force Methods for Resonance Frequency Determination

Basic Concept: Scaling Guitar Strings

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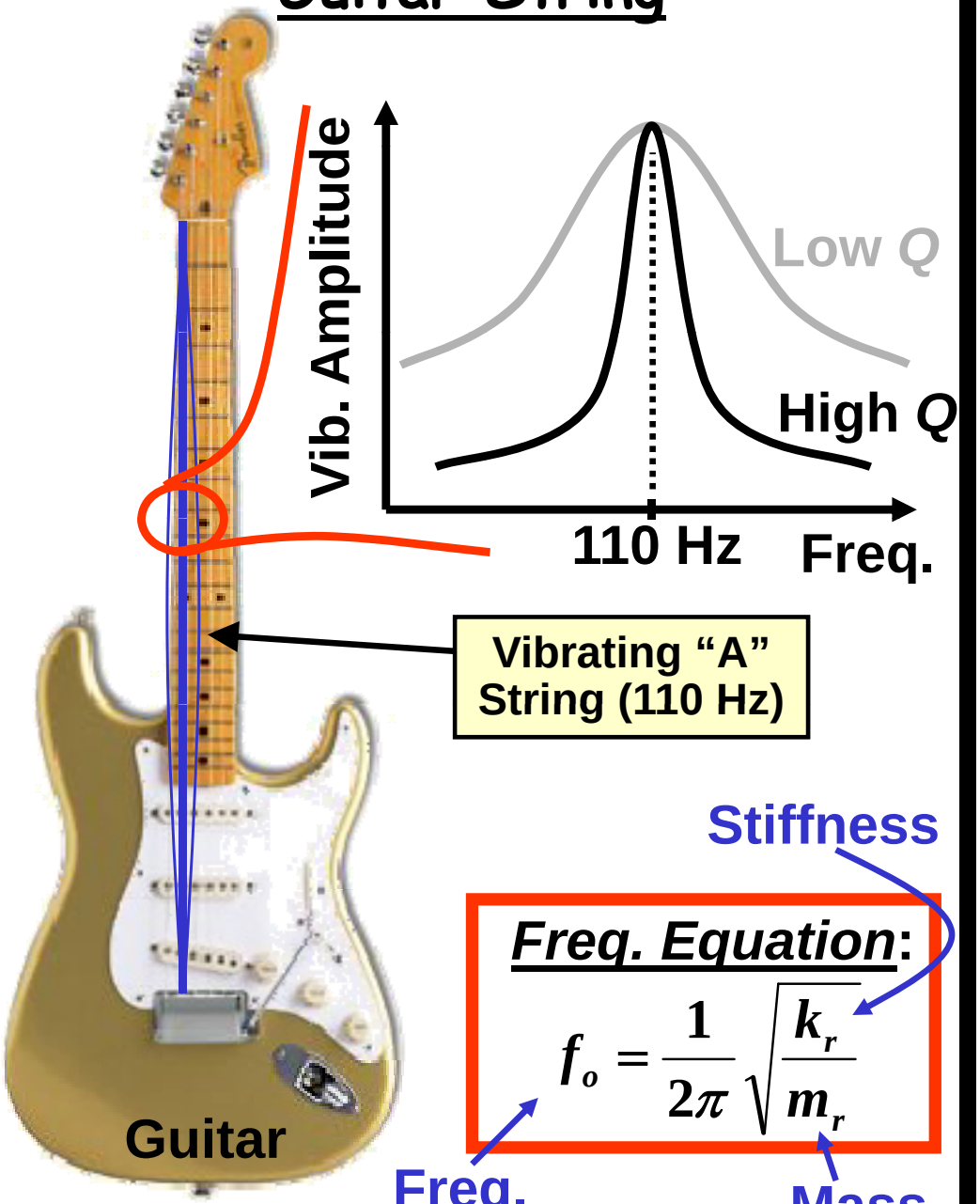
Guitar String



Freq. Equation:

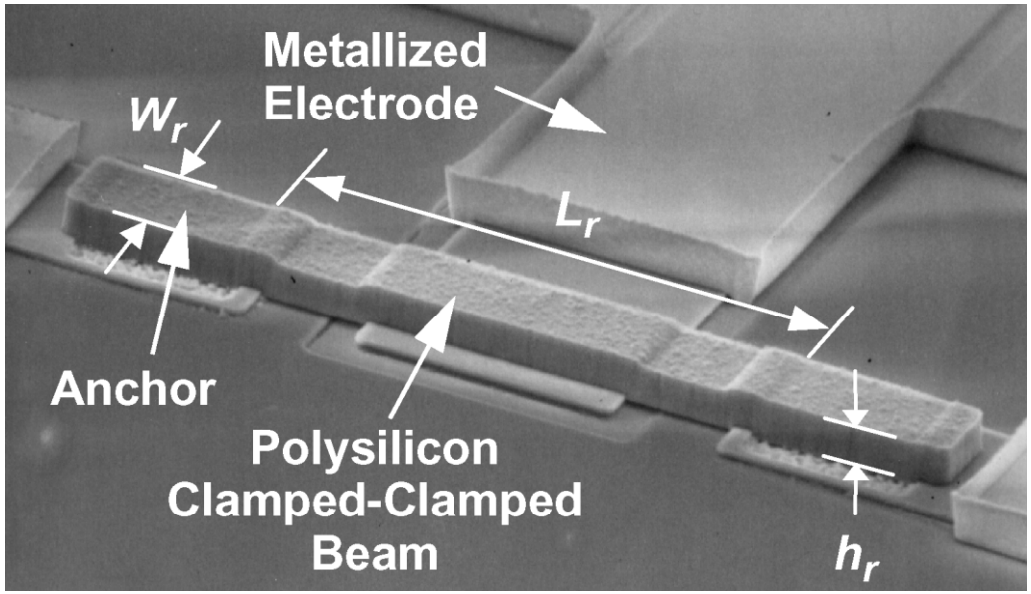
$$f_o = \frac{1}{2\pi}\sqrt{\frac{k_r}{m_r}}$$

Stiffness

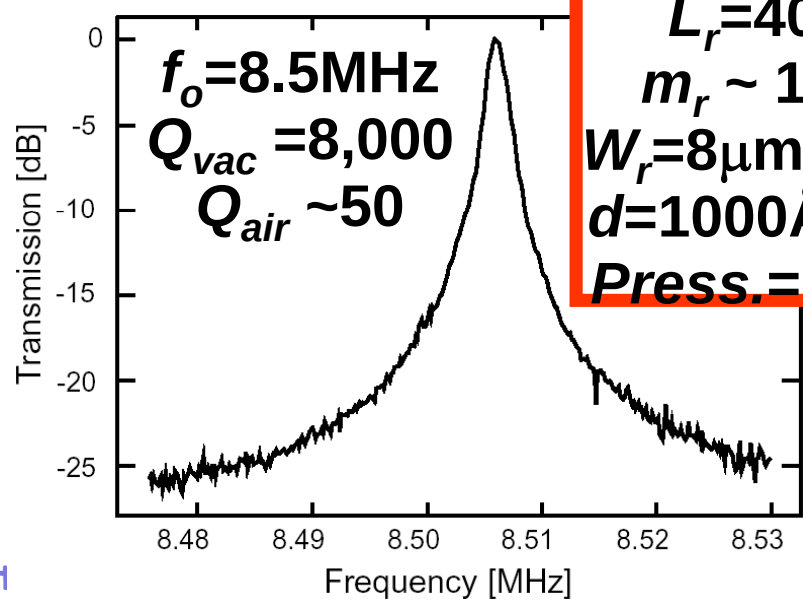
Freq.

Mass

μMechanical Resonator



[Bannon 1996]



***Performance*:**

L_r=40.8μm

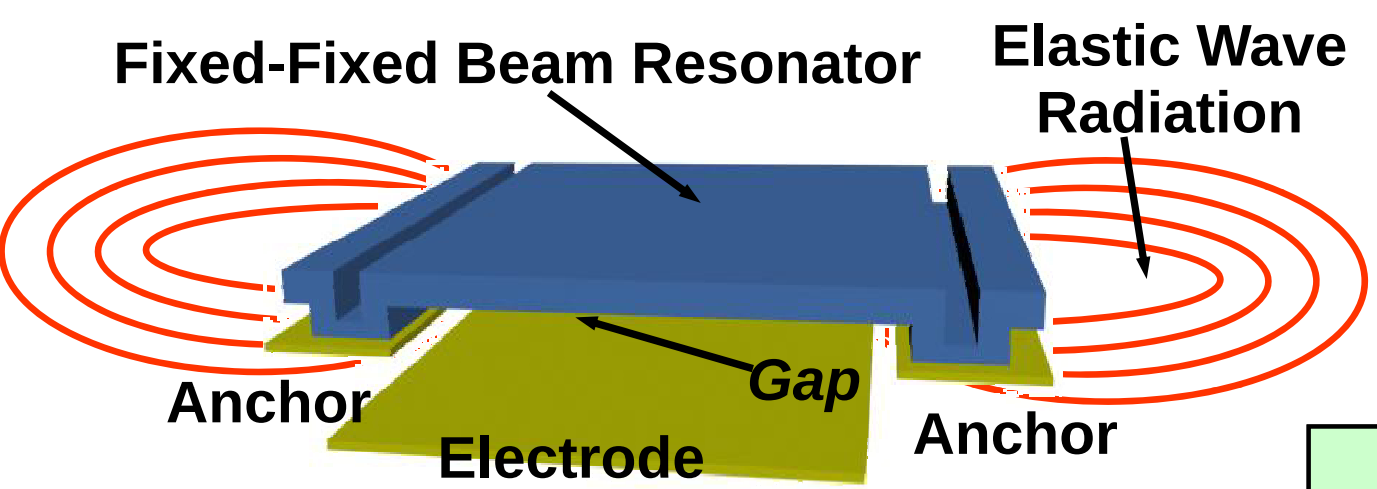
m_r ~ 10^{-13} kg

W_r=8μm, h_r=2μm

d=1000Å, V_P=5V

Press.=70mTorr

Anchor Losses

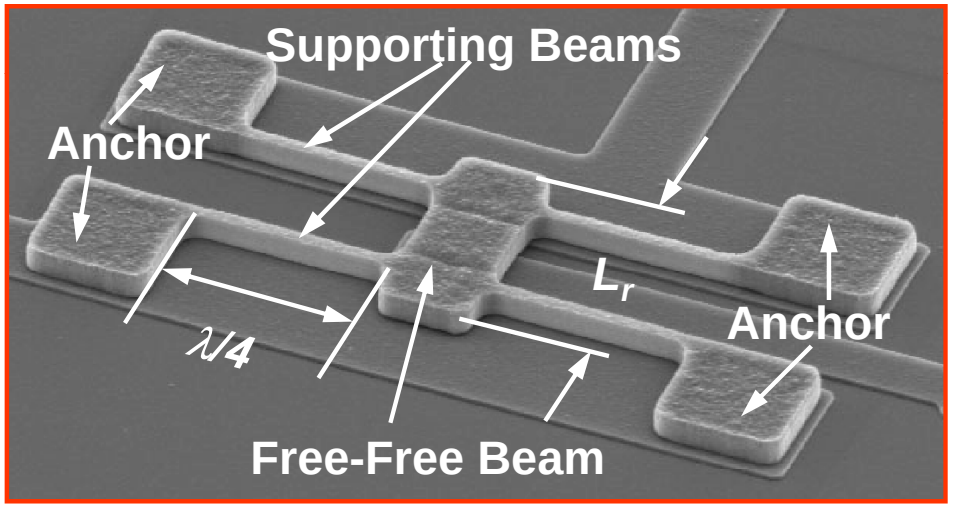


Q = 300 at 70MHz

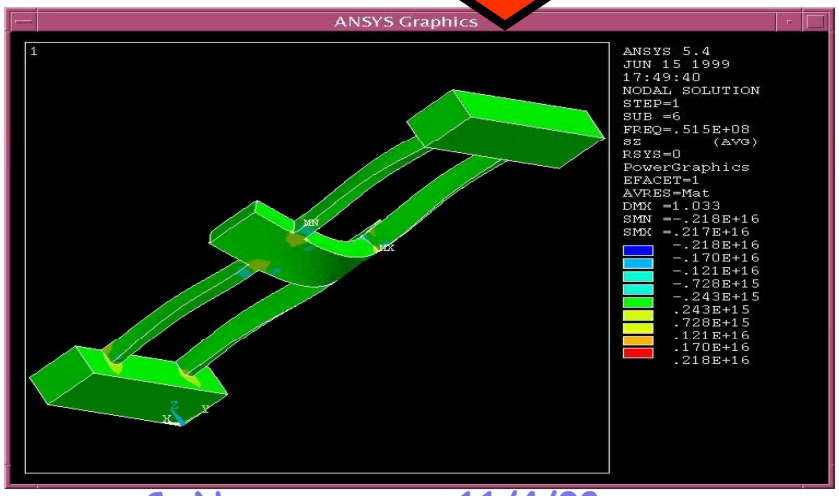
Problem: direct anchoring to the substrate ⇨ anchor radiation into the substrate ⇨ lower Q

Solution: support at motionless nodal points ⇨ isolate resonator from anchors ⇨ less energy loss ⇨ higher Q

Free-Free Beam Resonator

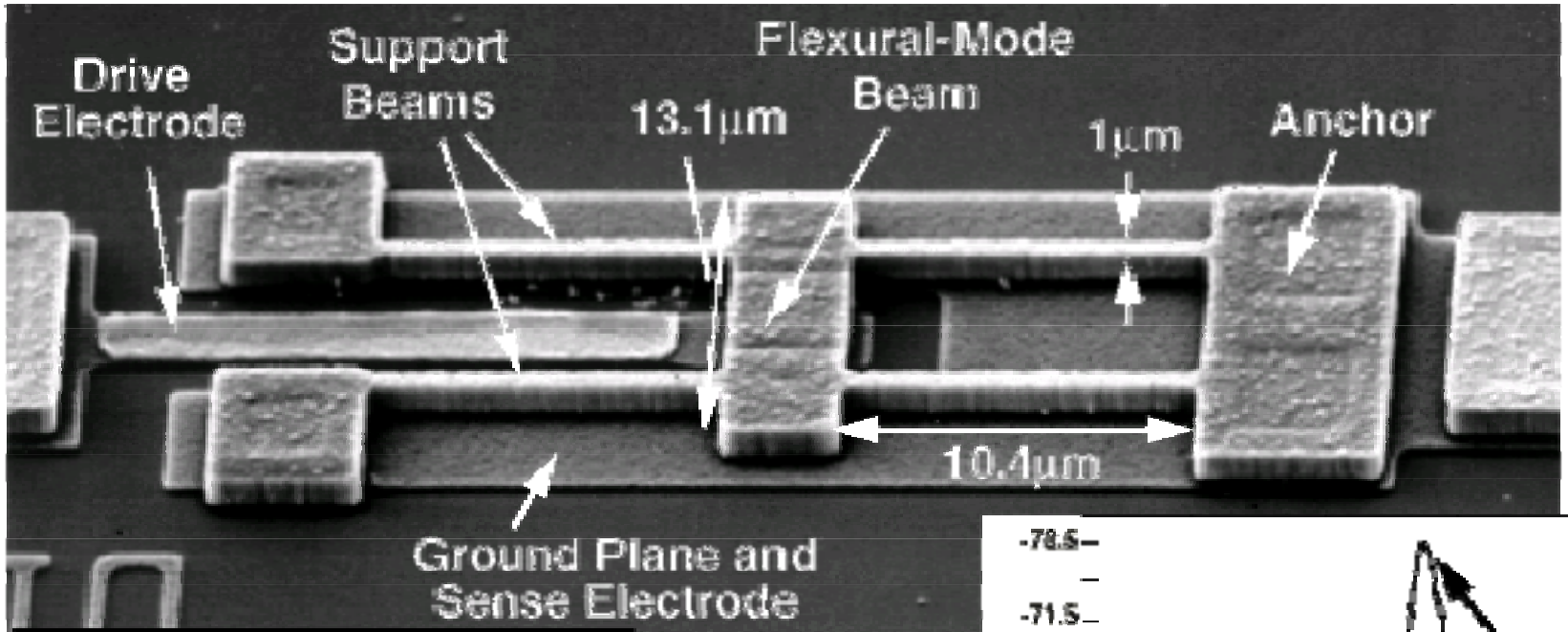


Q = 15,000 at 92MHz



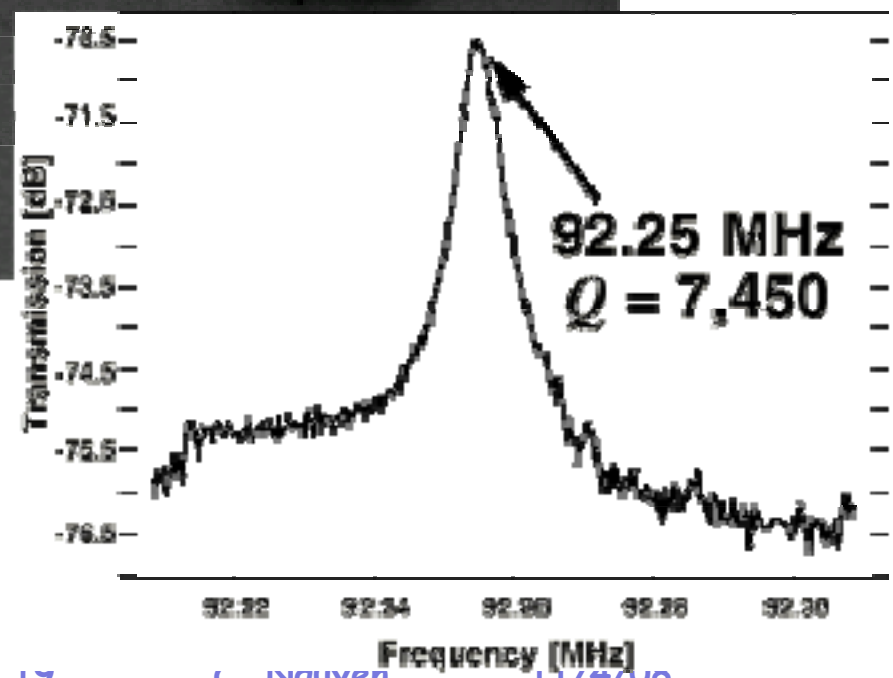
92 MHz Free-Free Beam μResonator

- Free-free beam μmechanical resonator with non-intrusive supports ⇨ reduce anchor dissipation ⇨ higher *Q*



Design/Performance:

L_r=13.1μm, W_r=6μm
h=2μm, d=1000Å
V_P=28-76V, W_e=2.8μm
f_o~92.25MHz
Q~7,450 @ 10mTorr



[Wang, Yu, Nguyen 1998]

Higher Order Modes for Higher Freq.

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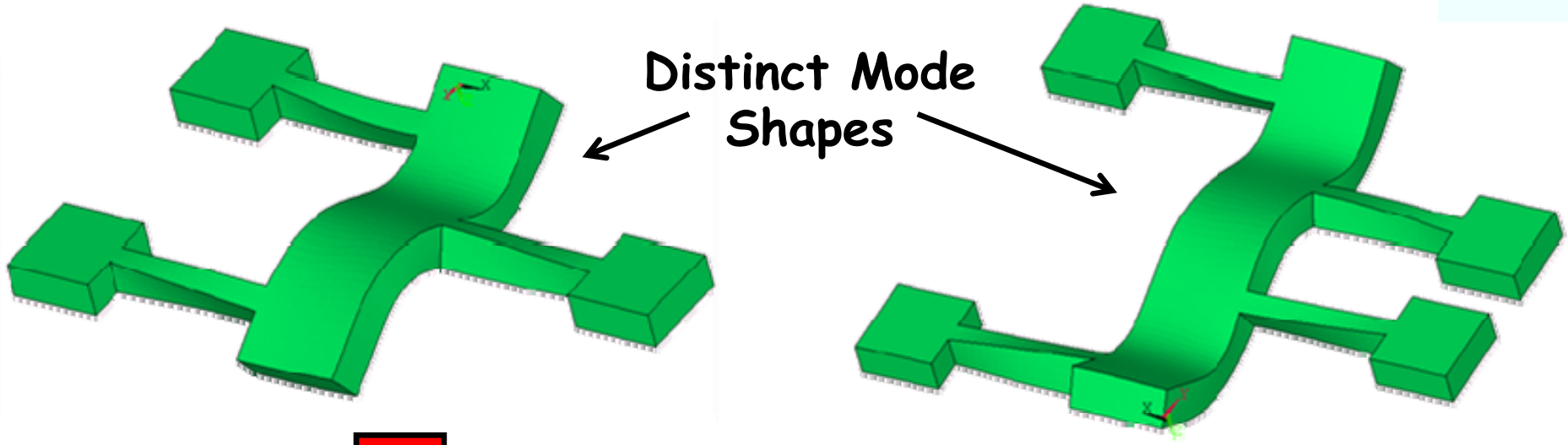
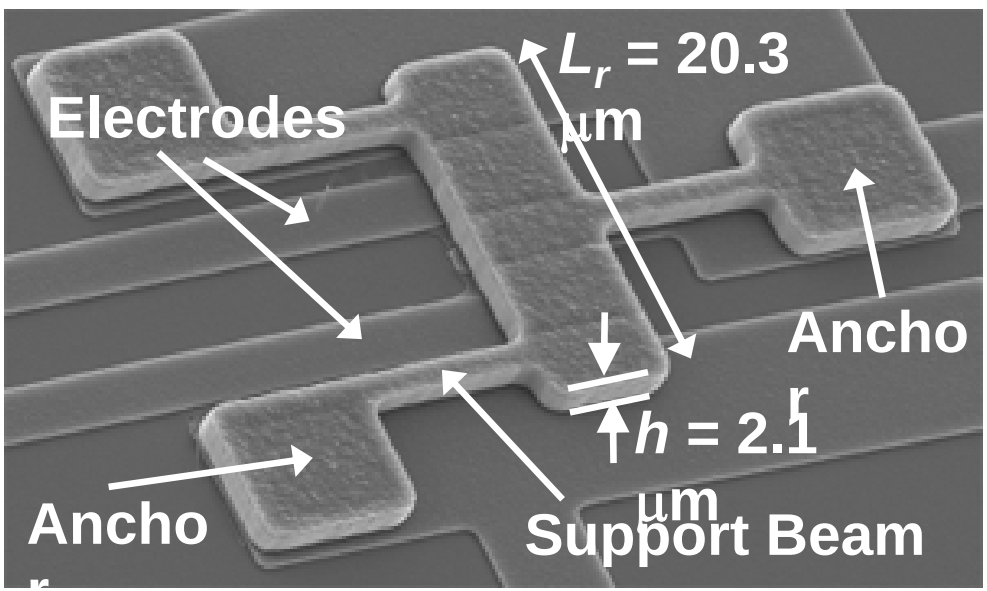
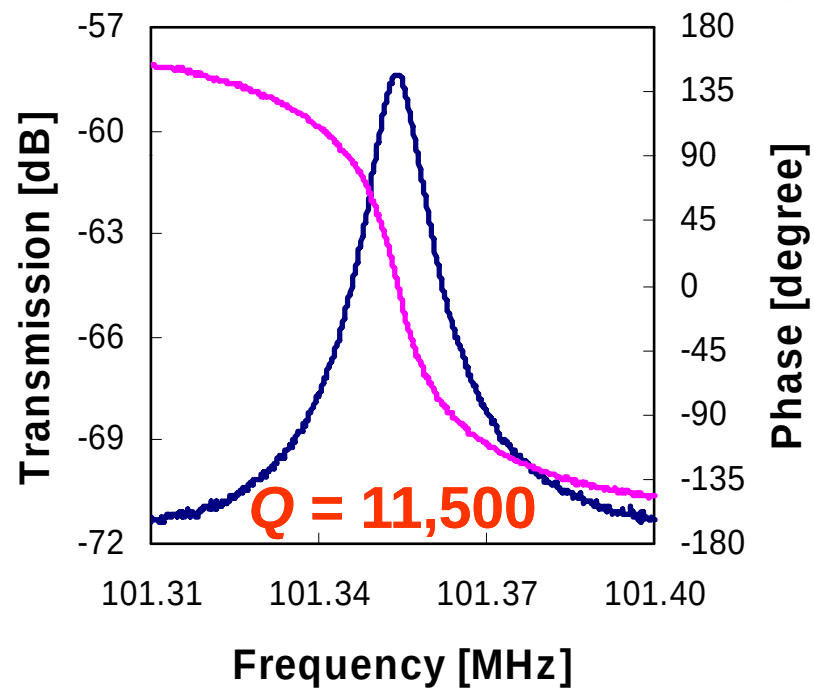
2nd Mode Free-Free Beam

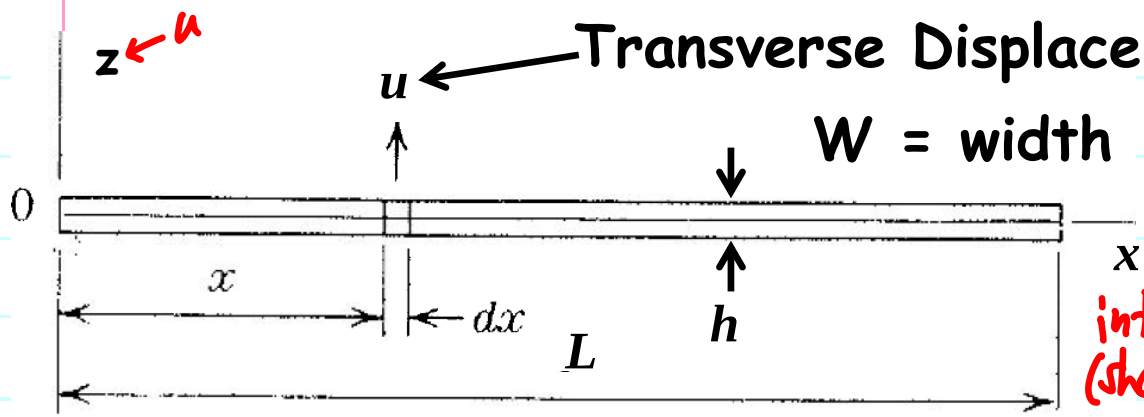
3rd Mode Free Free Beam





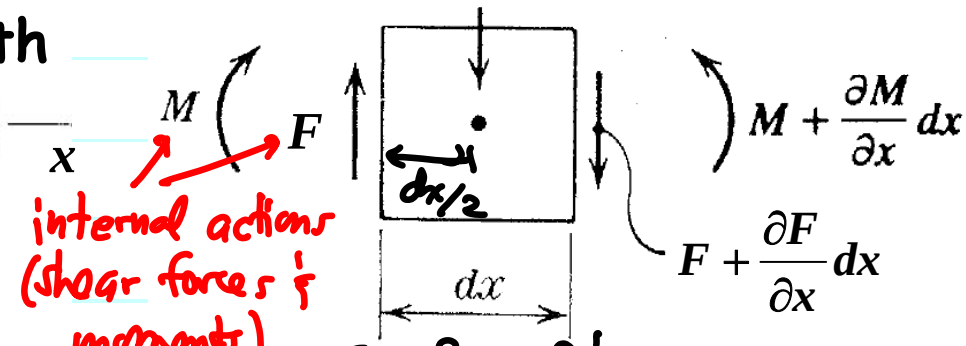


Flexural-Mode Beam Wave Equation



$$\rho A\, dx \frac{\partial^2 u}{\partial t^2} = ma$$

← inertial action



Free Body Diagram

- Derive the wave equation for transverse vibration:

Dynamic Equilibrium Condition for Forces in the z-direction:

$$F - \left(F + \frac{\partial F}{\partial x} dx\right) - \rho A dx \frac{\partial^2 u}{\partial t} = 0 \qquad (1)$$

$$\overbrace{\cancel{M} + F\frac{dx}{2} + F\frac{dx}{2} + \cancel{\frac{\partial F}{\partial x} dx \frac{dx}{2}} - \cancel{M} - \frac{\partial M}{\partial x} dx}^{0} \quad \text{neg.}$$

Do the same for moments:

$$-F dx + \frac{\partial M}{\partial x} dx \approx 0 \qquad (2)$$

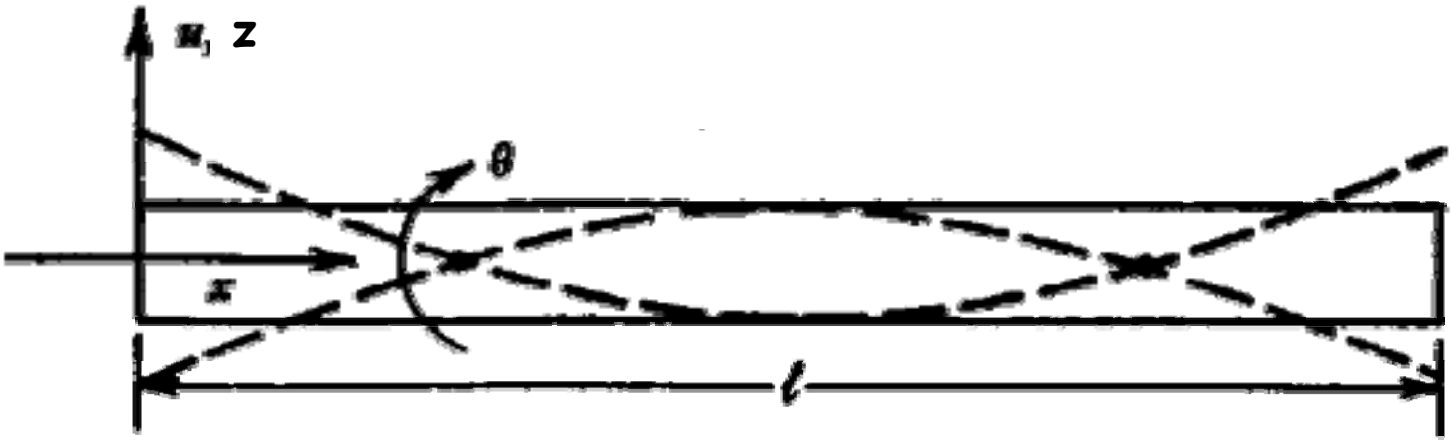
Combining (1) & (2):

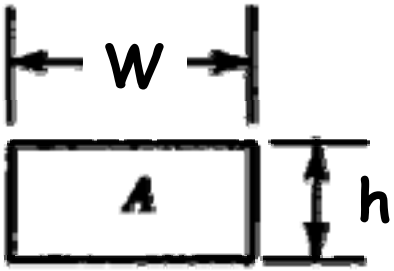
$$\frac{\partial^2 M}{\partial x^2} dx = -\rho A dx \frac{\partial^2 u}{\partial t^2} \rightarrow \frac{\partial^2}{\partial x^2}\left(-EI \frac{\partial^2 u}{\partial x^2}\right) = -\rho A \frac{\partial^2 u}{\partial t^2}$$

$$\left[\frac{\partial^2 u}{\partial x^2} = -\frac{M}{EI} \rightarrow M = -EI \frac{\partial^2 u}{\partial x^2}\right]$$

$$\boxed{\frac{\partial^2 u}{\partial t^2} = \left(\frac{EI}{\rho A}\right) \frac{\partial^4 u}{\partial x^2} \qquad I_y = \frac{Wh^3}{12}}$$

Example: Free-Free Beam





- Determine the resonance frequency of the beam
- Specify the lumped parameter mechanical equivalent circuit
- Transform to a lumped parameter electrical equivalent circuit
- Start with the flexural-mode beam equation:

$$\frac{\partial^2 u}{\partial t^2} = \left(\frac{EI}{\rho A}\right)\frac{\partial^4 u}{\partial x^4}$$

Free-Free Beam Frequency

- Substitute $u = u_1 e^{j\omega t}$ into the wave equation:

$$\frac{\partial^4 u}{\partial x^4} = \left(\omega^2 \frac{\rho A}{EI}\right) u \qquad (1)$$

- This is a 4th order differential equation with solution:

$$u(x) = \mathscr{A}\cosh kx + \mathscr{B}\sinh kx + \mathscr{C}\cos kx + \mathscr{D}\sin kx \qquad (2)$$

Gives the mode shape during resonance vibration.

- <u>Boundary Conditions</u>:

At $x = 0$	At $x = \ell$	
$\frac{\partial^2 u}{\partial x^2} = 0$	$\frac{\partial^2 u}{\partial x^2} = 0$	$M = 0$ (Bending moment)
$\frac{\partial^3 u}{\partial x^3} = 0$	$\frac{\partial^3 u}{\partial x^3} = 0$	$\frac{\partial M}{\partial x} = 0$ (Shearing force)

Free-Free Beam Frequency (cont)

- Applying B.C.'s, get A=C and B=D, and

$$\begin{bmatrix} (\cosh k\ell - \cos k\ell) & (\sinh k\ell - \sin k\ell) \\ (\sinh k\ell + \sin k\ell) & (\cosh k\ell - \cos k\ell) \end{bmatrix} \begin{bmatrix} \mathscr{A} \\ \mathscr{B} \end{bmatrix} = 0 \quad (3)$$

- Setting the determinant = 0 yields

$$\cos k\ell = \frac{1}{\cosh k\ell}$$

L ← length

- Which has roots at

$$k_1\ell = 4.730 \qquad k_2\ell = 7.853 \qquad k_3\ell = 10.996$$

These values of $k_n l$ correspond to the different modes of vibration!

- Substituting (2) into (1) finally yields:

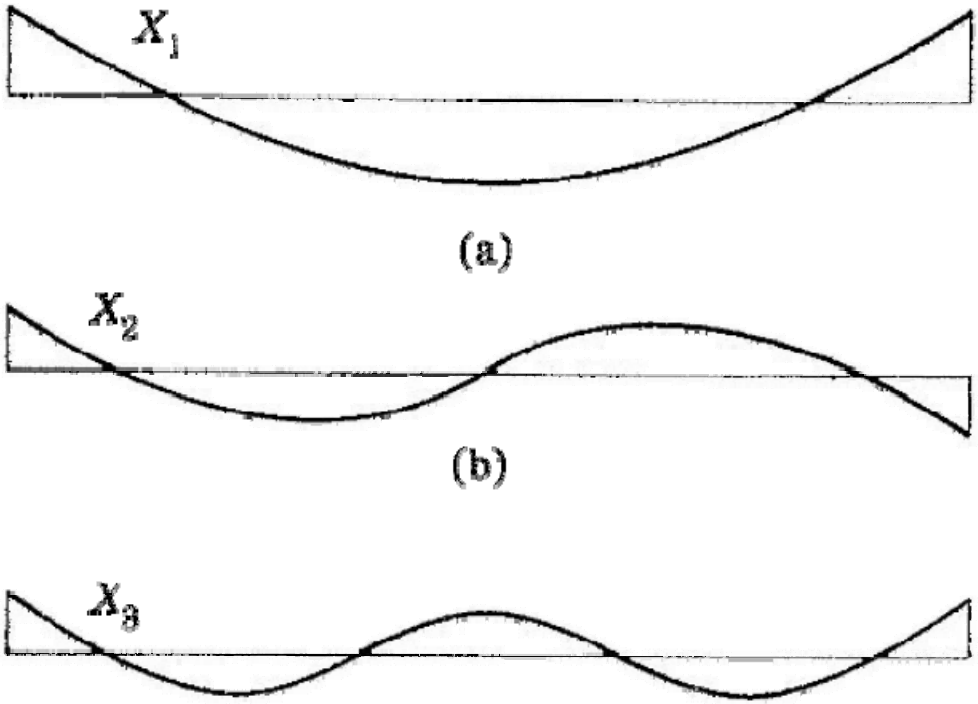
$$k^4 = \frac{\rho A}{EI}\omega^2 \longrightarrow \boxed{f_n = \frac{(k_n\ell)^2}{2\pi\ell^2}\sqrt{\frac{EI}{\rho A}}}$$

[Free-Free Beam Frequency Equation]

Higher Order Free-Free Beam Modes

Mode	n	Nodal Points	$k_n\ell$	f_n/f_1
Fundamental (f_1)	1	2	4.730	1.000
1st Harmonic	2	3	7.853	2.757
2nd Harmonic	3	4	10.996	5.404
3rd Harmonic	4	5	14.137	8.932
4th Harmonic	5	6	17.279	13.344

← More than 10x increase



Fundamental Mode (n=1)

1st Harmonic (n=2)

2nd Harmonic (n=3)

Mode Shape Expression

- The mode shape expression can be obtained by using the fact that A=C and B=D into (2), yielding

$$u_x = \mathscr{B}\left[\left(\frac{\mathscr{A}}{\mathscr{B}}\right)(\cosh kx + \cos kx) + (\sinh kx + \sin kx)\right]$$

- Get the amplitude ratio by expanding (3) [the matrix] and solving, which yields

$$\frac{\mathscr{A}}{\mathscr{B}} = \frac{\sin k\ell - \sinh k\ell}{\cosh k\ell - \cos k\ell}$$

- Then just substitute the roots for each mode to get the expression for mode shape

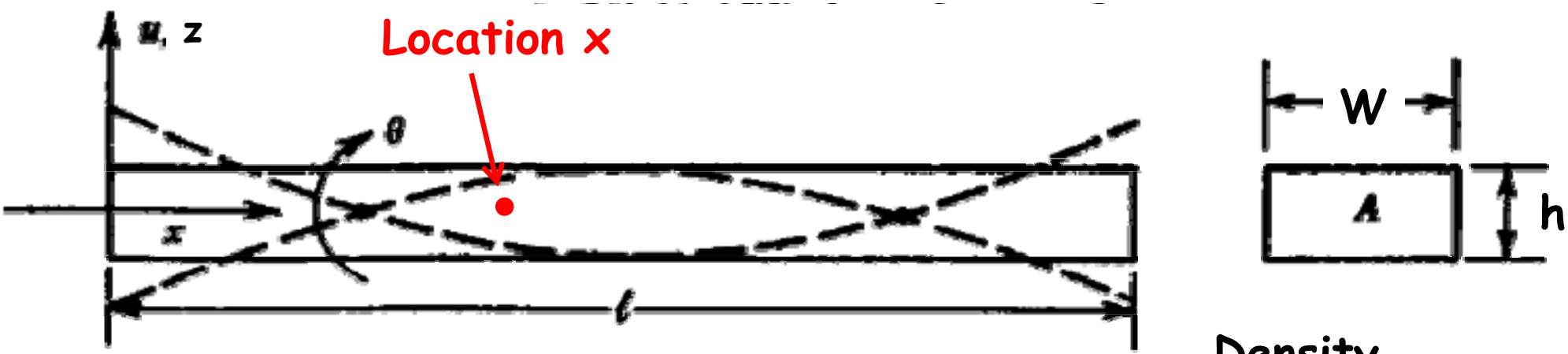
Fundamental Mode (n=1)

[Substitute $k_1\ell = 4.730$]

Lumped Parameter Mechanical Equivalent Circuit

Equivalent Dynamic Mass

- Once the mode shape is known, the lumped parameter equivalent circuit can then be specified
- Determine the equivalent mass at a specific location x using knowledge of kinetic energy and velocity



Density

Maximum Kinetic Energy

$$\text{Equivalent Mass} = M_{eq\,x} = \frac{K.E.}{\frac{1}{2}V_x^2} = \frac{\frac{1}{2}\rho A\int_0^{\ell} V^2(x)\,dx}{\frac{1}{2}V_x^2}$$

Maximum Velocity @ location x

Maximum Velocity Function