EE C247B – ME C218
Introduction to MEMS Design
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Lecture Module 9: Energy Methods

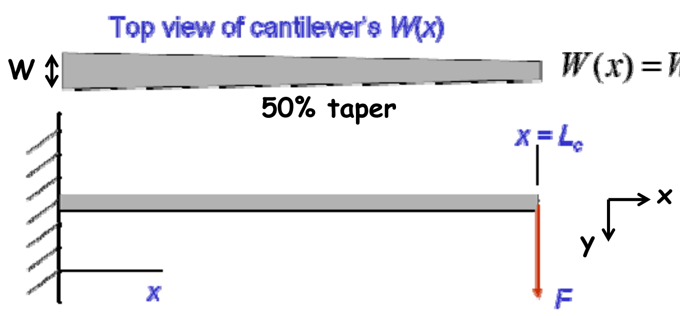
Lecture Outline

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

Energy Methods

More General Geometries

- Euler-Bernoulli beam theory works well for simple geometries
- But how can we handle more complicated ones?
- Example: tapered cantilever beam
- Objective: Find an expression for displacement as a function of location x under a point load F applied at the tip of the free end of a cantilever with tapered width W(x)



$$W(x) = W\left(1 - \frac{x}{2L_c}\right)$$

Solution: Use Principle of Virtual Work

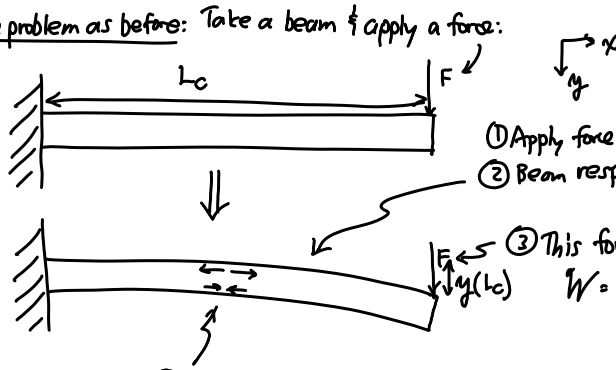
- In an energy-conserving system (i.e., elastic materials), the energy stored in a body due to the quasi-static (i.e., slow) action of surface and body forces is equal to the work done by these forces …
- Implication: if we can formulate **stored energy** as a function of the deformation of a mechanical object, then we can determine how an object responds to a force by determining the shape the object must take in order to **minimize** the **difference U** between the stored energy and the work done by the forces:

U = Stored Energy - Work Done

- Key idea: we don't have to reach U = 0 to produce a very useful, approximate *analytical* result for load-deflection

More Visual Description …

Same problem as before: Take a beam & apply a force:



① Apply force.
② Beam responds by bending.
③ This force has done work: $W = F \cdot y(L_c)$
④ Strain generated → This means the beam has received an influx of stored energy → magnitude determined by its deformed shape
⑤ Then:
U = Stored Energy − Work Done → 0
When we choose the right shape! (This is how we get the beam's response to F!)

Fundamentals: Energy Density

- Strain energy density: [J/m³] $W(Q) = \int_0^Q \frac{Q}{C} dQ$ → charging a capacitor from $0 \to Q$ takes this much work → stored energy on a capacitor
  - To find work done in straining material

This is a definition, so really can just say it's a definition.

$$w = \int_0^{\varepsilon_x} \sigma_x d\varepsilon_x \quad \text{x-axis normal stress term}$$

(ε_x ← value of strain @ position (x,y,z); $\sigma_x(\varepsilon_x)$ → relates stress to strain @ position (x,y,z))

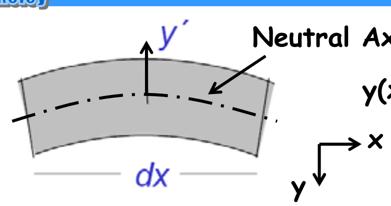
$$[\sigma_x = E\varepsilon_x] \Rightarrow w = \int_0^{\varepsilon_x} E\varepsilon_x d\varepsilon_x = \frac{1}{2}E\varepsilon_x^2$$

$W(q_1) = \int_0^{q_1} e(q) dq$, q = displacement, e = effect } Generic Definition of Work

- Total strain energy [J]:
  - Integrate over all strains (normal and shear)

$$W = \iiint \left(\frac{1}{2}E\left(\varepsilon_x^2 + \varepsilon_y^2 + \varepsilon_z^2\right) + \frac{1}{2}G\left(\gamma_{xy}^2 + \gamma_{xz}^2 + \gamma_{yz}^2\right) \right) dV$$

Bending Energy Density



y(x) = transverse displacement of neutral axis

- First, find the bending energy dW_{bend} in an infinitesimal length dx:

$$dW_{bend} = W dx \int_{-h/2}^{h/2} \frac{1}{2} E \epsilon_x^2(y') dy'$$

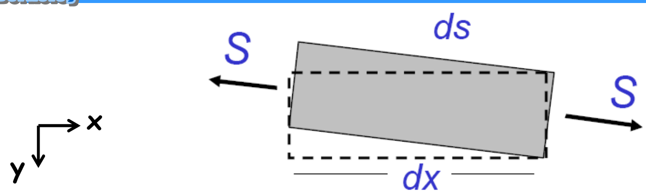
(W = width)

$$\left[\frac{1}{R} = \frac{d^2y}{dx^2},\ \epsilon_x = \frac{y'}{R}\right] \Rightarrow \epsilon_x(y') = y' \frac{d^2y}{dx^2}$$

$$dW_{bend} = W dx \int_{-h/2}^{h/2} \frac{1}{2} E \left[y' \frac{d^2y}{dx^2}\right]^2 dy' = \frac{1}{2} E \underbrace{\left(\frac{Wh^3}{12}\right)}_{I_z} \left(\frac{d^2y}{dx^2}\right)^2 dx$$

$$\therefore \boxed{W_{bend} = \frac{1}{2} E I_z \int_0^L \left(\frac{d^2y}{dx^2}\right)^2 dx}$$

Energy Due to Axial Load



- Strain due to axial load S contributes an energy $dW_{stretch}$ in length dx, since lengthening of the different element dx (to ds) results in a strain ε_x

Binomial Theorem

$$ds = \left[(dx)^2 + (dy)^2\right]^{1/2} = dx\left[1 + \left(\frac{dy}{dx}\right)^2\right]^{1/2} \approx dx\left[1 + \frac{1}{2}\left(\frac{dy}{dx}\right)^2\right]$$

$$\therefore \epsilon_x = \frac{ds - dx}{dx} = \frac{1}{2}\left(\frac{dy}{dx}\right)^2$$

Axial Strain Energy

$$\left[dW_{axial} = S\epsilon_x dx = \frac{1}{2}S\left(\frac{dy}{dx}\right)^2 dx\right] \Rightarrow \boxed{W_{axial} = \frac{1}{2}S\int_0^L \left(\frac{dy}{dx}\right)^2 dx}$$

Shear Strain Energy

$$\boxed{W_{shear} = \frac{3(EI_z)^2}{4GWh}\int_0^L \left(\frac{d^3y}{dx^3}\right)^2 dx}$$

Shear Modulus

- See W.C. Albert, "Vibrating Quartz Crystal Beam Accelerometer," Proc. ISA Int. Instrumentation Symp., May 1982, pp. 33-44

Applying the Principle of Virtual Work

- Basic Procedure:
  - Guess the form of the beam deflection under the applied loads
  - Vary the parameters in the beam deflection function in order to minimize:

Sum strain energies — Assumes point load

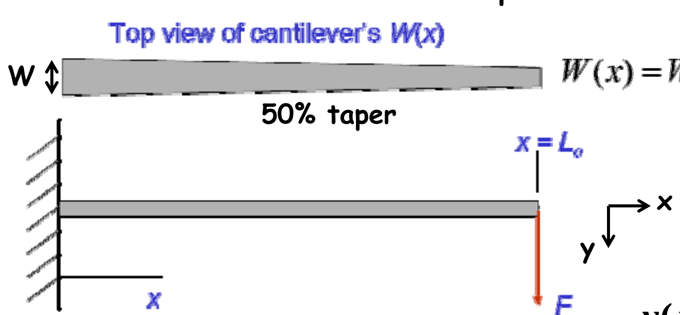
$$U = \sum_j W_j - \sum_i F_i u_i$$

Displacement at point load

  - Find minima by simply setting derivatives to zero
- See Senturia, pg. 244, for a general expression with distrubuted surface loads and body forces

Example: Tapered Cantilever Beam

- Objective: Find an expression for displacement as a function of location x under a point load F applied at the tip of the free end of a cantilever with tapered width W(x)



$$W(x) = W\left(1 - \frac{x}{2L_c}\right)$$

Adjustable parameters: minimize U

- Start by guessing the solution $\rightarrow$ $y(x) = c_2 x^2 + c_3 x^3$
  - It should satisfy the boundary conditions
  - The strain energy integrals shouldn't be too tedious
    - This might not matter much these days, though, since one could just use matlab or mathematica

Strain Energy And Work By F

$$U = \mathcal{W}_{bend} - F \cdot y(L_c)$$

$$\mathcal{W}_{bend} = \frac{1}{2} E \int_0^{L_c} I_z(x) \left(\frac{d^2 y}{dx^2} \right)^2 dx \quad \text{(Bending Energy)}$$

$$I_z(x) = \frac{W(x) h^3}{12}$$

$$\frac{d^2 y}{dx^2} = 2c_2 + 6c_3 x \quad \text{(Using our guess)}$$

$$W(x) = W(1 - \frac{x}{2L_c})$$

Tip Deflection

$$= \frac{1}{24} EWh^3 \int_0^{L_c} (1 - \frac{x}{2L_c})(2c_2 + 6c_3 x)^2 dx - F(c_2 L_c^{\ 2} + c_3 L_c^{\ 3})$$

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Find c_2 and c_3 That Minimize U

- Minimize U → basically, find the c_2 and c_3 that brings U closest to zero (which is what it would be if we had guessed correctly)
- The c_2 and c_3 that minimize U are the ones for which the partial derivatives of U with respective to them are zero:

$$\frac{\partial U}{\partial c_2} = 0 \qquad \frac{\partial U}{\partial c_3} = 0$$

- Proceed:
  - First, evaluate the integral to get an expression for U:

$$U = EWh^3 \left\{ \frac{5c_3^{\ 2}}{16} L_c^{\ 3} + \frac{c_2 c_3}{3} L_c^{\ 2} + \frac{c_2^{\ 2}}{8} L_c \right\} - F\left(c_2 L_c^{\ 2} + c_3 L_c^{\ 3}\right)$$

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Minimize U (cont)

- Evaluate the derivatives and set to zero:

$$\frac{\partial U}{\partial c_2} = 0 = \left(\frac{EWh^3}{3}c_3 - F\right)L_c^{\,2} + \left(\frac{EWh^3}{4}c_2\right)L_c$$

$$\frac{\partial U}{\partial c_3} = 0 = \left(\frac{5}{8}EWh^3c_3 - F\right)L_c^{\,3} + \left(\frac{EWh^3}{3}c_2\right)L_c^{\,2}$$

- Solve the simultaneous equations to get c_2 and c_3:

$$c_2 = \left(\frac{84}{13}\right)\frac{FL_c}{EWh^3} \qquad c_3 = -\left(\frac{24}{13}\right)\frac{F}{EWh^3}$$

The Virtual Work-Derived Solution

- And the solution:

$$y(x) = \left(\frac{24F}{13EWh^3}\right)\left(\left(\frac{7}{2}\right)L_c - x\right)x^2$$

- Solve for tip deflection and obtain the spring constant:

$$y(L_c) = \left(\frac{24F}{13EWh^3}\right)\left(\frac{5}{2}\right)L_c^{\,3} \qquad k_c = F/y(L_c) = \left(\frac{13EWh^3}{60L_c^{\,3}}\right)$$

- Compare with previous solution for constant-width cantilever beam (using Euler theory):

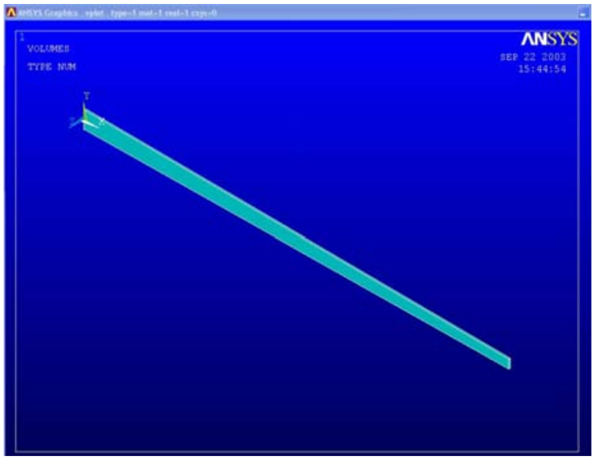
$$y(L_c) = \left(\frac{4F}{EWh^3}\right)L_c^{\,3}$$

⟶ 13% smaller than tapered-width case

Comparison With Finite Element Simulation

- Below: ANSYS finite element model with

  $L = 500\ \mu m$ $W_{base} = 20\ \mu m$ $E = 170\ GPa$
  $h = 2\ \mu m$ $W_{tip} = 10\ \mu m$



- Result: (from static analysis)
  - $k = 0.471\ \mu N/m$
- This matches the result from energy minimization to 3 significant figures

Need a Better Approximation?

- Add more terms to the polynomial
- Add other strain energy terms:
  - Shear: more significant as the beam gets shorter
  - Axial: more significant as deflections become larger
- Both of the above remedies make the math more complex, so encourage the use of math software, such as Mathematica, Matlab, or Maple
- Finite element analysis is really just energy minimization
- If this is the case, then why ever use energy minimization analytically (i.e., by hand)?
  - Analytical expressions, even approximate ones, give insight into parameter dependencies that FEA cannot
  - Can compare the importance of different terms
  - Should use in tandem with FEA for design

