2.2.17 Design of Machine Elements — Lecture 16, 17

Uncertainity

There are several uncertainities in machine design. The uncertainities arise from several factors including

— Variation in material composition and properties

— Variation in geometric features (e.g. diameter of a bar, thickness of a sheet, etc.) across a parts and across different parts

— Validity of mathematical models

— Intensity of stress concentrations

— Effect of corrosion, wear

Uncertainities need to be accommodated. The primary methods are deterministic and stochastic methods. The deterministic method establishes a design factor based on absolute uncertainities of a loss-of-function parameter and a maximum-allowable-parameter.

The parameter could be load, stress, deflection, etc.
Design factor n_d is defined as

$$n_d = \frac{\text{loss-of-function parameter}}{\text{maximum allowable parameter}}$$

e.g. $$n_d = \frac{\text{Yield strength}}{\text{maximum von-Mises stress}}$$

Problem 1

Consider that maximum load on a structure is known with an uncertainty of ± 20 percent and load causing failure is known within ± 15 percent. If load causing failure is 10 kN, determine the design factor and maximum load that can be applied.

Sol

Worst case

Design factor ≈ 1.41

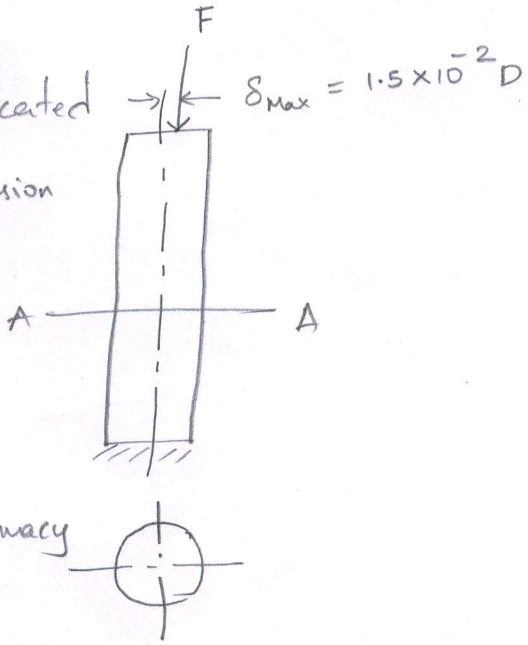
Min. load for failure is 8.5 kN.

Load can be determined to 20 % uncertainity
thus, we can apply up to $x \cdot 1.2 = 8.5 \Rightarrow x = 7.1$ kN

2.2.2017 Design of Machine Elements — Lecture 16

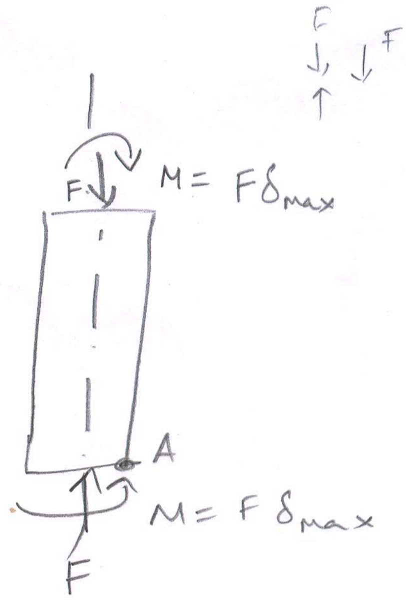
Problem 1

- Given that diameter can be fabricated to with ±1% of nominal dimension
- Support can be centered within ±1.5 %. of nominal diameter
- Weight is known to within 2%. accuracy
- Strength to ± 3.5 %.



Ans)

Peak stress would be at A



$$|\sigma_{peak}| = \frac{1.02\ F_{nominal}}{0.98\ A_{nominal}} + \frac{(1.02\ F_{nominal}) \times 0.015\ D_{nominal} \times 32}{\pi\ D^3_{nominal} \times 0.97}$$

$$\Rightarrow |\sigma_{peak}| = \frac{F}{A_{nominal}} \times \left[\frac{1.02}{0.98} + \frac{1.02 \times 0.015 \times 8}{0.97} \right]$$

$$= F/A_{nominal} \times 1.167$$

If a weight is dropped from height h maximum load F_{max} experienced would be

$$F_{max} = W\left(1 + \sqrt{1 + \frac{2hk}{W}}\right)$$

(k: rod spring constant; W: weight)

For h = 0, which is the case here

$$F_{max} = 2\, m_{nominal} \times g =$$

Thus, for max stress not to exceed strength we have

$$\frac{m_{nominal}\, g}{A_{nominal}} \times 2 \times 1.167 < 0.965\, (\sigma_s)_{nominal}$$

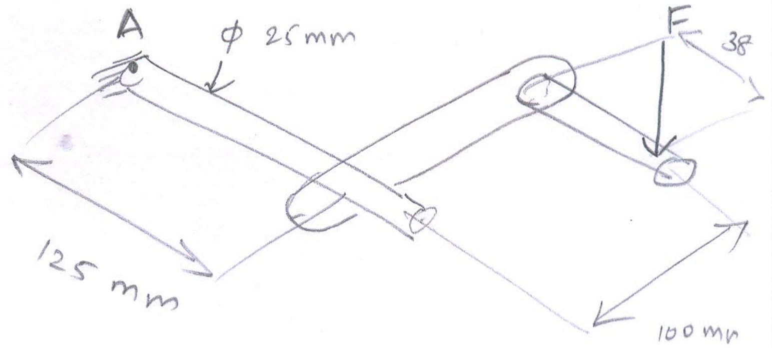
$$\frac{m_{nominal}\, g}{A_{nominal}} \equiv \sigma_{nominal}$$

$$\Rightarrow \text{Design factor } n_d \approx \frac{2 \times 1.167}{0.965}$$

$$\approx 2.42$$

Problem

Determine maximum normal and shear stress at A



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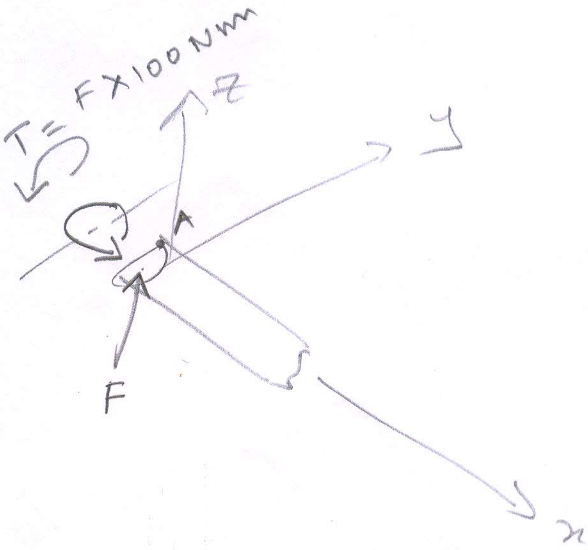


At A

$$\sigma_{xx} = \frac{My}{I} = \frac{F \times 163 \times 10 \times 64}{\pi \times 20^4} = 269.8 \frac{N}{mm^2}$$

$$\sigma_{xy} = \frac{Tr}{J} = \frac{F \times 100 \times 10 \times 32}{\pi \times 20^4} = 82.8 \frac{N}{mm^2}$$

$$\sigma_{xz} = \frac{F}{A} = \frac{F\,4}{\pi \times 20^2} = 4.14 \frac{N}{mm^2}$$

$$\Rightarrow [\sigma] \approx \begin{bmatrix} 270 & 82.8 & 0 \\ 82.8 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

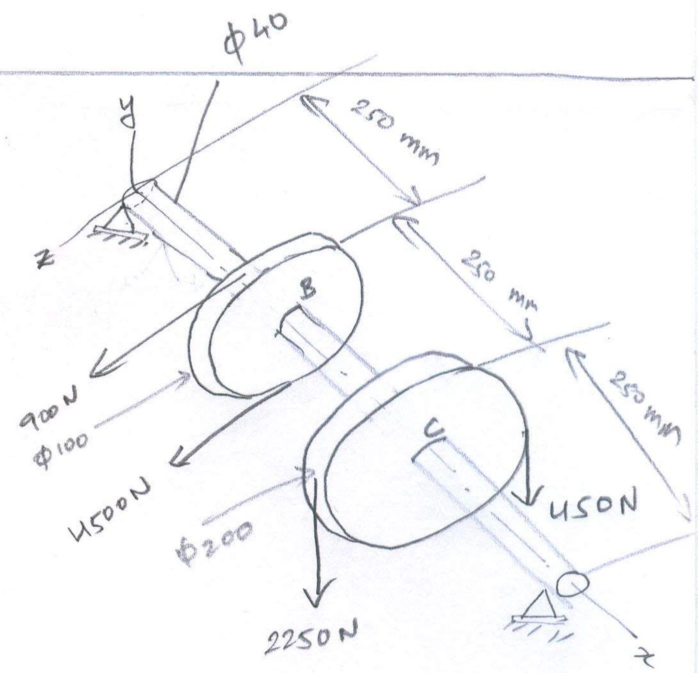
$$\Rightarrow \tau_{max} = \sqrt{\left(\frac{\sigma_{xx}}{2}\right)^2 + \sigma_{xy}^{\ 2}}$$

$$= \sqrt{\left(\frac{270}{2}\right)^2 + 82.8^2}$$

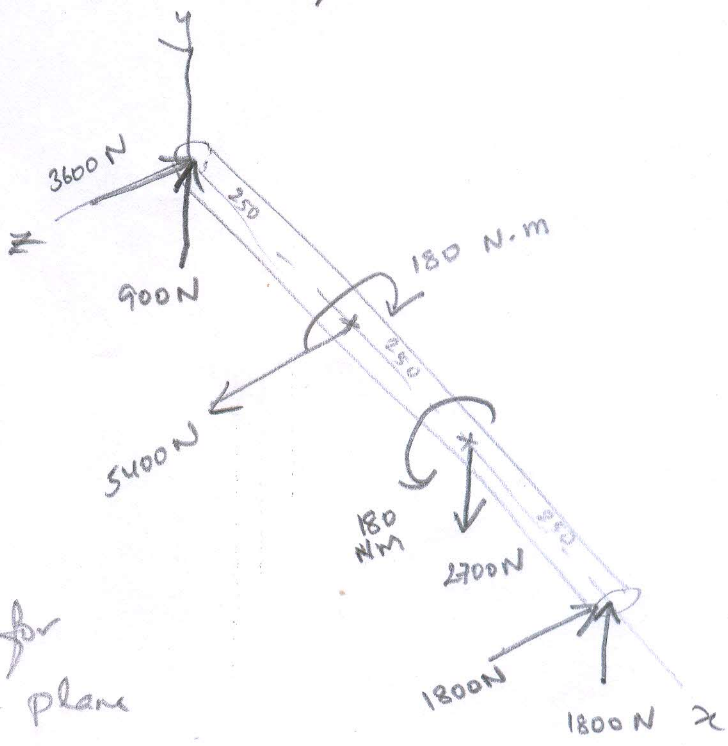
$$= 158.37 \text{ MPa}$$

Example 3-9

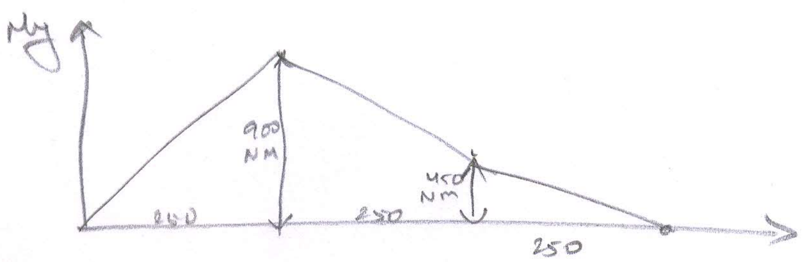
Considering bending and torsional stresses only determine greatest tensile, compressive, shear and von Mises stress in the shaft



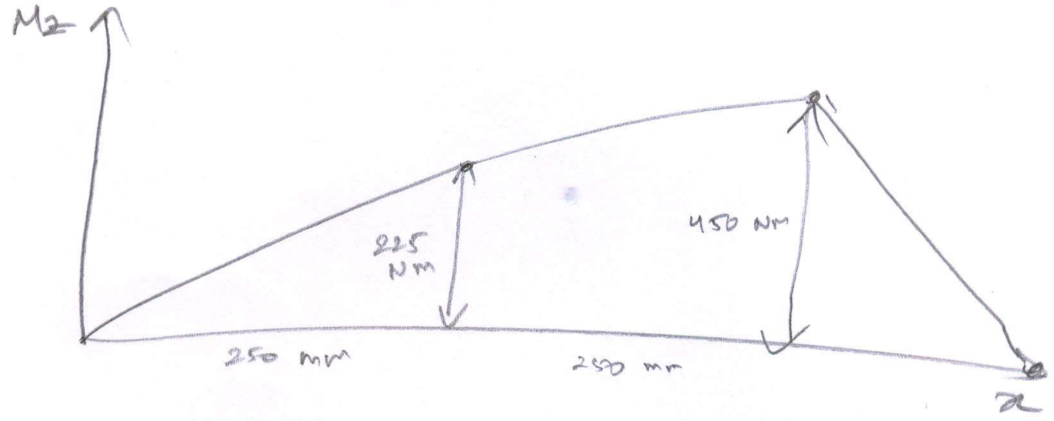
Free body diagram for the shaft



Bending moment diagram for My, i.e. bending in xz plane



Bending moment diagram for M_z, i.e. for bending in xy plane



Max tensile stress would occur at B

$$(\sigma_x)_{Max\ at\ B} = \frac{900 \times 0.02 \times 64}{\pi \times 0.04^4} = 143.24\ MPa$$

From torsion, max shear stress occurs ~~at~~ in between BC

$$\tau_{max} = \frac{Tr}{J} = \frac{180 \times 0.02 \times 32}{\pi \times 0.04^4} = 14.3\ MPa$$