Rules of Thumb for Shaft Design

- Keep shafts as short as possible
- Bearings should be close to applied loads
- Place any stress risers away from highly stressed regions of the shaft (local strengthening at risers may be helpful)
- Consider hollow shafts when weight is critical (weight to stiffness ratio should be low)
- For deflection concerns, choose a low carbon steel (why?)
- The first natural frequency of the shaft should be three times as high as the highest forcing frequency expected in service.

Design of Shafts based on loading modes

Fully reversed bending and constant Torsion

Shear stress due to torsion is steady—there will be a **mean component of shear**, **but not an alternating component of shear**

$$T_{\max} = T_{\min} = T$$

$$\tau_m = \frac{T_{\max} \, {}^{d}\!/_{2} (32) + T_{\min} \, {}^{d}\!/_{2} (32)}{\pi d^4}$$

$$= \frac{T(16) + T(16)}{\pi d^3} = \frac{32T}{\pi d^3}$$

$$\tau_a = \frac{T_{\max} \, {}^{d}\!/_{2} (32) - T_{\min} \, {}^{d}\!/_{2} (32)}{\pi d^4}$$

$$= 0$$

Stresses due to bending will yield an **alternating component**, **but not a mean moment**—see if you can convince yourself of this.

Designing Shafts for Fully Reversed Bending and Steady Torsion

The failure envelope for reversed bending and static torsion for test specimens is given by the following relationship:

$$\left(\frac{s_a}{S_e}\right)^2 + \left(\frac{t_m}{S_{ys}}\right)^2 = 1$$

Include a factor of safety, Nf

$$\left(N_f \frac{s_a}{S_e}\right)^2 + \left(N_f \frac{t_m}{S_{ys}}\right)^2 = 1$$

Sys = $\dfrac{S_y}{\sqrt{3}}$

$$\left(N_f \frac{s_a}{S_e}\right)^2 + \left(N_f \frac{\sqrt{3}t_m}{S_y}\right)^2 = 1$$

$$s = K_f \frac{32M}{pd^3}$$

$$t_m = K_{fs} \frac{16T_m}{pd^3}$$

$$\left(N_f \frac{K_f \dfrac{32M}{pd^3}}{S_e}\right)^2 + \left(N_f \frac{\sqrt{3}K_{fs} \dfrac{16T_m}{pd^3}}{S_y}\right)^2 = 1$$

Solve for d.