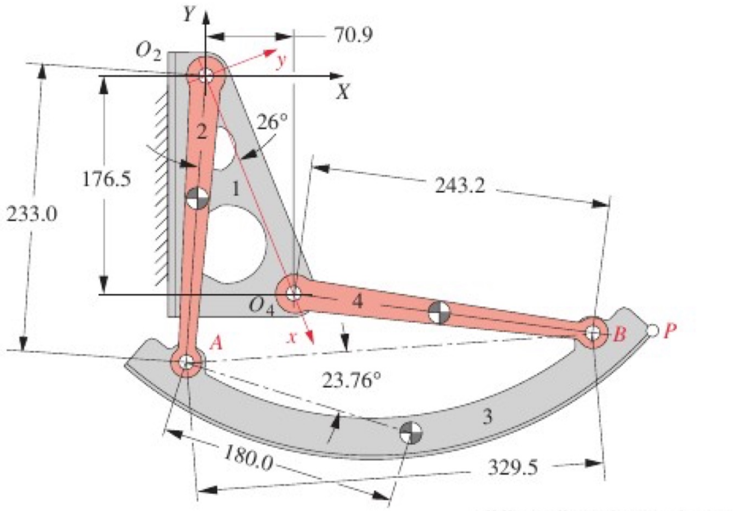
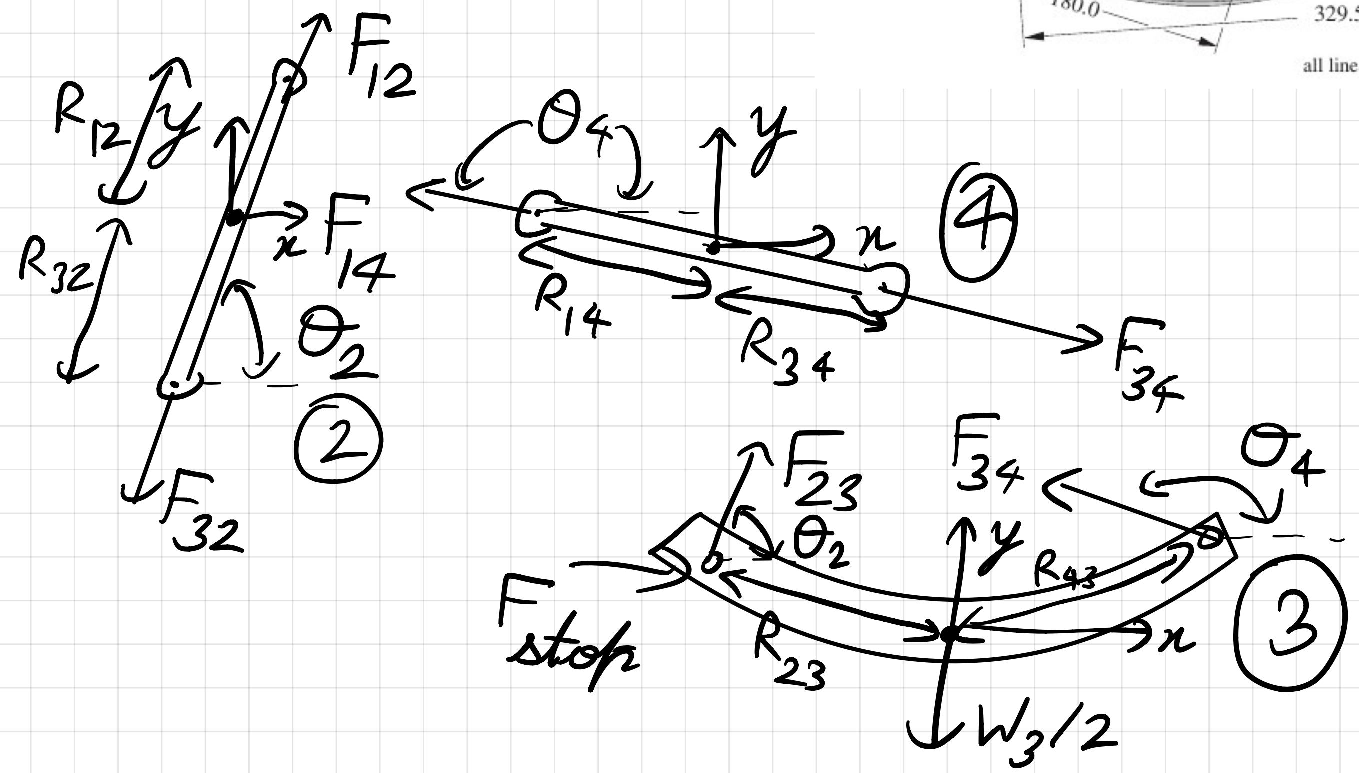
Ganesh Iyer ME-423: Tut-1
210100059

2. [Norton, Chapter 3] Figure shows an aircraft overhead bin mechanism in end view. For the position shown, draw free-body diagrams of links 2 and 4 and the door (3). There are stops that prevent further clockwise motion of link 2 (and the identical link behind it at the other end of the door) resulting in horizontal forces being applied to the door at points A. Assume that the mechanism is symmetrical so that each set of links 2 and 4 carry one half of the door weight. Ignore the weight of links 2 and 4 as they are negligible.
Also determine the pin forces on the door (3), and links 2 & 4 and the reaction force on each of the two stops. Available data:

R_{23}	180 mm @ 160.345°
R_{43}	180 mm @ 27.862°
W_3	45 N
θ_2	85.879°
θ_4	172.352°



all linear dimensions are in mm



Eqbm for 3 :

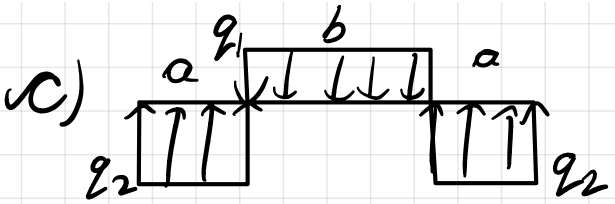
$$\Sigma F_x = 0 \Rightarrow F_{stop} + F_{23,x} + F_{43,x} = 0$$

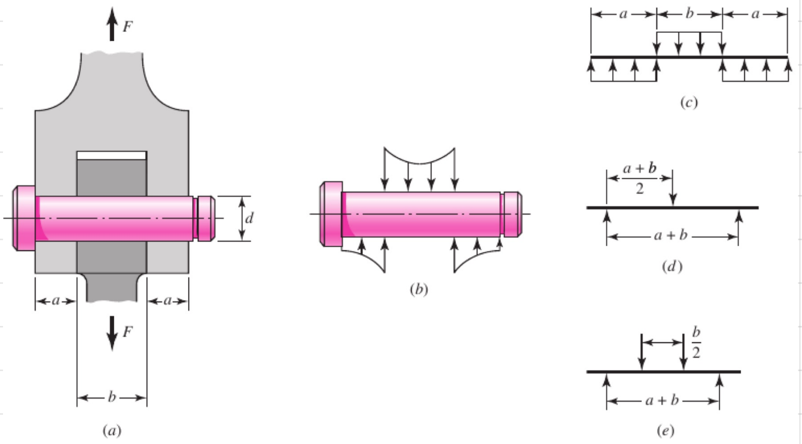
$$\Sigma F_y = 0 \Rightarrow F_{23,y} + F_{43,y} - W_3/2 = 0$$

$$\Sigma M = 0 \Rightarrow -F_{stop} \cdot R_{23,x} + F_{23,y} \cdot R_{23,x} - F_{23,x} \cdot R_{23,y} + F_{43y} \cdot R_{43x} - R_{43,y} \cdot F_{43,x} = 0$$

We can solve for the variables, and get pin force F_{23}, F_{43}, F_{stop}, and reaction on 2, 4
$-F_{23}$, $-F_{43}$.

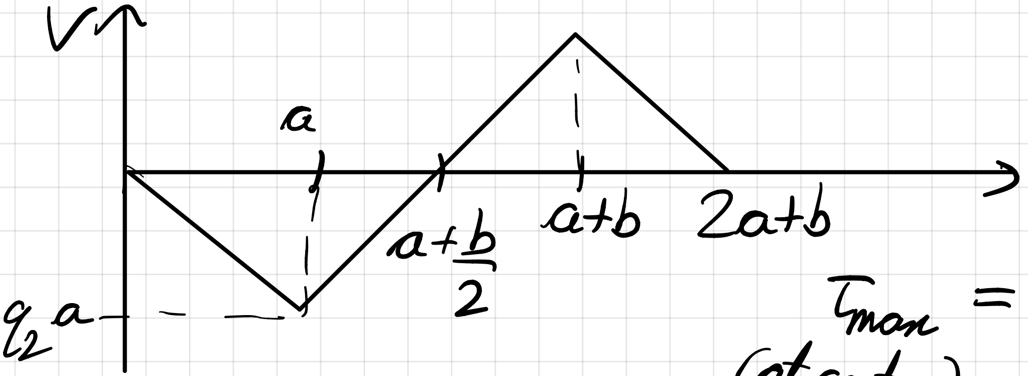
3. [Shigley, Chapter 3] A pin in a knuckle joint carrying a tensile load F deflects somewhat on account of this loading, making the distribution of reaction and load as shown in part (b) of the figure. A common simplification is to assume uniform load distributions, as shown in part (c). To further simplify, designers may consider replacing the distributed loads with point loads, such as in the two models shown in parts d and e. If a = 1.20 cm, b = 1.8 cm, d = 1.20 cm, and F = 4500 N, estimate the maximum bending stress and the maximum shear stress due to V for the three simplified models. Compare the three models from a designer's perspective in terms of accuracy, safety, and modeling time.





$$q_1 b = F \Rightarrow q_1 = \frac{\cancel{4500} \times 10^3}{\cancel{18}} = 250\ KN/m$$

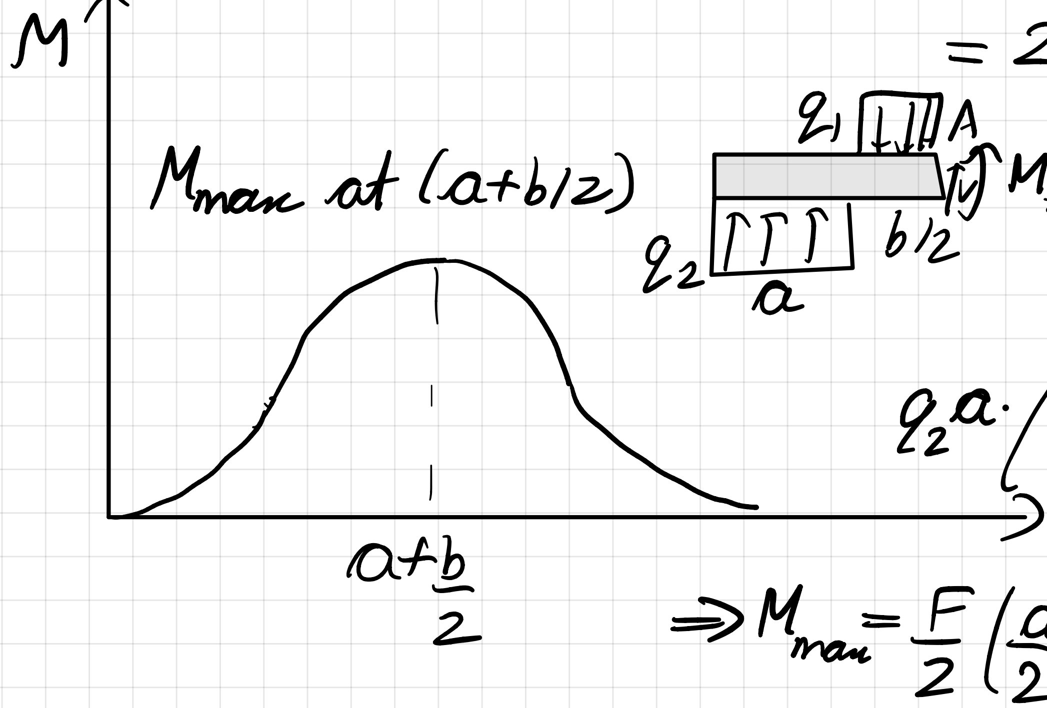
$$2 q_2 a = F \Rightarrow q_2 = \frac{\cancel{4500} \times 10^3}{2 \times \cancel{12}} = 187.5\ KN/m$$



$V_{max} = q_2 a = F/2$

For circular beam,

$$\underset{(\text{at centre})}{\tau_{max}} = \frac{4 V_{max}}{3A} = \frac{4 \times F/2}{3 \times \pi d^2/4} = 26.52\ MPa$$



Balance T about A,

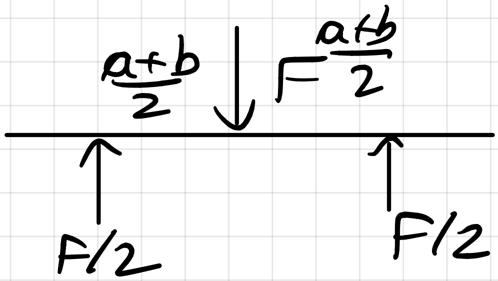
$$q_2 a \cdot \left(\frac{a+b}{2}\right) - q_1 \frac{b}{2} \cdot \frac{b}{4} = M_{max}$$

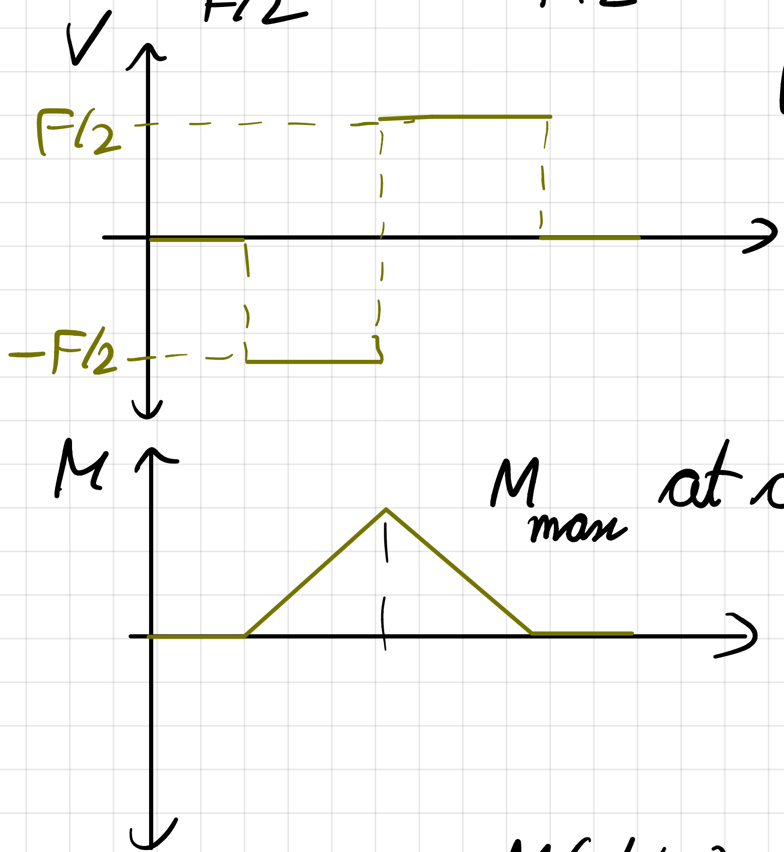
$$\Rightarrow M_{max} = \frac{F}{2}\left(\frac{a}{2} + \frac{b}{4}\right) = 23.625\ N\text{-}m$$

$$I = \frac{\pi d^4}{64} = 1.018 \times 10^{-9}$$

$$\sigma_{max} = \frac{M(d/2)}{I} = \frac{23.625 \times 0.6 \times 10^{-2}}{1.018 \times 10^{-9}} = 139.26 \text{ MPa}$$

d)

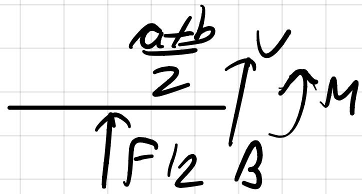




$V_{max} = F/2$

$$\tau_{max} = \frac{4 \times F/2}{3A} = 26.52 \text{ MPa}$$

M_{max} at centre

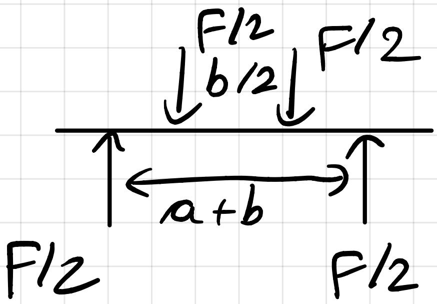


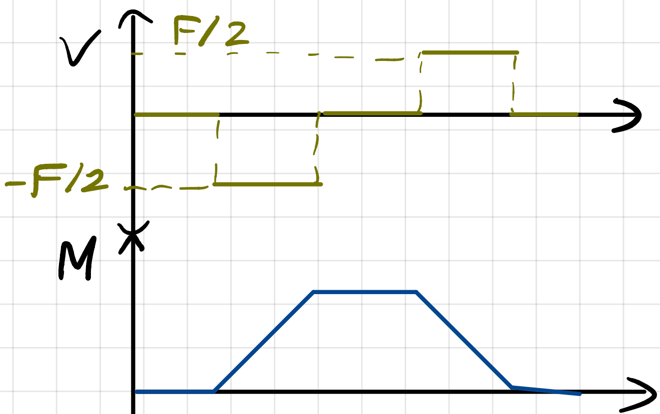
About B,

$$M = \frac{F}{2}\left(\frac{a+b}{2}\right) = 33.75 \text{ N-m}$$

$$\sigma_{max} = \frac{M(d/2)}{I} = 198.91 \text{ MPa}$$

e)





$V_{max} = F/2$

$$M_{max} \text{ (around centre)} = \frac{F}{2}\left(\frac{a+b}{2}\right) - \frac{F}{2}\left(\frac{b}{4}\right) = \frac{F}{4}\left(a + \frac{b}{2}\right) = 23.625 \text{ N-m}$$

$$\tau_{max} = \frac{4 \times F/2}{3A} = 26.52\ MPa$$

$$\sigma_{max} = \frac{M(d/2)}{I} = 139.26\ MPa$$

Distributed Load most accurate, since actual loads are not point loads.

τ_{max} same in all, but σ_{max} highest in 3-point loading, most conservative, good from safety point of view.

From modelling perspective, 3-point loading has min no. of loads so easiest to analyze.

5. [Shigley, Chapter 3]. A gear reduction unit uses the countershaft shown in the figure. Gear A receives power from another gear with the transmitted force F_A applied at the 20° pressure angle as shown. The power is transmitted through the shaft and delivered through gear B through a transmitted force F_B at the pressure angle shown.

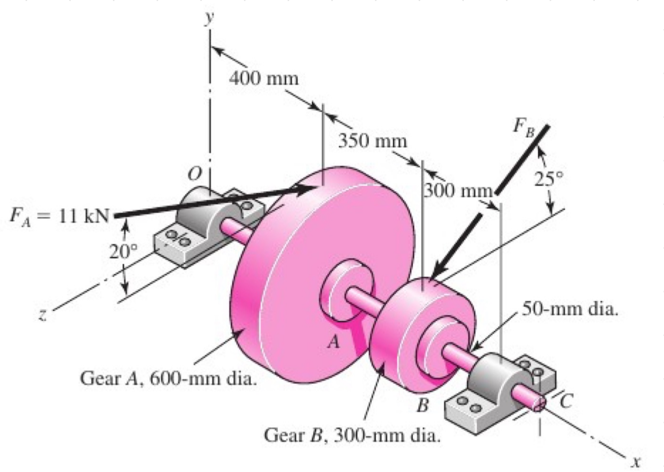
(a) Determine the force F_B, assuming the shaft is running at a constant speed.

(b) Find the magnitudes of the bearing reaction forces, assuming the bearings act as simple supports.

(c) Draw shear-force and bending-moment diagrams for the shaft. If needed, make one set for the horizontal plane and another set for the vertical plane.

(d) At the point of maximum bending moment, determine the bending stress and the torsional shear stress.

(e) At the point of maximum bending moment, determine the principal stresses and the maximum shear stress.

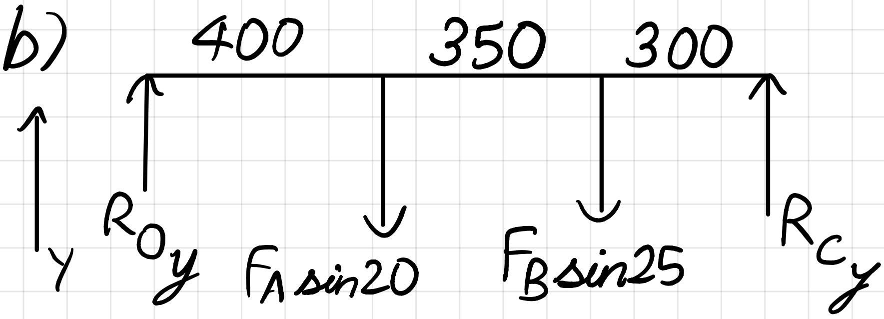


a) Torque Balance about axis,

$$F_A \cos 20 \times r_A = F_B \cos 25\, r_B$$

$$\Rightarrow F_B = 22.81\ KN$$

b)

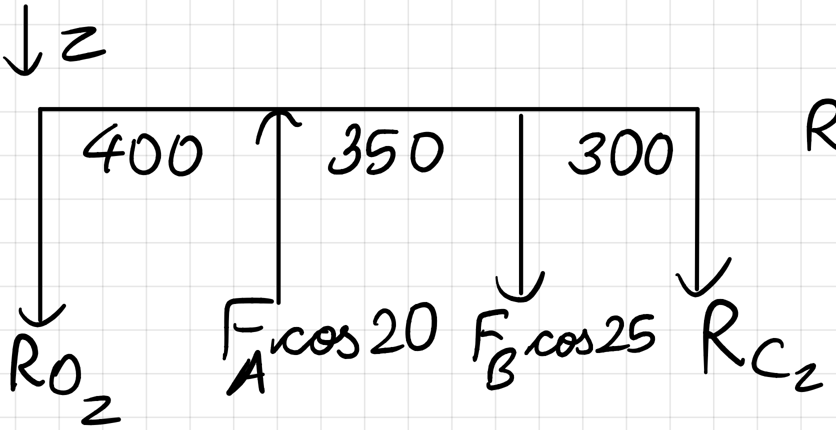


$R_{O_y} + R_{C_y} = 13.4\ KN$

$F_A \sin 20 \times 400 + F_B \sin 25 \times 750 = R_{C_y} \times 1050$

$R_{C_y} = 8.32\ KN$

$R_{O_y} = 5.08\ KN$



$R_{O_z} + R_{C_z} = -10.33$

$F_A \cos 20 \times 400 = F_B \cos 25 \times 750 + R_{C_z} \times 1050$

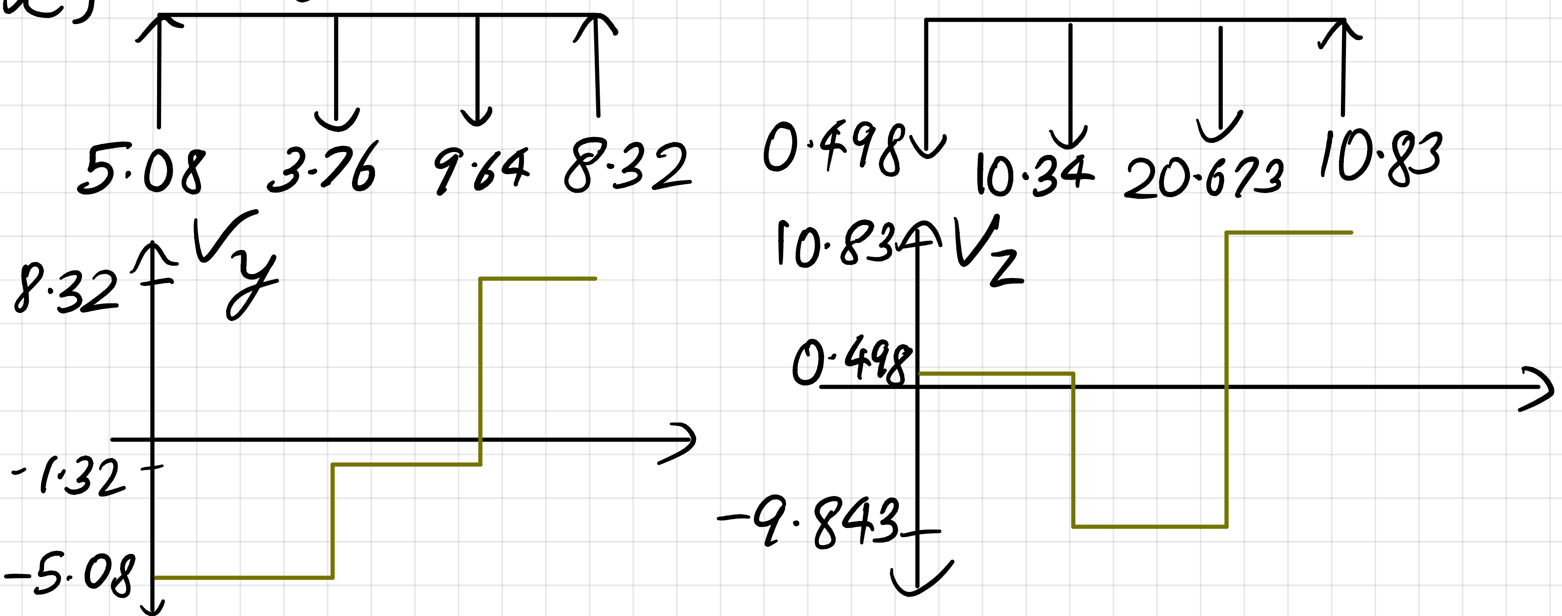
$R_{C_z} = -10.83\ KN$

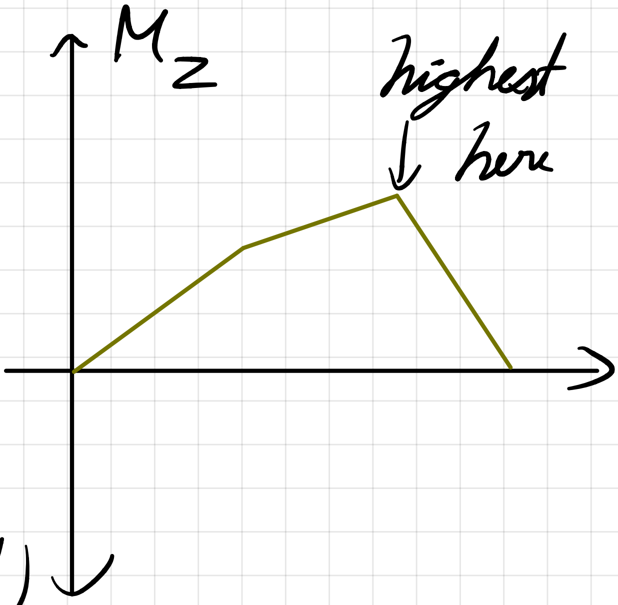
$R_{O_z} = 0.498\ KN$

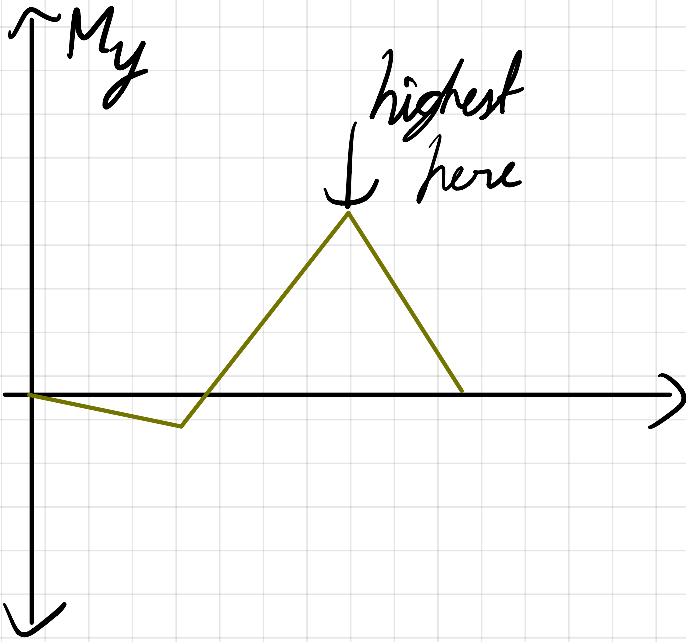
$R_C = \sqrt{R_{C_y}^2 + R_{C_z}^2} = 13.657\ KN$

$R_O = \sqrt{R_{O_y}^2 + R_{O_z}^2} = 5.1\ KN$

c)







d)

Critical location at B, max bending moment.

$$M_z = 300 \times R_{Cy} = 2.5 \text{ KN-m}$$

$$M_y = 300 \times R_{Cz} = 3.25 \text{ KN-m}$$

$$M = \sqrt{M_y^2 + M_z^2} = 4.1 \text{ KN-m}$$

$$T = 11 \cos 20 \times r_A = 3.1 \text{ KN-m}$$

$$\sigma = \frac{Mr}{I} = \frac{Md}{2 \times \frac{\pi d^{\cancel{4}3}}{\cancel{64}32}} = \frac{32M}{\pi d^3} = \frac{32 \times 4.1}{\pi (0.05)^3}$$

$$= 334 \text{ MPa}$$

$$\tau = \frac{Tr}{J} = \frac{16T}{\pi d^3} = 126.3 \text{ MPa}$$

e) $$\sigma_{1,2} = \frac{\sigma}{2} \pm \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= 376, -42.4 \text{ MPa}$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = 209 \text{ MPa}$$