Esercitazione – Threaded Connection

For loose pg.16 + pg.17

Exercise 1 → Loose Screw
↳ Opposite of Tightened.

Eye bolt → loose
↳ With static case

1.5 kN motor

Screw = 6,8

Coarse Pitch

40mm long

Define d_3 → according to the table of metric threads

$$\sigma_{MAX} = \frac{F_{MAX}}{A_3} = \frac{60000}{\pi d_3^2}$$

↳ or A_2 ↳ up to us

$$\sigma_{AMM} = \frac{R_{su}}{\eta} = \frac{480}{1,5}$$

$\eta \to 1,5$

↳ since 6,8 is ductile.

Screw name to be found:

UNI 5737 $M(D_{nom}) \times p \times l - s$

pitch

nominal thread diameter.

length

strength class

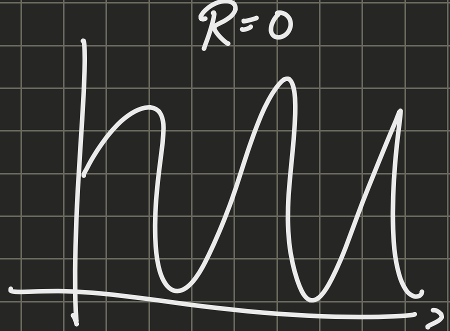
In the table (pg.5) stress area is A_2, already calculate and A_3 is the minor area, so we could directly find the area.

We find $d_3 \geq 7,73$ → looking at the minor diameter column, we see that $D = 10mm$, has $d_3 = 8,160$ which is the first the satisfies our condition on d_3.

Exercise → loose screw with fatigue

↳ Assume that it's loaded more times than before so now we have to consider fatigue.

Does the same screw from before work?



$$\sigma_a = \frac{F_A}{2A_3} = \frac{F_{A,max}}{\frac{\pi}{2} d_3^2} = \frac{30000}{\pi \cdot 8{,}16^2} = 143{,}5$$

$\sigma_m = 143{,}4\,MPa$

$$\frac{F_{A,max} - \cancel{F_{A,min}}^{\,0}}{2A_3}$$

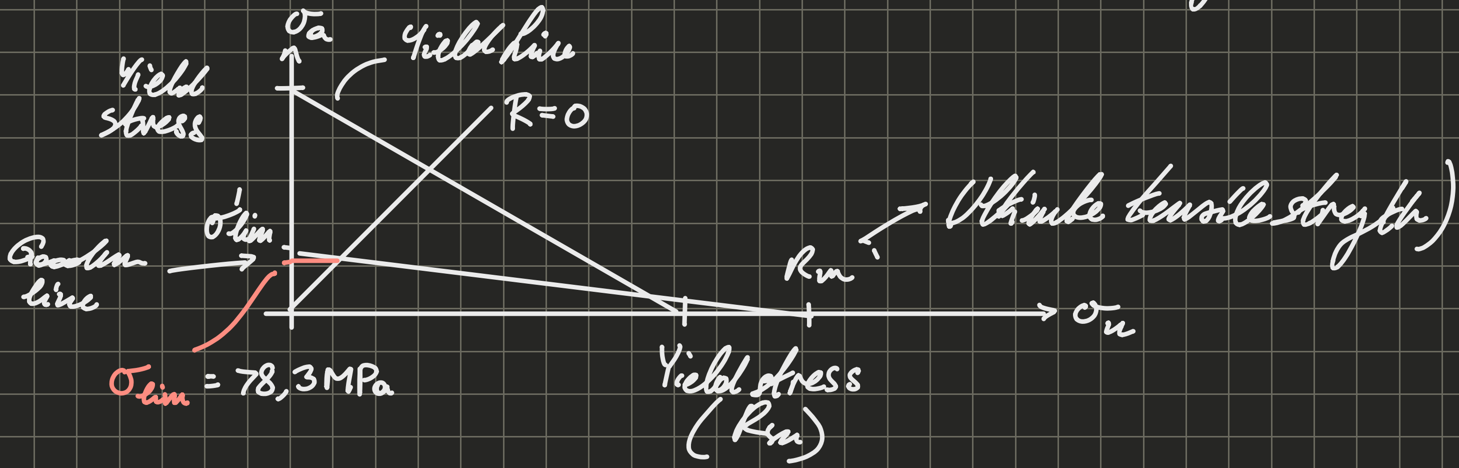
$$\frac{F_{A,max} + \cancel{F_{A,MIN}}^{\,0}}{2A_3}$$

$$R=0 \Rightarrow \sigma_a \le \frac{\sigma_{lim}}{\eta} \rightarrow \text{unknown}$$

$+\,\tau = 0$

Table gives σ_{lim} for fatigue, then we have to apply the σ_m through Haigh diagram

$\sigma'_{lim} = 90\,MPa$ → from table, considers already b_2, b_3 and k_{ff}.



$\sigma_{lim} = 78{,}3\,MPa$

$\eta = 0{,}54$ → Problem

↳ as we have said, we should always try to avoid fatigue loading.

We can find a new screw, by again keeping d_3 as the unknown, and imposing $\eta = 1{,}5 \rightarrow$ since once again ductile.

Exercise $\rightarrow$ Non-symmetric screw load.

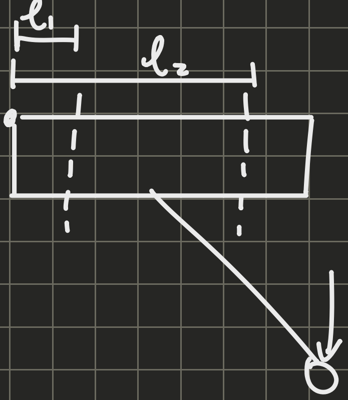
We can find the most loaded bolt and use the same bolt type to all other loads.

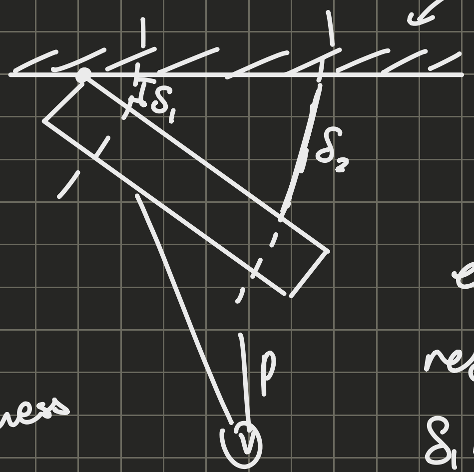
We cannot calculate using isostatic equilibrium since it is hyperstatic.

We can simplify recognizing the symmetry on the rows, so we can consider 2 instead of 4.

Because of the load in this case we can see a moment around c that needs to be balanced

Exagerated for visual





The screw elongate respectively by S_1 and S_2

$$\delta_1 = l_1 \Theta \qquad P_A = \overset{\text{Stiffness}}{k} S_1 = k \Theta l_1$$

$$\delta_2 = l_2 \Theta \qquad P_B = k S_1 = \underbrace{k}_{\text{Unknowns or complicated}} \Theta l_2$$

$\Big\}$ load acting on the single bolt $\Rightarrow$ we will need to x2 for the symmetry

Equilibrium around C: $+\circlearrowright$

$Pl = 2P_A l_1 + 2P_B l_2 \rightsquigarrow k\theta = \dfrac{Pl}{\underbrace{2(l_1^2 + l_2^2)}_{\text{all knowns}}}$ → we have now defined these

⇒ we find P_A and P_B, then we solve the problem
This is a simplified method to avoid doing a full hyperstatic.

Exercise → design of a turnbuckle.

Static
20kN preload
30kN static external load.

Coupler tends to be cast iron → brittle → $R_m = 200$ MPa
Tie rods (carbon steel) → C45 ($R_{su} = 400$ MPa)
Pieces are customized, so no standard threads.
→ So we cannot use strength classes, since strength classes are just for screws not for anything that is threaded.

Because it is subject to a harsh environment, $\eta = 5$ and coarse pitch
→ most material at base (⇒ strongest thread)

$\sigma = \dfrac{F_{max}}{A_{res} \text{ or } A_3} \Rightarrow \sigma_{adm} = \dfrac{400}{5} = 80$ MPa

$A_3 \geq 625\ \text{mm}^2 \Rightarrow$ Rods M36 x 4
→ nominal diameter, external diameter of screw

Once we have sized the rod, now we can determine the T_{AR}, we don't need to calculate T_u because we don't have a head that catches a surface, generating T_u.

We find a τ stress component and the σ stress, associated with external forces.

Then we use Von-Mises and reconfirm $\eta \geq 5$

Now we need to size the buckle.

We need to find the length of the threaded version of the coupler, we do so by evaluating the shear stresses.

$$F_{MAX,rod} = 50000\,N \leq \tau_{amm} \underbrace{(\pi d_3 \overset{\text{unknown}}{l})}_{\text{Area of thread}}$$

$$\tau_{amm} = \frac{R_{sm}}{\eta\sqrt{3}}$$

$$l \geq 33$$

Like for the gears which we thin, we have issues of it acting like paper. Since l is small compared to the nominal diameter, we say that it should be $(1 \div 1{,}25)d \Rightarrow$ we chose $l = 40\,mm$

Sizing the lead screw on the coupler.

$F_{MAX,rod} =$

Since it's brittle we can use Von-Mises, but Galileo - Rankine, which asks us to draw more circles.

<u>Points to remember from exercise:</u>

- → Use the right criterion based on the material.
- → Not all threads are screws so we can't always use strength classes.

Exercise → pressure vestle bolt.

Pressure varies → fatigue

$\eta = 2$

$EI_{flanges} = 4\, EI_{bolt}$

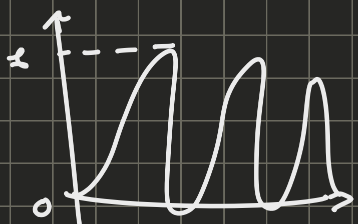
Preload for 5 MPa on gasket.

The stress is symmetric

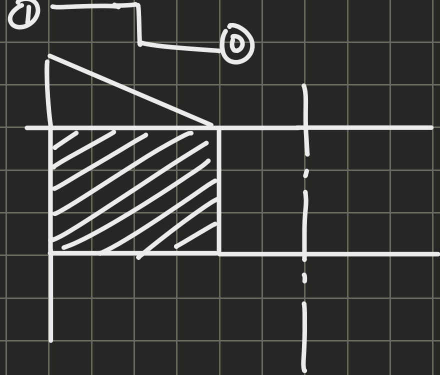
$$\text{If } P_g = 5 = \frac{P_S \cdot n_B}{\pi\underbrace{\left(\frac{D_G^2 - d_G^2}{4}\right)}_{\text{given in pg. 17}}}$$

n_B → number of bolt.

P_S: unknown. force on single bolt



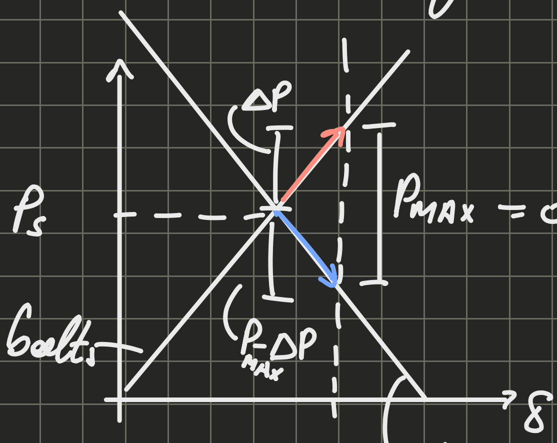
When is service we have a load.
We consider the area of
effect external load on the average area of the gasket.



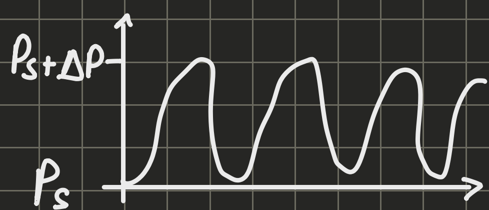
The pressure on each bolt through the load is:

$P_{MAX} = 12026N,$

If this were static they would superimpose.
But due to the gasket effect.



Pressure on bolt is:



unknown

Stiffness bolt

$$\Delta P = P \frac{k_b}{k_b + k_f} = \frac{1}{5} P$$

stiffness gasket/flange

we known $k_f = 4k_b$

$$\Rightarrow P_{b,max} = P_s + \frac{P_{MAX}}{5}$$

$$\Rightarrow P_{b,min} = P_s + \frac{P_{min}}{5} = P_s$$

0 in this case

We the calculate σ, then τ, the do static check and fatigue check

↳ special case, since we have pre load, so we don't start at 0, but we translate along the axis.

Important points:

↳ Haige diagram → not starting from origin

↳ Gasket effect