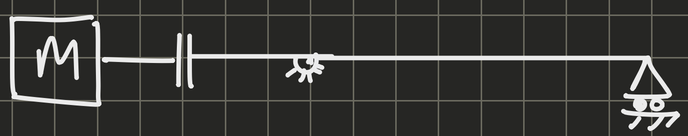
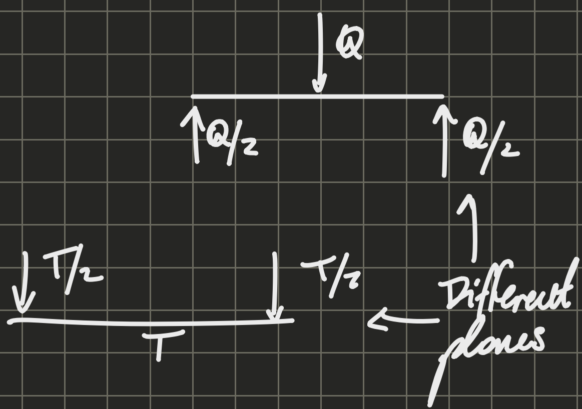
Esercitazione - Bearings

Example → as always we start from the example from the first exercise session.



We never design bearings we select them from a catalogue.



→ No axial loads, so our bearing needs to be able to handle just radial loads.

Since the load is simmetric we only need to solve only one of the bearings. We usually always try to use the same bearings, for supply reasons, so it's easier to buy all the bearings of one type. Some try to find the most loaded bearing and use the one that we choose for that case everywhere. If the are more or less the same we use the same bearing.

Specifications
- ↳ Life = 50000 h
- ↳ Reliability: 96%
- ↳ Lube: ISO VG 150
- ↳ Contamination factor $\eta_c = 0{,}8 \rightarrow$ in a_{23}

Grab bearing catalogue

↳ Operating machine = 50°C

↳ Angular Speed (ω) = 10 rad/s ($n = 95{,}5$ rpm) → what we use.

m = Modified rating life.

rad/s $= \frac{60}{2\pi}$ rpm

Hours of service.

Basic rating life

reliability

$$L_{4mh} = a_1 \cdot a_{23} \cdot L_{10h} = a_1 \cdot a_{23} \cdot L_{10} \cdot \frac{10^6}{60 \cdot n} = a_1 \cdot a_{23} \cdot \left(\frac{C}{P}\right)^p \cdot \frac{10^6}{60 \cdot n}$$

Number of cycles a control bearing can withstand given factors

→ Given with probability of life at 90%.

↳ Basic rating life

↳ Given

a_2 = accounts for quality of material

a_3 = lubrication & operation temperature.

Unknown is C

→ 4 since we are asked 96% reliability.

C is an effect of the bearing, we find it and then we go to the catalogue and find a bearing with C greater than what we calculated.

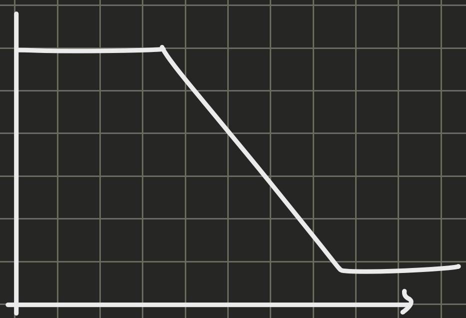
The design requires the bore to have a diameter of 15 mm.

For smaller diameters bearings are made for all integer steps, as they get bigger they begin to skip some numbers. It is possible to have custom sizes for extra money.

In addition to inner diameter sizing we can size the outer diameter. The bigger the bearing the higher the load it can bear.

If we are forced to have high loads with small bearings we have to go back and redesign since these are not fixable together.

Our designs will always have competition in different factors, in many cases they will clash and we will need to do more iterations until we find a design that works.



Bearing have a very low fatigue limit which makes them useless.

We will have fillet radii in the catalogue.
Fillet radii are generally really small, so we have to compensate for that.

How to calculate the different factors:

$$L_{nmh} = a_1 \cdot a_{23} \cdot \left(\frac{C}{P}\right)^p \frac{10^6}{60 \cdot n}$$

dynamic factor

↳ 95,5

C is the unknown

50000

External load.

because only radial load

$$P = F_r = \sqrt{R_{A,xy}^2 + R_{A,xz}^2} = \sqrt{\left(\frac{Q}{2}\right)^2 + \left(\frac{T}{2}\right)^2} = 1118 N$$

$p = 3 \rightarrow$ ball bearings ✓ $p = \frac{10}{3}$ cylinder bearings → X we are using a ball bearing.

- a_1 is taken from a catalogue, where we are given it based on the reliability.

$$a_1(96\%) = 0{,}55$$

$$a_{23} = a_{23}\left(\eta_c \frac{P_u}{P}\right)$$

$\eta_c \frac{P_u}{P}$ — P_u → fatigue limit → the problem is that this is bearing dependent, therefore we need to start a trial & error procedure
↳ In this case we take SKF 6202

P → 1118 N

↳ known: 0,8

To find a_{23} we use the lines based on K

conditions of system, based on lubricant.

$K = \frac{\nu}{\nu_1}$ → standard condition → based on d_m and rpms

↳ $\frac{D+d}{2}$

↳ 0,53 in this case

$\eta_c \frac{P_u}{P} = 160$ ⇐

$a_{23} = 0{,}95$

We now know everything and we can find C

$C = 9151\,N$ → this is the requirement

↳ This is larger than the C for SKF 6202, so it is not the right one, but we have a starting point for the next calculation.

So we find a bearing with higher C than 9151 N, we fix the calculations which we use to find the new C of our system, if we choose SKF 6302 we find a C = 8465 which is less than that for SKF 6302 so it is fine!

If we can not get bigger than a SKF 6202,
we can change:

↳ contamination factor
↳ lubricant
↳ operating temperature

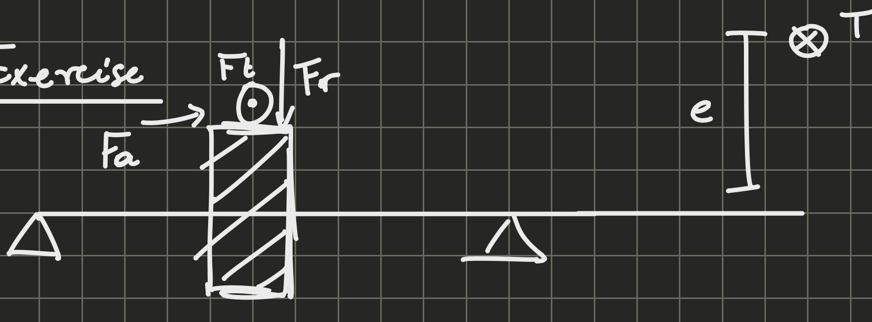
To try and get to what we want.

The final step is to design the bearing in our system such that it creates the constraint we want it to create.

Repercussions on the shaft

↳ The bearing SKF 6302 has a max fillet radius of 1, so we change the design to adhere. We then have to redo the checks for the shaft.
If it fails we can do things like add spacers to forego reducing the radius, but in the worst case we might have to redesign the whole system.

Exercise



T is constant, while F_t is cyclical
F_t and F_r will be fatigue

↳ this will be a mean τ, it will not create alternating components.

For the bearings the fatiguing load is the one that is constant on the shaft, since the outer ring does not spin with the shaft.

$L_{10mh} = 300000$

$\eta_c = 0,2$

ISO VG 68

$T = 60°C$

Reliability = 10%

As a twel we try to go to single file roller bearings
We use the NUP since it's the only one that doesn't allow relative motion, which is good since we are trying to make hinges.

SKF NUP 212

$P = 1751N$

We do the same as the first exercise

We need 30000 h, we get 10^8 h, this is a problem, since it means we can loosen the considerations to reduce it.

Instead of NUP, we use smaller roller bearings which are not NUP, with lower C values, then we will somehow fix everything to create hinge. The other option is to go to ball bearings.