Threaded - Connections

Mechanical connection: machine elements responsible for connecting mechanical components

↳ First classification:

- ↳ Detachable Connection
  - ↳ screws, bolt, pins, keys, splined profiles
- ↳ Fixed Connections (cannot be dissembled without damage)
  - ↳ Welding
  - ↳ Pressfits

Threaded (screw) connections

- ↳ Male screw
- ↳ Lead screw / nut screw / female screws

What makes the connection is the friction and the geometry.

Specific characteristics of screw and connections:

- ↳ components can be made of different materials
- ↳ no special tools
- ↳ most standardised machine elements
- ↳ typically more expensive and heavy relative to other connections.

Basic types of screws pg. 4

↳ Most important
  ↳ Through bolt
  ↳ Tap bolt
  ↳ Stud

There are also set screws, to help define a position.

There are many different kinds of screw heads.

Basic types of nuts: pg. 5

There are solutions to prevent unscrewing.

Screw + Nut (+ washer) = Bolt

Screw Thread Types (all standardized)

↳ Most common for us is the metric thread.
↳ Next most common is Whitworth

Saw tooth and trapezoidal are used for power transmission or for very high loads.

Edison Thread, just for light bulbs.

Metric → M
Whitworth → W

Trapezoid → Tr
Sawtooth → S

ISO grades (strength classes)

10.9 → 90% of Yield Stress ⟹ 10,9 = 900 MPa
↳ 1000 MPa = Yield Stress
↳ $\frac{UTS}{100}$ → Ultimate Tensile Strength / Stress
idk which one

6,8 = 600 · .8 = 480 MPa

The strength class is just a different way to call materials by their strength.

Knowing the class we know the mechanical proprieties.

To each class, corresponds a material.

<u>Never</u> use a nut of a lower grade than a screw, since the system will fail due to the nut.

↳ We design based on the screw, so if the nut is better it doesn't matter.

Metric thread shapes : pg. 8
↳ We use tables to browse standard nuts.

Stress area : $A_{res} = \frac{\pi}{4}\left(\frac{d_2 + d_3}{2}\right)^2$

Like gear with modules we have columns to define how desirable it is to choose that screw, with the third colomn being the least desirable

Minor diameter → base diameter.

Nominal diameter is what we design based on.

Screws are defined by coarse or fine pitch.

The previous table on reported the coarse which is singular to the diameter.

Each diameter has 3 fine pitches.

Coarse Pitch:
- ↳ Faster screwing
- ↳ Less chance of thread damage
- ↳ high strength

Fine pitch
- ↳ Finer position adjustments
- ↳ less tendency to unscrew
- ↳ higher strength of core
- ↳ higher thread increases.

Coarse or fine → 1 coarse or a few fine pitches.
↳ pg. 10

Screws cannot be used for centering, because the position does not require the whole to be within tollerances.

Usually, the holes are:

UP to M24 = D+1,5mm

Above M24 = D+2mm

Nomenclature

Standard +
Type of geometry +
Nominal Diameter
Pitch + → for coarse it can be skipped
Length +
Material grade

Informally we can define just the geometry, diameter and pitch

And for non-standards, we say the diameter, pitch and geometry.

Design (how to choose)

Failure modes
↳ static case
  ↳ yielding and then fracture → occurs at the first thread in contact
↳ Fatigue (cracking) — same reason
  ↳ under the head (since notch)
  ↳ beginning of threaded region
  ↳ first thread in connection between thread and lead thread → this is because of the uneven distribution of load, the first thread takes 50% of the

load.

Strength of screws

↳ Cases:

↳ Not tightened screws (loose)

↳ tightened screws (preloaded)

↳ Freedom of rotation, and unloaded until loaded

↳ we can put a preload, so it is loaded eventhough the machine is not loaded.

↳ The design for the two is different

For designing we will be using nominal stresses, since things like notch effect are not easy to consider.

We will be using fatigue limits for screws that have already been tested.

↳ The notch effect is around 4-10, so we usually try to not use threaded connections for fatigues.

Algorith for sizing and checks: pg.14

Generally pre-sizing (pre-choosing) and checking.

With symmetry we can use the symmetry to find force.
If they are not symmetric, the structure is hyperstatic and there are some tricks we can use to solve.

Loose Screws.

↳ The stress is uniaxial, directed along the axis.

For the static case: $\sigma_a = \frac{P}{A}$

$$\sigma_{MAX} = \frac{F_{A,MAX}}{A_3}$$

→ minor area (to be on safe side

↳ or we can use area of d_2 (weighted average diameter)

For the fatigue case:

$$\begin{cases} \sigma_a = \frac{\Delta\sigma}{2} = \frac{F_{A,max} - F_{A,min}}{2A_3} \\ \sigma_{med} = \frac{F_{A,max} + F_{A,min}}{2A_3} \end{cases}$$

↳ mean stress

↳ alternating stress

A_3 is mandatory in this case

Static Check (as always)

$$\sigma_{MAX} \leq \frac{R_{su}}{\eta} = \sigma_{amm}$$

we use η to select and find η to check

Fatigue

↳ again

$$\sigma_a \leq \frac{\sigma_{lim}}{\eta} = \sigma_{amm}$$

σ_{lim} is given by the standard in this case, based on screw material.

The standard does not consider the effect of σ_{med}, if it is present then we have to use the Haige diagram.

Preloaded Screws:

↳ Preloads are used to compress parts together.

The preload is a design parameter, how do we choose this.

How is the pre-load chosen?

↳ We cannot preload as much as we can since that will cause damage.

↳ We use special tools to set preloads, we use torque wrenches to assign a given torque.

↳ We use torque since it's the only way to load a screw.

<u>Relationship between torque and axial load?</u>

Torque has to combat T_{GA} and T_K

T_{GA} ↓ friction between threads.

T_K ↳ friction

$T_A = T_{GA} + T_K$

↳ Total torque

T_{GA} is what causes the preload, but we cannot ignore T_K.

$$T_u = \mu_u \cdot F_S \cdot \frac{d_w}{2}$$ mean diameter of nut $\approx 1,5D$ of screw.

The preload is $\int dF_S = F_S$

$\int dF_u = 0$ since the direction changes around the axis.

$dF_u = dF_S \cdot \tan\varphi$

$$T_{GA} = \int dF_u \cdot \frac{d_2}{2} = \int dF_S \cdot \tan\varphi \cdot \frac{d_2}{2} = F_S \tan\varphi \cdot \frac{d_2}{2}$$

The component of the friction changes the force triangle, the only thing that changes though is that:

$$dF_u = dF_S \tan(\varphi+\rho) \Rightarrow T_{GA} = F_S \tan(\varphi+\rho)\frac{d_2}{2}$$

Geometric contribution

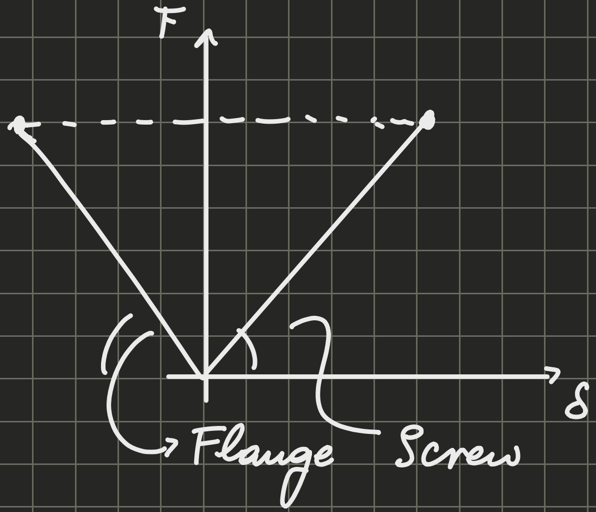
↳ Friction contribution.

Since before we considered square threads, but we are using triangular threads, we consider an apparent friction angle (that considers this geometry and we get:

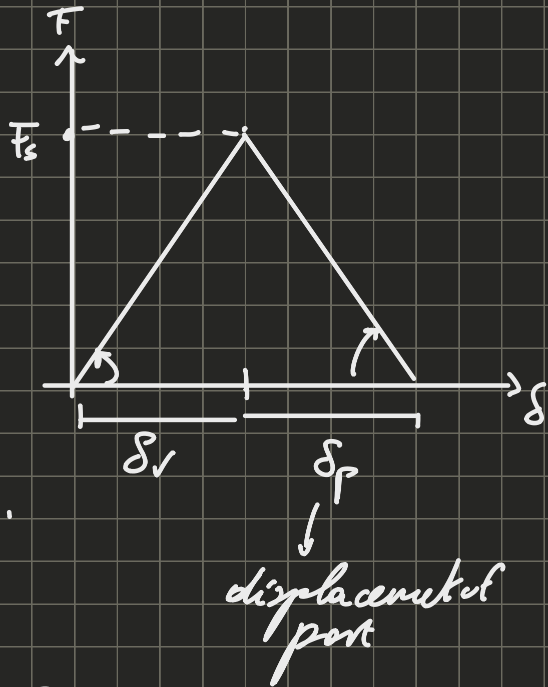
$$T_A = T_u + T_{GA} = F_S\left[\mu_u \frac{d_w}{2} + \tan(\varphi+\rho')\cdot\frac{d_2}{2}\right]$$

Elastic Analysis of loaded connection

Because of the "gasket effect", we cannot pre-load bolts one after another, we need to cross.

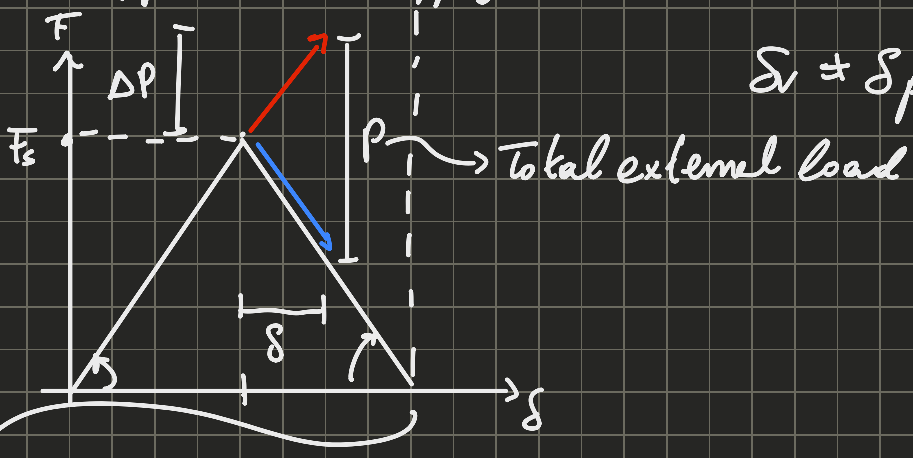


To make more legible



The slope is defined by the stiffness.

What happens when we apply external load?



$\delta_v \neq \delta_p$

δ is the same for the bolt and part.

limit for joint, once we get here the joint fails.

We need to find the force that causes the bolt to fail.

$$k_v = \frac{\Delta P}{\Delta \delta}$$

$$k_p = \frac{P - \Delta P}{\Delta \delta} \Rightarrow \Delta P = P \cdot \left(\frac{k_v}{k_v + k_p}\right) \Rightarrow P_v = F_s + \Delta P$$

we use $F_s + P_v$ to check the bolt and P_{max} to check if the system as a whole is ok.

Static or fatigue calculations → pg. 28+29