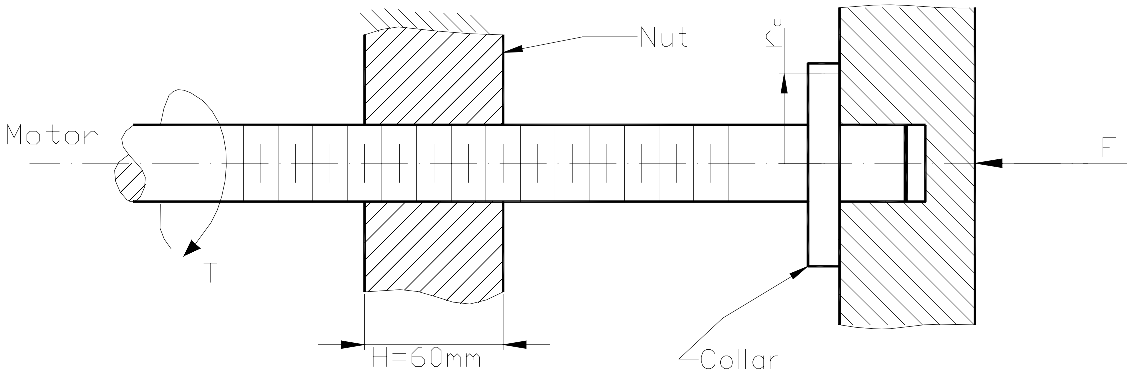
Screws and Threaded Fasteners

1) A single square-thread power screw has an input power of 3 kW at a speed of 1 rev/s. The screw has a major diameter of 36 mm and a pitch of 6 mm. The frictional coefficients are 0.14 for the threads and 0.09 for the collar, with a collar friction radius of 45 mm. **a)** Find the axial resisting load F and the combined efficiency of the screw and collar. **b)** Indicate whether the screw is self-locking or not. **c)** Find the most critical section and determine safety factor assuming that the load is static. Nut and power screw are made of AISI 1040 HR steel. (Do not consider buckling)



a) Root diameter, $d_r = d - p = 36 - 6 \quad d_r = 30\ mm$
Mean diameter, $d_m = d - p/2 = 36 - 6/2 \quad d_m = 33\ mm$
Since screw is single threaded lead is equal to pitch, $l = p = 6$ mm

Input torque is $T = P/n$ $\qquad T = \dfrac{3kW}{1\dfrac{rev}{s}\cdot 2\pi\dfrac{rad}{rev}}$ $\qquad T = 477N.m$

The relation between torque and axial resisting force (F) is, $T = \dfrac{Fd_m}{2}\left(\dfrac{l + \pi\mu d_m}{\pi d_m - \mu l}\right) + \mu_c r_c F$

The first part of this equation is the torque required to resist axial force and thread friction and the second part is the torque required for collar friction.

$$477N.m = F\frac{33\cdot 10^{-3}m}{2}\frac{6\cdot 10^{-3}m + \pi\cdot 0{,}14\cdot 33\cdot 10^{-3}m}{\pi\cdot 33\cdot 10^{-3}m - 0{,}14\cdot 6\cdot 10^{-3}m} + 0.09\cdot 45\cdot 10^{-3}m\cdot F$$

$F = 65kN$

The overall efficiency is: $e = \dfrac{Work\ output}{Work\ input}$

$$e = \frac{Fl}{T2\pi} = \frac{65kN\cdot 6\cdot 10^{-3}m}{477N.m\cdot 2\pi} = 0{,}13 \ \text{ or } \%13$$

b) Self-locking is obtained for square threads when $\mu > tg\lambda$ where $\lambda = tg^{-1}\dfrac{l}{\pi\cdot d_m}$

In this problem $\quad 0.14 > \dfrac{6}{\pi\cdot 33} = 0{,}06$ so screw is self-locking alone.

c) Compressive stress between collar and nut: $\sigma_c = \dfrac{F}{A_r} = \dfrac{4F}{\pi d_r^2} = \dfrac{4 \cdot 65 \cdot 10^3 N}{\pi \cdot (30 \cdot 10^{-3} m)^2}$

$\sigma_c = 92 \ MPa$

Shear stress due to collar torque between collar and nut: $\tau = \dfrac{T_c \cdot c}{J} = \dfrac{16(\mu_c \cdot r_c \cdot F)}{\pi \cdot d_r^{\ 3}}$

$$\tau = \frac{16(263 \cdot 10^3 N.mm)}{\pi \cdot 30^3} \cong 49.6 MPa$$

Combined stress between collar and nut: $\tau_{\max} = \sqrt{\left(\dfrac{\sigma_c}{2}\right)^2 + \tau^2} = \sqrt{\left(\dfrac{92}{2}\right)^2 + 49.6^2}$

$\tau_{\max} = 67.6 MPa$

Shear stress between the motor and the nut: $T_{input} = 477 \ N.m$

$$\tau = \frac{16T}{\pi d_r^3} = \frac{16 \cdot 477 \cdot 10^3 N.mm}{\pi \cdot 30^3} \qquad \tau = 90 MPa \qquad *$$

Average thread shear stress: $\tau = \dfrac{F}{A_s} = \dfrac{F}{\pi d_r \dfrac{p}{2}\dfrac{H}{p}} = \dfrac{65 \cdot 10^3 N}{\pi \cdot 30 \cdot \dfrac{60}{2}} \cong 23 MPa$

Average bearing stress in the screw threads: $\sigma_b = \dfrac{F}{A_b} = \dfrac{F}{\pi\left(d^2 - d_r^{\ 2}\right)\dfrac{H}{p}} = \dfrac{65 \cdot 10^3 N}{\pi\left(36^2 - 30^2\right)\dfrac{60}{6}}$

$\sigma_b = 5.2 MPa$

The most critical section is at the left side of the nut. Here τ due to torsion is 90 *MPa*.

The safety factor is $n = \dfrac{S_{sy}}{\tau} = \dfrac{S_y / 2}{\tau}$ $\qquad$ S_y =290 *MPa* from Table A-20

$$n = \frac{290 MPa}{2 \cdot 90 MPa} = 1.6$$

2) A permanent bolted joint is designed with a M20 coarse-pitch (5.8 grade, rolled-threads) bolt. The bolt is tightened by 30 *kN* preload. Then an external load, P, varying between 10 and 36 *kN* is applied to the joint. **a)** Does separation occur between parts when external load is maximum? If not, at what maximum external load separation occurs? **b)** Find the factor of safety of the bolt against fatigue failure. (Take $k_m/k_b = 3$, R=%99)

a) At separation resultant load on members is zero, $F_m = 0$

$$F_m = \frac{k_m}{k_b + k_m} P_{sep} - F_i = 0$$

$$F_i = \frac{1}{\dfrac{k_b}{k_m} + 1} P_{sep} \qquad P_{sep} = \frac{4}{3} F_i = \frac{4}{3} \cdot 30 kN \qquad P_{sep} = 40 \ kN$$

When a 40 *kN* external force is applied, separation occurs between parts. In this problem the maximum external load is 36 *kN* so separation does not occur.

b) The portion of external load ($P = 10 \ldots 36 \ kN$) taken by bolt is,

$$P_{b\max} = \frac{k_b}{k_b + k_m} P_{\max} = \frac{1}{1 + \frac{k_m}{k_b}} \cdot 36 \qquad P_{b\max} = 9kN \quad , \quad P_{b\min} = 2.5$$

So, the maximum bolt load is $F_{b\max} = P_{b\max} + F_i = 39kN$

And the minimum bolt load is $F_{b\min} = P_{b\min} + F_i = 32.5kN$

Hence the mean and the alternating bolt loads are:

$$F_{bm} = \frac{F_{b\max} + F_{b\min}}{2} = 35.75kN \qquad F_{ba} = \frac{F_{b\max} - F_{b\min}}{2} = 3.25kN$$

Tensile strength area of M20 coarse-pitch bolt is $A_t = 245\ mm^2$ (Table 14-2).

$$\sigma_a = \frac{Kf^*F_{ba}}{A_t} = \frac{7.15kN}{245mm^2} = 29.2\,MPa \qquad \sigma_m = \frac{F_{bm}}{A_t} = \frac{35.75k.N}{245mm^2} = 145.92MPa$$

From Table 14-7, S_{ut} =520MPa K_f = 2.2 (Table14-8, grade 5.8, rolled threads, C_{load} = 0.7, $C_{surf} \cong 0.86$ (Table14-7), remaining factors are unity (note that reliability factor = 0.814).

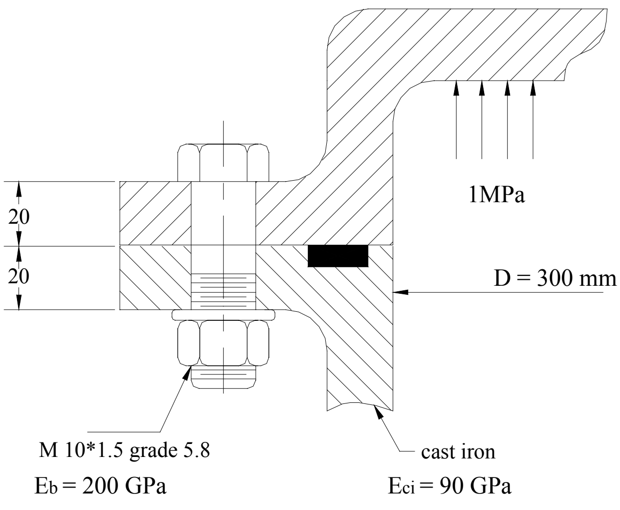
$$\sigma_i = \frac{Kfm^*F_i}{A_t} = \frac{1^*30kN}{245mm^2} = 122.4\,MPa$$

$$S_e = 0.7 \cdot 0.86 \cdot 0.814 \cdot (0.5 \cdot 520MPa) = 127.4MPa$$

Plot the modified Goodman diagram and the loading line as shown in Fig. 13-17

n = 2.8 (Using Eq. 14-16 or from geometry)

3) The head of a 300 *mm* diameter cylinder vessel will be designed. The internal pressure in the cylinder is P_{in} = 1MPa. The bolts have initial tightening load of 12 *kN*. **a)** Find the stiffness of the bolt and members. **b)** Determine the number of bolts (N) for a safety factor of 2 for fatique loading.



a) If the members of the joint have the same modulus of elasticity and assuming $\alpha = 45°$, we can find the stiffness with the formula below.

$$k_m = \frac{\pi E_{ci} d}{2\ln\left(5\cdot\frac{l+0.5d}{l+2.5d}\right)}$$

l_{grip} = 20+20=40 *mm*

$$k_m = \frac{\pi\cdot 90\cdot 10^9\cdot 10\cdot 10^{-3}}{2\ln\left(5\frac{40+0.5\cdot 10}{40+2.5\cdot 10}\right)} = 1140\,\text{kN/mm}$$

The bolt stiffness is,

$$k_b = \frac{A_b E_b}{l} = \frac{\pi d^2 E_b}{4l} \qquad k_b = \frac{\pi\left(10\cdot 10^{-3}\right)^2\cdot 200\cdot 10^9}{4\cdot 40\cdot 10^{-3}} = 393\frac{kN}{mm}$$

Stiffness of the members can also be calculated by other approaches.
(see Section 14-8)

b) External force per bolt: $P = \frac{P_{in}\cdot A}{N} = \frac{10^6\cdot\pi\cdot 0.3^2}{4\cdot N} = \frac{70.68}{N}kN$

Resultant bolt load: $F_b = \frac{k_b}{k_b+k_m}P + F_i \qquad F_b = \frac{393}{393+1140}\cdot\frac{70.68}{N} + 12 = \frac{18.14}{N} + 12\ \ kN$

Tensile stress area for M10×1.5 is A_t =58 mm^2 (Table14-2)
S_y =520 *MPa* (Table14-7)

$$\sigma_y = \frac{F_y}{A_t} \qquad F_y = \sigma_y\cdot A_t \qquad \sigma_{all} = \frac{S_y}{n} \qquad F_{all} = \frac{S_y}{n}\cdot A_t$$

$$F_{all} = 58mm^2\cdot\frac{520}{2}MPa \qquad F_{all} = 15.08kN = \frac{18.14}{N} + 12kN$$

N = 5.89 so, use 6 bolts.

Preload must be checked against separation.(F_m = 0)

$$F_m = \frac{k_m}{k_m+k_b}\cdot\frac{P}{N} - F_i = 0 \qquad F_i = \frac{1140}{1140+393}\cdot\frac{70.68}{6} = 8.76 < 12kN$$

Preload is enough.