Introduction to Engineering Design

ME 310: Mechanical Design

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Outline

Overview of Bolted Joints

Bolt Geometry and Properties

Bolt Stress Analysis

Bolted Joint for Fatigue Loading

Bolted Joints

Bolted joints are held by threaded fasteners

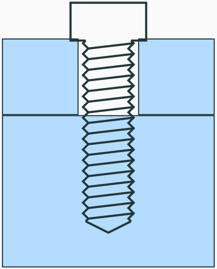
- Bolts
- Screws
- Nuts

What's the difference between a bolt and a screw?

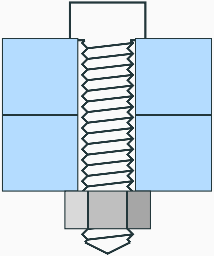
How Bolted Joints Work

- Clamped members are held together by compressive load from a bolt or screw
- Bolt itself is under tensile load

Applications of Bolted Joints

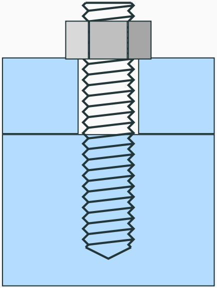


(a) Screw

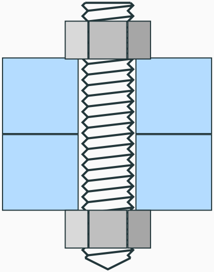


(b) Bolt and nut

Application of Bolted Joints 2



(c) Stud and nut



(d) Threaded rod

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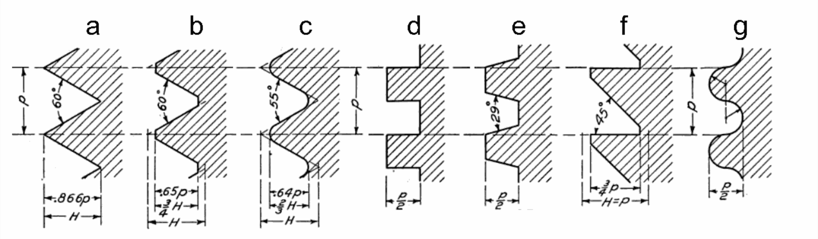
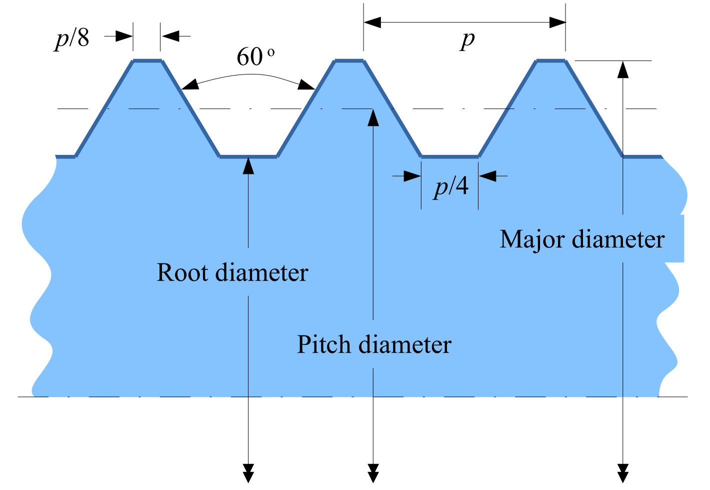
Thread Geometry



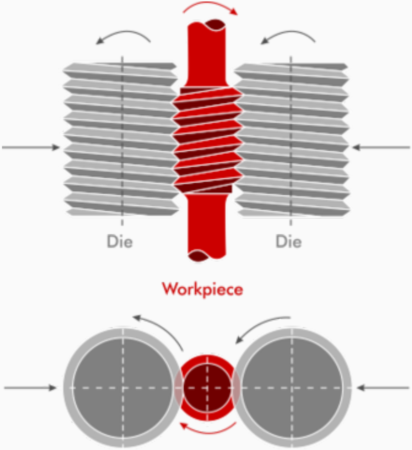
Thread Forms

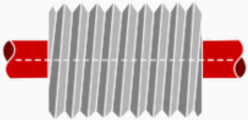


Standard ISO Bolt Sizes

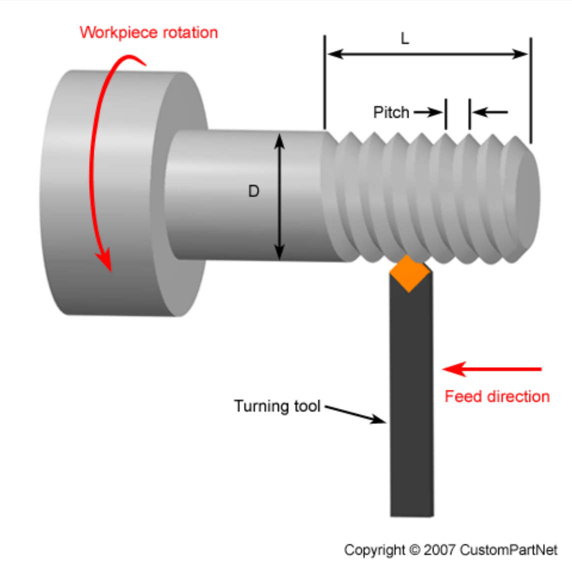
Nominal Diameter d	Coarse Threads			Fine Threads		
	Pitch	Minor ⌀	Stress Area	Pitch	Minor ⌀	Stress Area
3	0.5	2.39	5.03	-	-	-
3.5	0.6	2.76	6.78	-	-	-
4	0.7	3.14	8.78	-	-	-
5	0.8	4.02	14.2	-	-	-
6	1	4.77	20.1	-	-	-
7	1	5.77	28.9	-	-	-
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030
42	4.5	36.9	1121	-	-	-
48	5	42.7	1473	-	-	-

How are Screw Threads Made?: Thread Rolling

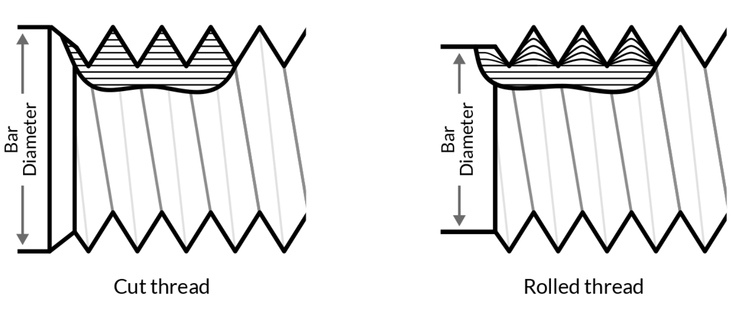




How are Screw Threads Made?: Thread Cutting



Advantage of Rolled vs Cut Threads



Strength of Bolts

- Because of threads and how they are formed, bolts are not loaded to their yield or ultimate tensile strength.
- Instead, bolt strength is measured by a *proof strength*, S_p.
- For approximation, $S_p = 0.9S_y$

Bolt Material Strength

SAE Class	Diameter d (mm)	Proof Strength S_p (MPa)	Yield Strength S_y (MPa)	Tensile Strength S_{ut} (MPa)	Elongation (%)	Reduction of Area (%)
4.6	5 – 36	225	240	400	22	35
4.8	1.6 – 16	310	-	420	-	-
5.8	5 – 24	380	-	520	-	-
8.8	17 – 36	600	660	830	12	12
9.8	1.6 – 16	650	-	900	-	-
10.9	6 – 36	830	940	1040	9	9
12.9	1.6 – 36	970	1100	1220	8	8


8.8
10.9

Outline

Overview of Bolted Joints

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Bolt Stress Analysis

Bolted Joint for Fatigue Loading

Bolted Joint Stress Analysis

- Two modes of failure
- Thread failure $\rightarrow$ shear stress in thread
- Tensile faiure $\rightarrow$ tensile stress in bolt

Shear Stress in Threads

- Thread surface area is

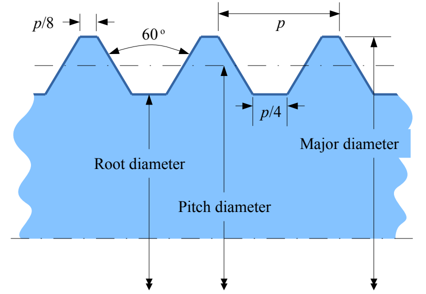
$$A_{shear} = \pi d(0.75t)$$

- Allowable shear stress on bolt (MDET)

$$\tau_{allow} = \frac{S_y}{\sqrt{3}} = 0.577S_y$$

- Force to cause thread failure

$$F_{thread} = 0.577S_y\pi d(0.75)t$$



Bolt Tensile Stress

- Bolt tensile area

$$A_t \approx \frac{\pi}{4}\left(0.9d\right)^2$$

- Tensile load to yield the bolt threads

$$F_{bolt} = A_t S_y \approx \frac{\pi}{4}(0.9d)^2 S_y$$

Preventing Thread Failure

- To prevent thread failure, make sure tensile failure happens first (or at the same time)
- Setting $F_{bolt} = F_{nut}$

$$t = 0.47d$$

- The thickness nut or depth of threaded hole should be at least half major diameter.
- Now we can worry only about tensile failure

Torque - Tensile Load Relationship

- Setting $\mu = 0.15$

$$T = 0.2 F_i d$$

- Metal on metal friction is 0.15?

Proper Bolted Joints

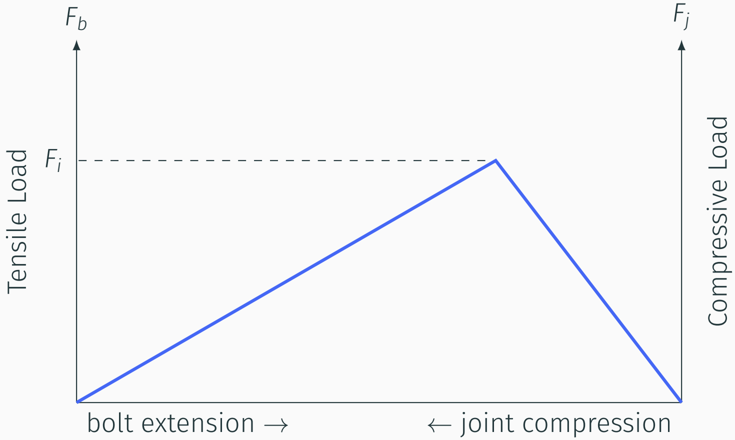
- Correct bolt length and proper nut/threaded hole thickness

$$t \geqslant 0.47d$$

- Proper tightening with calculated torque $\rightarrow$ torque wrench is your friend
- Locking mechanism: locknuts, slotted nuts, two nuts, toothed lock washers.

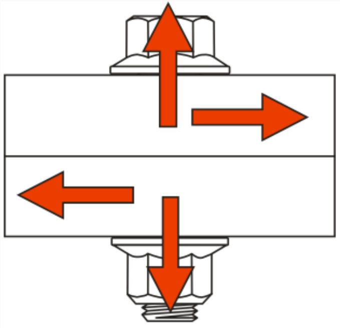


Bolt-Joint Interaction

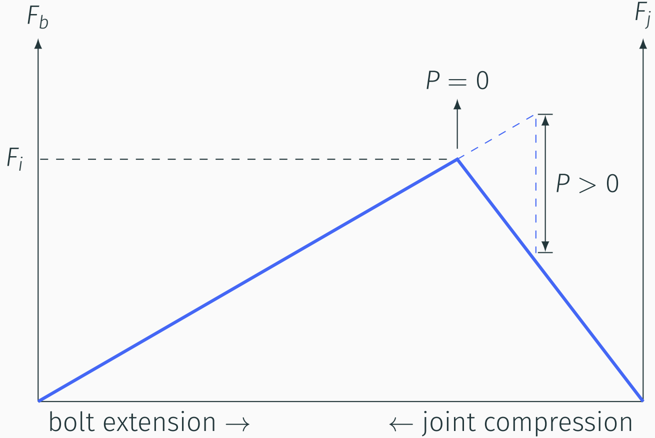


Types of Bolted Joints

- Tension Joints
- Friction Joints



Load Distribution in Tension Joints



Load Distribution in Tension Joints with External Tensile Load

From previous diagram

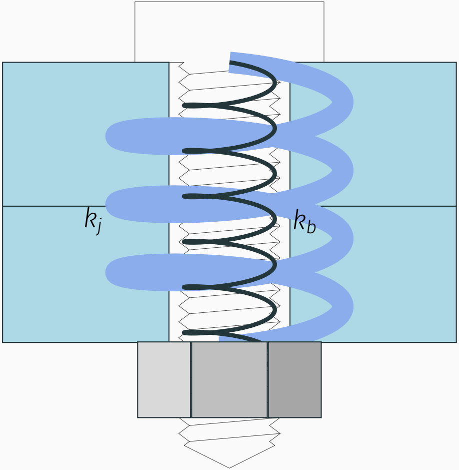
$$F_b = P_b + F_i = CP + F_i$$

$$F_j = (1 - C)P - F_i$$

where $C = k_b/(k_b + k_j)$.

Load carried by bolt and clamped members depend on stiffness ratio

Bolted Joint Stiffness



$$k_{total} = k_b + k_j$$

Bolt Size Selection for Tension Joints

The only job we have is to keep the clamped part together

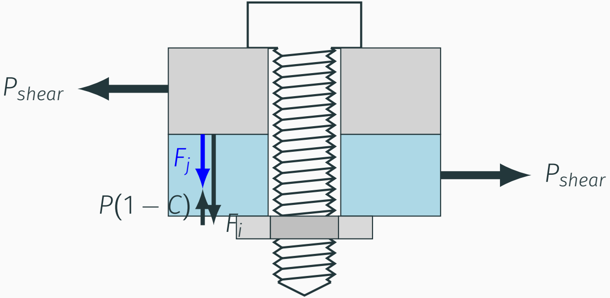
$$F_j = 0 = (1 - C)P - F_i$$
$$P = \frac{F_i}{1 - C}$$

For non-permanent joints, $F_i = 0.75 S_p A_t$ and $C \approx 0.25$

For the bolted joint to have a safety factor of N_s

$$A_t = \frac{N_s P}{N S_p}$$

Friction Joints



- Prevent members from sliding by *friction* not interference
- Compressive load exerted by bolt generate friction

Bolt Size Selection for Friction Joints

$$P_{shear} = -\mu F_j$$

Again assuming \$ $\mathrm{F_i}$ = 0.75 $\mathrm{S_p}$ $\mathrm{A_t}$\$ and $C = 0.25$

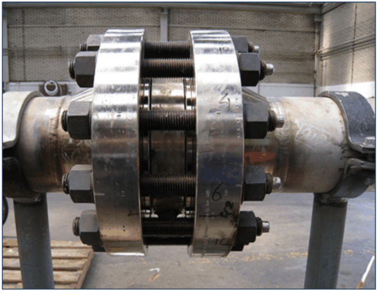
$$\begin{aligned}
P_{shear} &= -\mu F_j = -\mu \left(P(1 - C) - F_i \right) \\
&= \mu(0.75 S_p A_t - 0.75 P) \\
A_t &= \frac{1}{S_p}\left(\frac{P_{shear}}{0.75\mu} + P \right)
\end{aligned}$$

For safety factor N_s with N bolts

$$A_t = \frac{N_s}{N S_p}\left(\frac{P_{shear}}{0.75\mu} + P \right)$$

Flange Joints

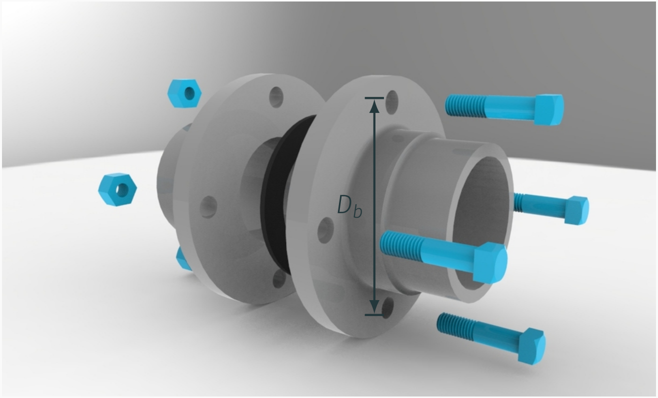
- Circular bolt arrays for sealing purpose.
- Mainly used for pipe connections and pressure vessels



Gasketed Flange Joints

- soft materials: rubber, plastic, or soft metals between clamped member
- How does that help?
- How does it effect the joint strength?

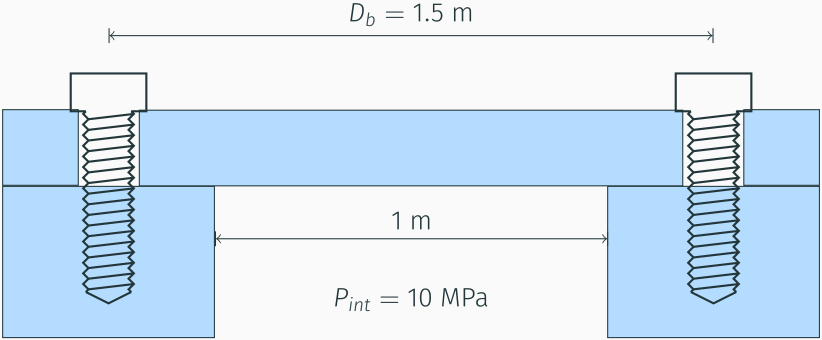
Equation for Flange Joints



$$3 \leqslant \frac{\pi D_b}{Nd} \leqslant 6$$

Flange Joint on Pressure Vessel Example

A cylindrical pressure vessel is pressurized to 10 MPa. The cross section of the vessel is shown below. The flange joint to keep the cover on the vessel is made up of 12 grade 12.9 M8 coarse thread bolts. Is this joint safe? If not, determine the proper bolt size.



Solution

- Assume constant pressure, N_s = 2

$$A_t = \frac{PN_s}{S_p}$$

- The vertical load comes from the pressure force, which is

$$P_{total} = p\pi r^2 = 10 \times 10^6 \times \pi \times 0.5^2 = 7.85 \text{ MN}$$

Solutions

- We will assume that the load is distributed evenly on all 12 bolts, so that
$$P = \frac{P_{total}}{N} = \frac{7.85\ \text{MN}}{12} = 6.54 \times 10^5\ \text{N}$$
- Assume that we use the bolt material grade 12.9 whose proof strength is 970 MPa, the required tensile area is

$$A_t = \frac{(2)(6.54 \times 10^5)}{970 \times 10^6} = 1.35 \times 10^{-3}\ \text{m}^2 = 1350\ \text{mm}^2$$

Solutions

- The calculated required tensile area, even with the strongest material (grade 12.9), is larger than M8 coarse thread (36.6 mm^2) and therefore the given design is unsafe!
- Redesigning the flange joint, it must satisfy the sizing equation above and the bolt distance equation, namely

$$3 \leqslant \frac{\pi D_b}{Nd} \leqslant 6$$

- Since D_b = 150 cm, set the inequality to 4.5 to solve.

$$\frac{\pi D_b}{Nd} = 4.5$$
$$Nd = 1.05$$

Solutions

- Reapply the tension joint equation to determine the total required tensile area, we have

$$NA_t = \frac{N_s P}{S_p} = \frac{2(7.85 \times 10^6)}{970 \times 10^6}$$
$$= 1.62 \times 10^{-2}\ \text{m}^2$$

Solutions

- 2 equations, 3 unknowns
- Are the unknowns all independent?

$$A_t \approx \frac{\pi}{4}(0.9d)^2$$

$$NA_t \approx N(0.81)\frac{\pi}{4}d^2 = 1.62 \times 10^{-2}$$

$$Nd^2 = 2.58 \times 10^{-2}$$

Solutions

- Solving the system of equations, we have

$$\frac{Nd^2}{Nd} = \frac{2.58 \times 10^{-2}}{1.05} = 0.0245 \text{ m} = 24.5 \text{ mm}$$

Solutions

- Since there is no standard bolts with that size, we pick the next larger bolt, M27 $\times$ 2 (fine thread)
- Finally, A_t = 496 mm^2 (from M27) is used to solve for the number of required bolts

$$N = \frac{1.62 \times 10^{-2}}{496 \times 10^{-6}} = 25.4$$

- Therefore, we need 26 M27 $\times$ 2 for this flange joint.

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Bolt Sizing for Fatigue Loading

- If properly tightened, endurance limit of bolt is constant regardless of average stress

$$N_s = \frac{S_e}{\sigma_a}$$

$$\sigma_a = \frac{CK_f(P_{\max} - P_{\min})}{2NA_t}$$

Bolt Strength under Fatigue

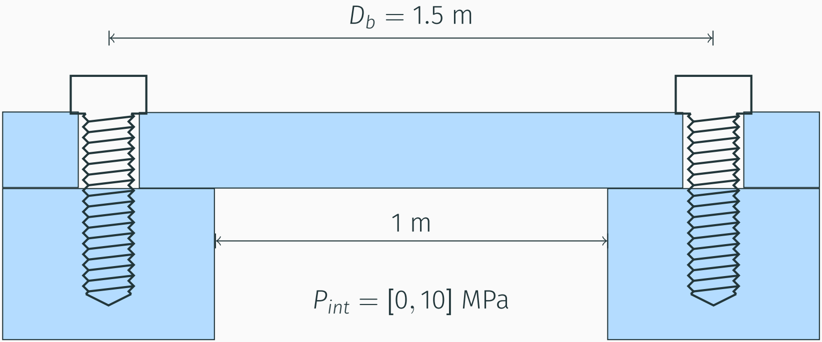
Material Grade	Endurance Limit (S_e)
8.8	129 MPa
9.8	140 MPa
10.9	162 MPa
12.9	190 MPa

Stress Concentration Factor of Bolts

Hardness	SAE Class (ISO Threads)	K_f Rolled threads	K_f Cut threads
Below 200 Bhn (annealed)	5.8 and below	2.2	2.8
Above 200 Bhn (hardened)	8.8 and above	3.0	3.8

Bolted Joint Design under Fatigue Example

A cylindrical pressure vessel is pressurized and depressurized repeatedly between 0 and 10 MPa during its operation. The cross section of the vessel is shown below. Design a proper flange joint with a safety factor of 2.



Solution

- Definitely fatigue related
- Max load = 7.85 MN, from previous example
- Min load = ?

$$
\begin{aligned}
\sigma_{\text{max}} &= \frac{CK_f P_{\text{max}}}{NA_t} = \frac{0.25(3)(7.85 \times 10^6)}{NA_t} = \frac{5.89 \times 10^6}{NA_t} \\
\sigma_{\text{min}} &= 0 \\
\sigma_a &= \frac{2.94 \times 10^6}{NA_t}
\end{aligned}
$$

Solutions

- Grade 12.9 $\rightarrow S_e$ = 190 MPa

$$N_s = 2 = \frac{S_e}{\sigma_a} = \frac{190 \times 10^6}{\frac{2.94 \times 10^6}{NA_t}}$$

$$NA_t = 3.10 \times 10^{-2}\ \text{m}^2$$

- Check for yield under tensile load.

$$NA_t = \frac{N_s P}{S_p} = \frac{2(7.85 \times 10^6)}{970 \times 10^6} = 1.62 \times 10^{-2}\ \text{m}^2$$

Solutions

- Fatigue requires larger area $\rightarrow$ deciding factor

$$\frac{\pi D_b}{Nd} = 4.5$$
$$Nd = 1.05$$

- We can then solve for the required diameter.

$$NA_t = N(0.8)\frac{\pi}{4}d^2 = 3.10 \times 10^{-2}$$
$$Nd^2 = 4.93 \times 10^{-2}$$
$$\frac{Nd^2}{Nd} = \frac{4.93 \times 10^{-2}}{1.05} = 0.047 \text{ m}$$

- The required bolts are then M48 $\times$ 5, and the required number of bolts is

$$N = \frac{3.10 \times 10^{-2}}{1473 \times 10^{-6}} = 21$$

- So we would need 21 M48 $\times$ 5 bolts.

Final Notes on Bolted Joint Design for Fatigue Loading

- Properly tighten bolt
- Choose rolled threads whenever possible

Any Questions?