University of Nottingham
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Design and analysis of bolted joints

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Learning Objectives

Part 1:

- To be familiar with the types of joints including joining technologies, loading conditions, and design considerations of bolted joints

Part 2:

- To be able to determine the stiffness of the bolt & clamped members, resultant loads of pre-tensioned joint
- To be able to design against joint failure & to calculate reserve factor of a pre-tensioned joint
- To understand the behaviour of pre-tensioned joint under cyclic loading

Part 3:

- **Worked examples**

Function & types of Joints/Fasteners

- A joint or fastener is a device used to connect or join two or more components.

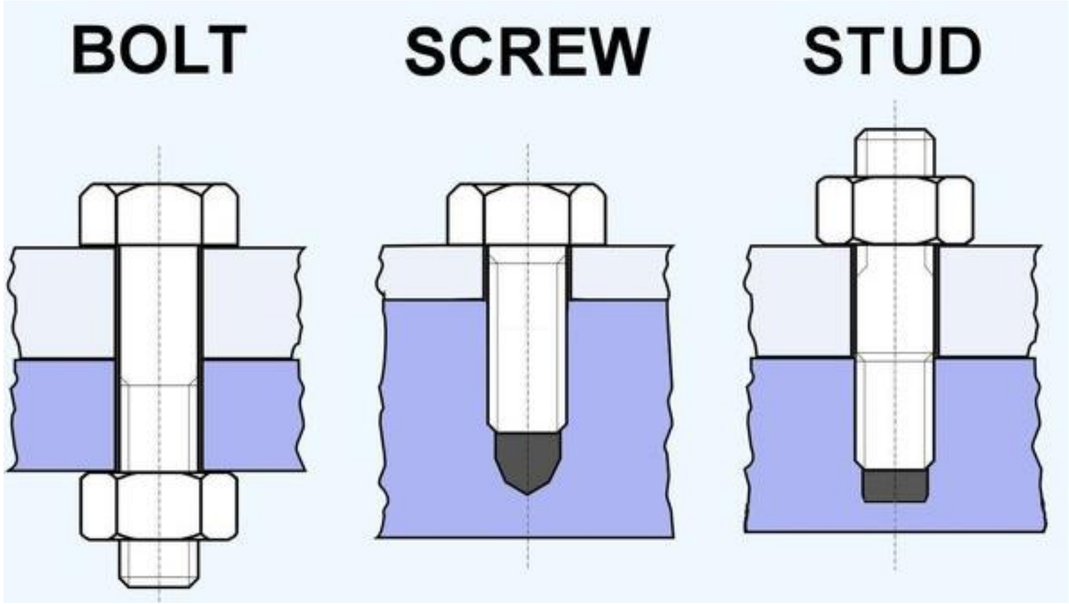
  A layman might consider **threaded fasteners** uninteresting machine elements, but it is **unimaginable** that most machines & structures can be built without.

- **Non-permanent joints**
  – Screws, threaded and unthreaded fasteners and setscrews, etc.
- **Permanent joints**
  – Rivets, welded and adhesive joints, friction joints, etc.
  – Fasteners used as permanent joints that are never disassembled.

A bolt, a screw or a stud?

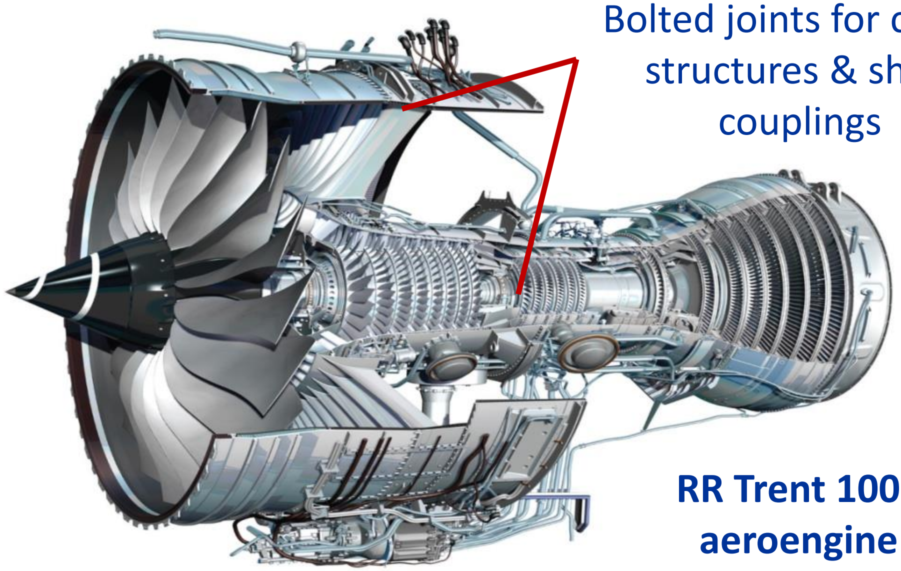
Bolted joint Part 1

- A **bolt** is a threaded fastener mated with a nut.
- A **screw** has either pre-formed or self-made internal threads.
- A **stud** is an externally threaded headless fastener. One end mates with a tapped component and the other with a standard nut.



Bolts, screws and studs are all commonly used in different applications, the general design consideration and method are similar.

Examples of bolted joints

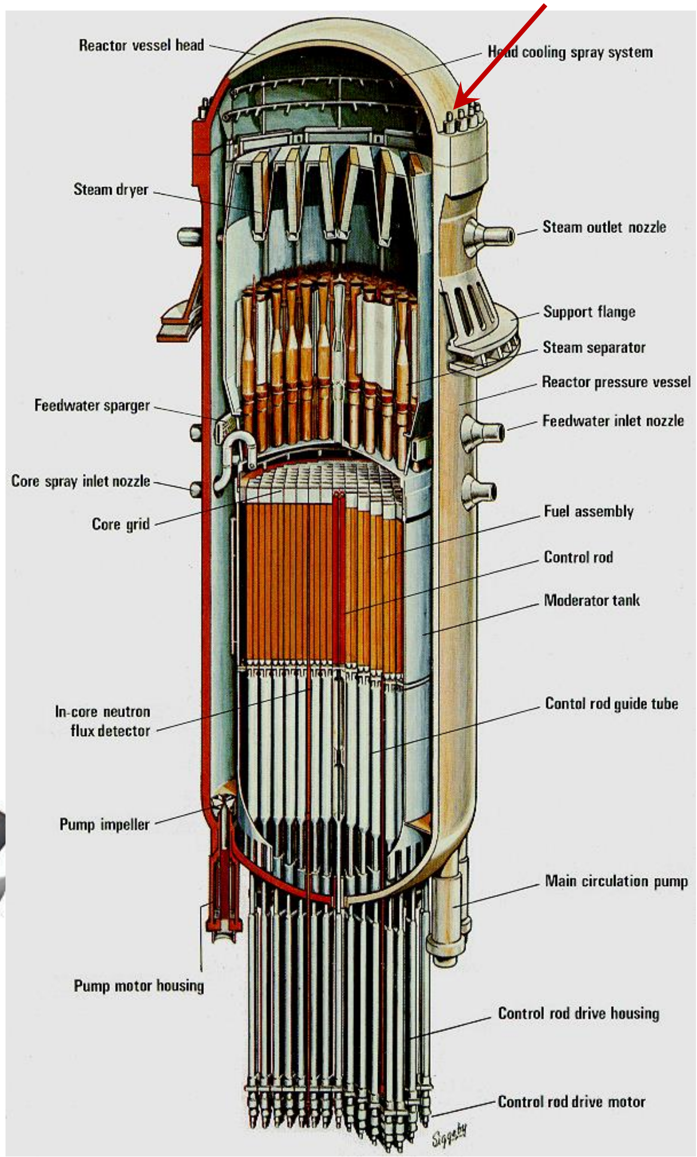


Bolted joints for casing structures & shaft couplings

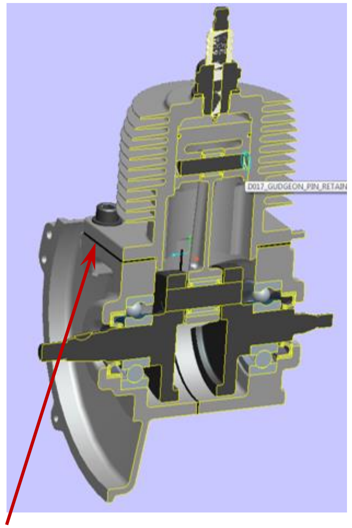
RR Trent 1000 aeroengine

Wind turbine blades joined by bolts to the hub

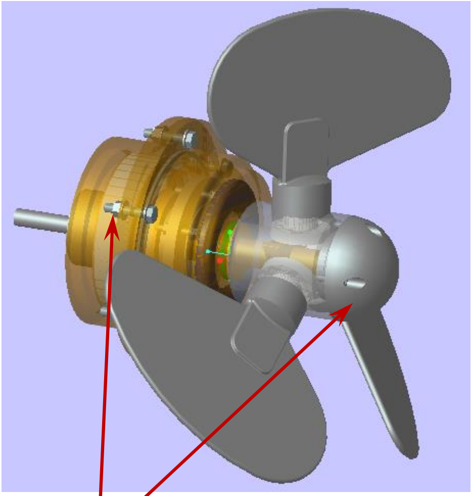
Bolted joints (studs)



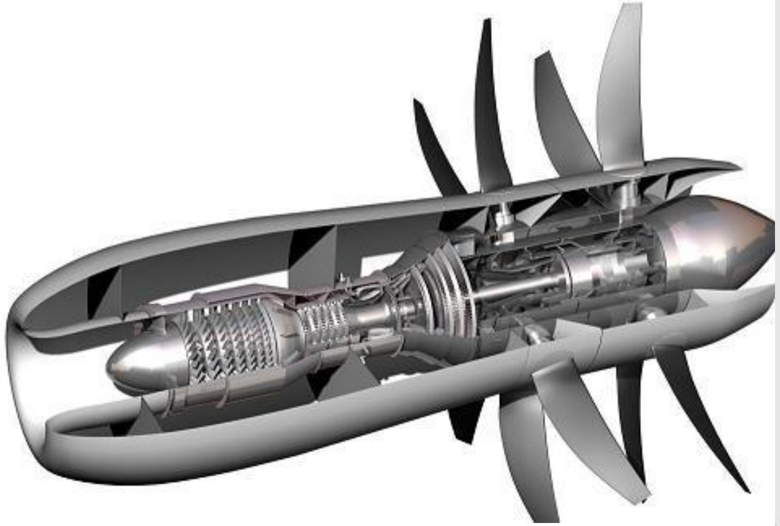
Nuclear Reactor Pressure Vessel



Bolted joints (socket screw) of a two-stroke engine



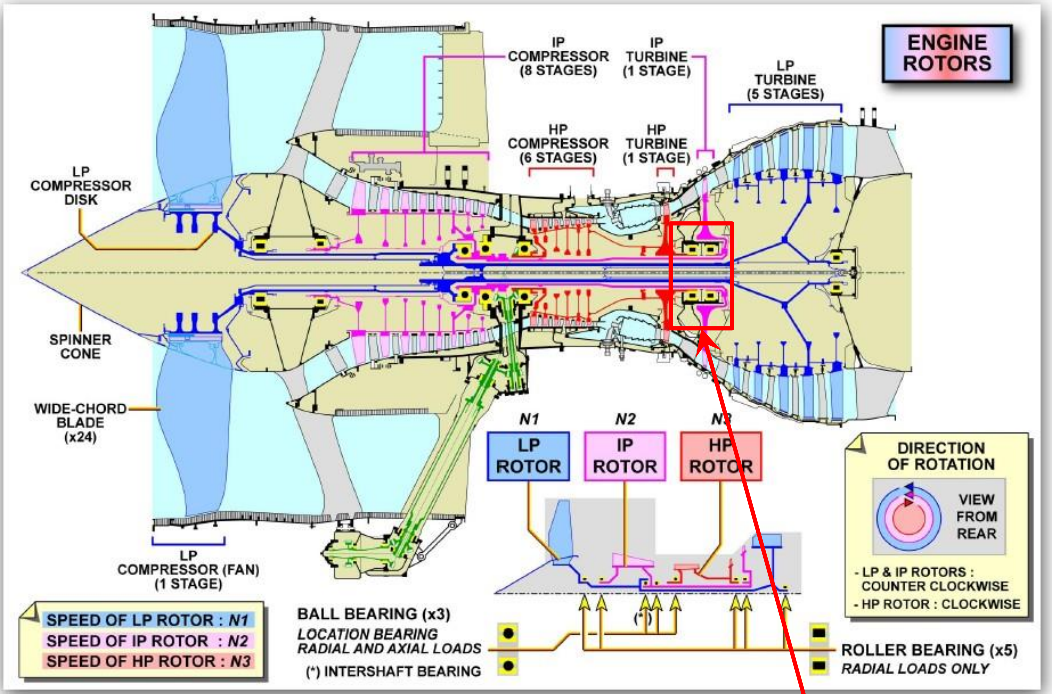
Bolted joints of a variable pitch propeller

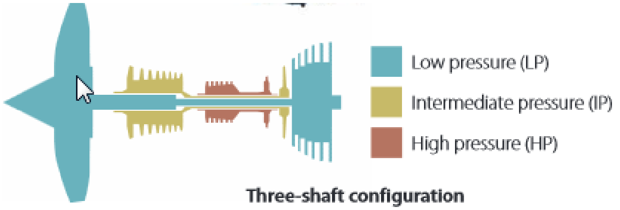


A past research project with RR in design optimisation of Open rotor jet engine

Labyrinth seals in aeroengine applications

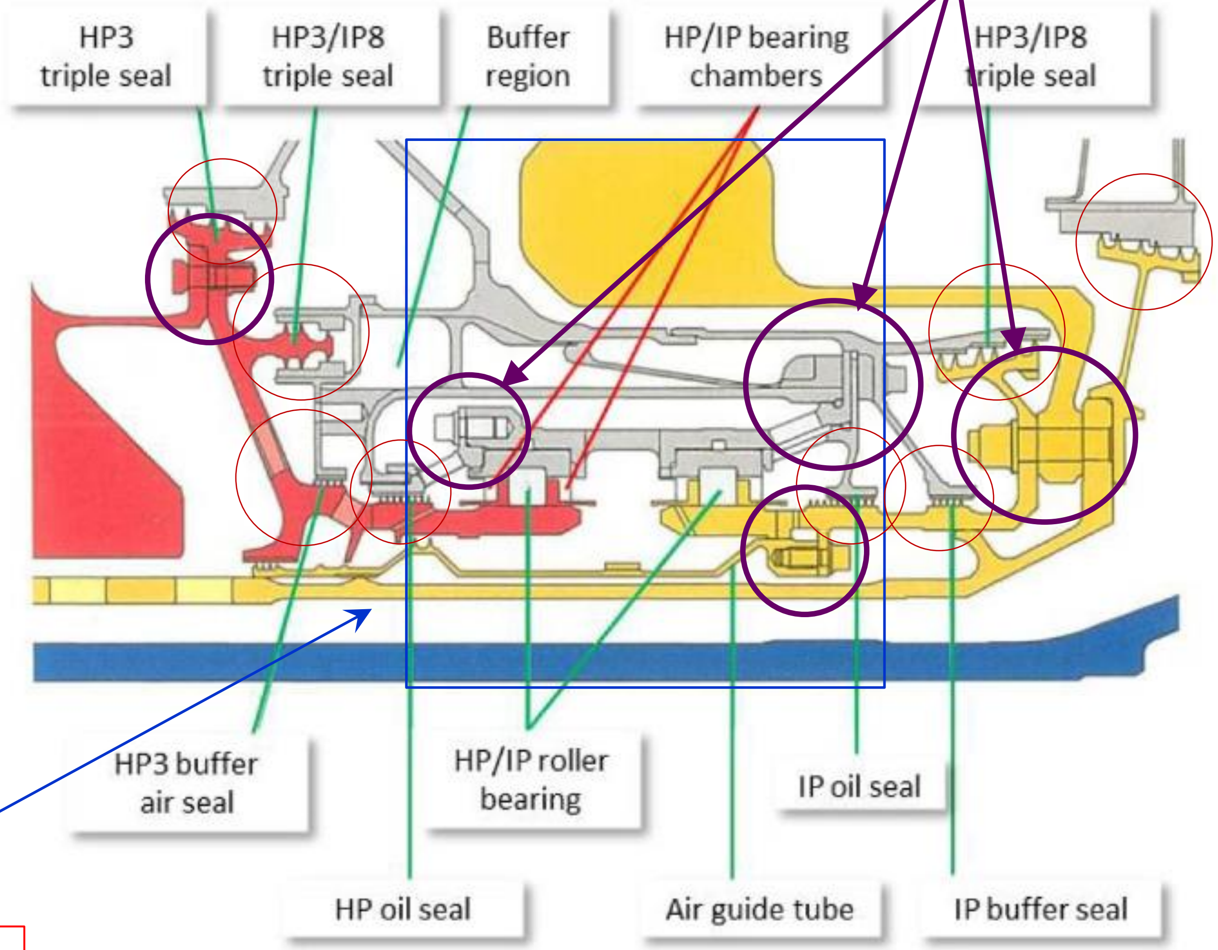
- **Rolls-Royce Trent 1000 three shaft jet engine**





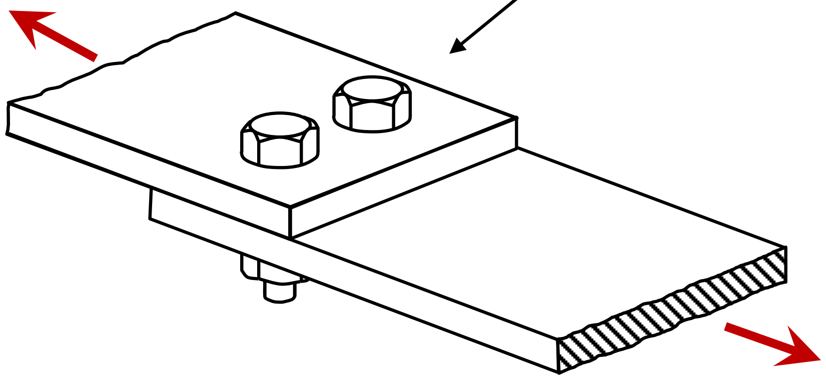
HP-IP bearing chamber

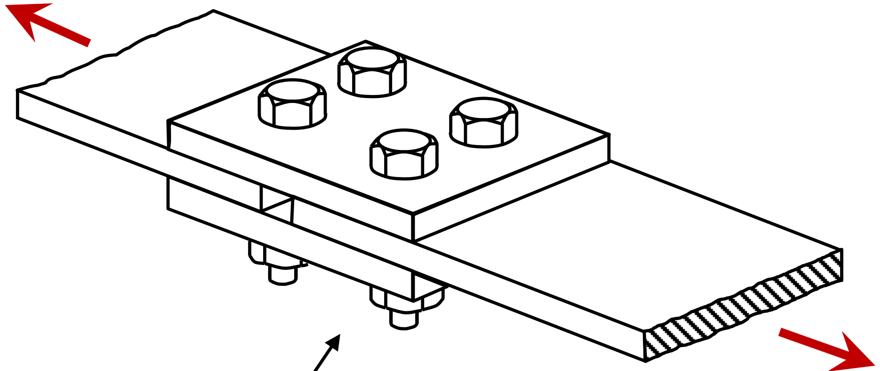
Bolted joints (studs)



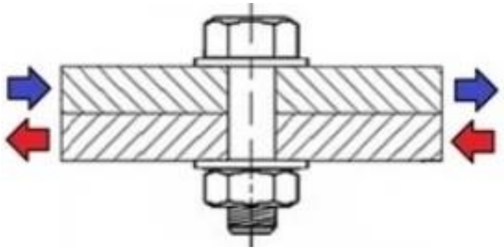
Shear joints

Components to be joined overlap each other in a **lap joint**

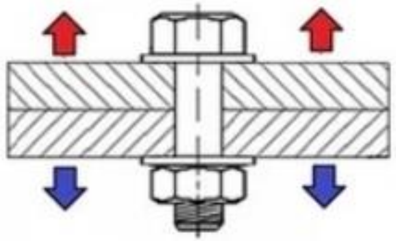




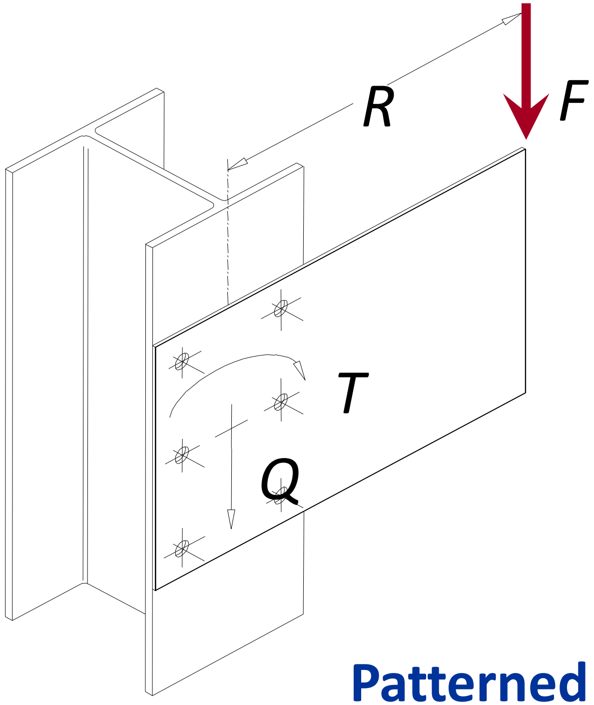
An additional component is used to bridge the gap in a **butt joint**



Shear joints



Bolted joints under axial loading



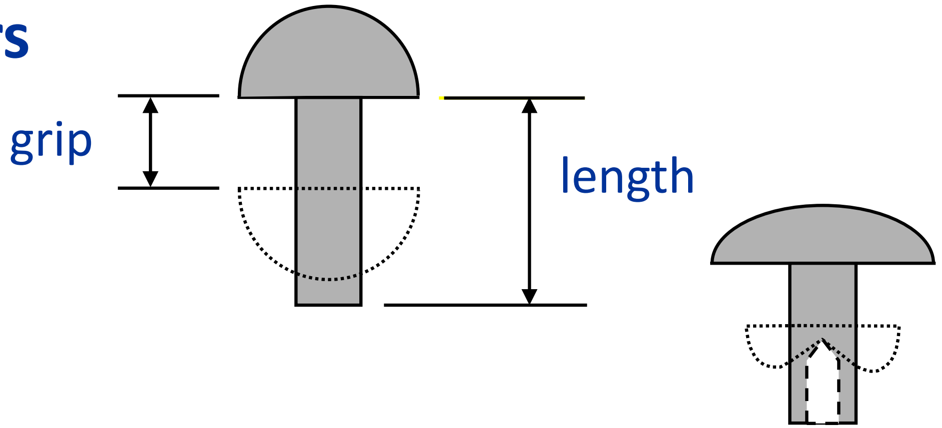
Patterned joints

Note: Design and calculation of shear joints should use a different approach. It is not covered in this session.

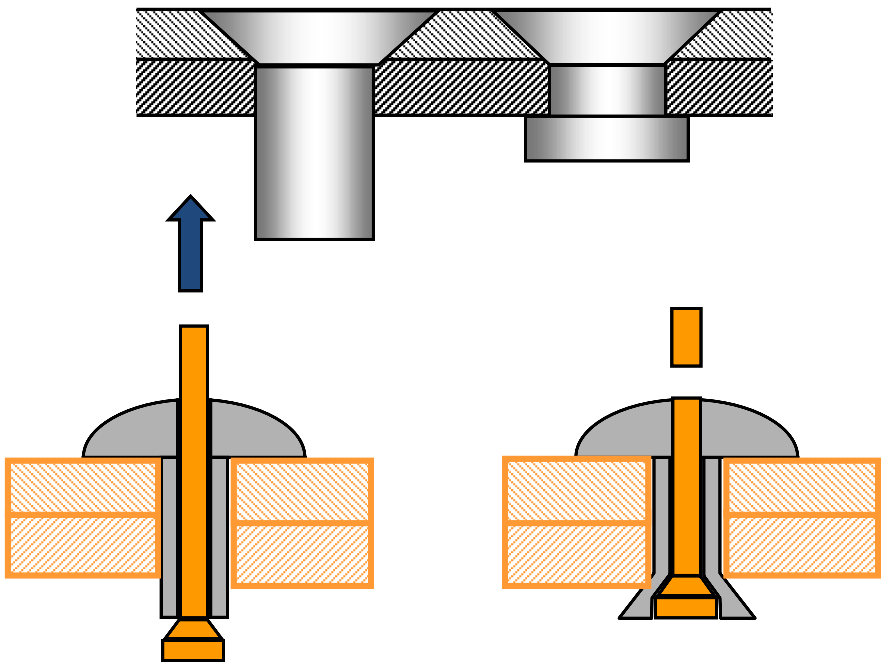
Refer Shigley's Mechanical Engineering Design, Ch8 for detail

Rivets

Rivets are non-threaded fasteners



- **Solid rivets** – assembled from one side and 'upset' on the other
  - **BS 4620: 1970** for general purposes
  - For airframe applications BS A 361, 351(Ni), 362(Ti).
- **Tubular rivets** have a hole down the axis – upsetting is easier
- **Blind rivets** are tubular and upset from the installation side



Bolted joint Part 1

Examples of Riveted structures

7,500,000 rivets used

More example of Rivets

- Rivets remain an important method of assembly of aircraft structures
- More or less permanent, they don't work loose
- But can be drilled out if repairs are needed
- **Boeing 747** uses over 2.5x 10^6 fasteners

Bolted joint Part 1

Rivets

- Advantages:
  - Low cost
  - Rapid assembly (thousands/hour when automated)
  - Permanent (but can be drilled out)
  - Can join dissimilar materials (watch out for galvanic corrosion)
  - Wide range of shapes and materials
- Disadvantages:
  - Slow compared with welding and adhesives
  - **Shear loading (poor in tensile loading)**
  - Joints leak unless sealed

Bolted joint Part 1

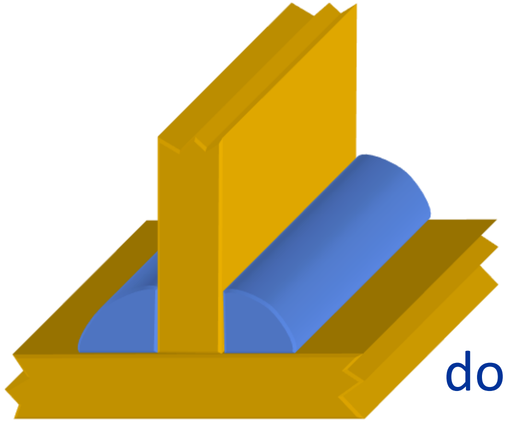
Welded joints

Welding is a fabrication process that joins materials by using high heat to melt the parts and filler material together.

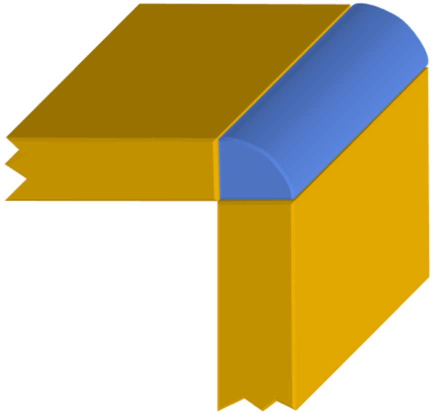
Gas flame, **electric arc**, **laser**, **electron beam**, **friction** and **ultrasound** can all be used as **energy sources for heating**.

V groove or **butt joint**

single fillet **lap joint**



double fillet **T joint**

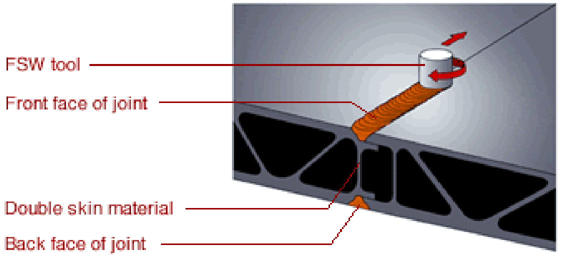


fillet welded **corner joint**

Examples of welded joints

Bolted joint Part 1

Airbus 380, lower fuselage panels (**laser welding**)



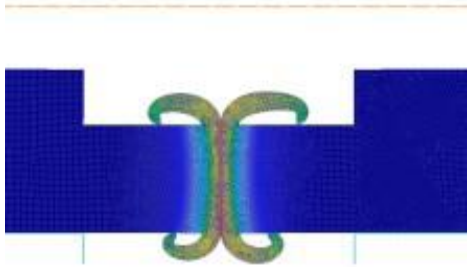
Hitachi High speed train (**Friction Stir Welding**)

The angel of the North

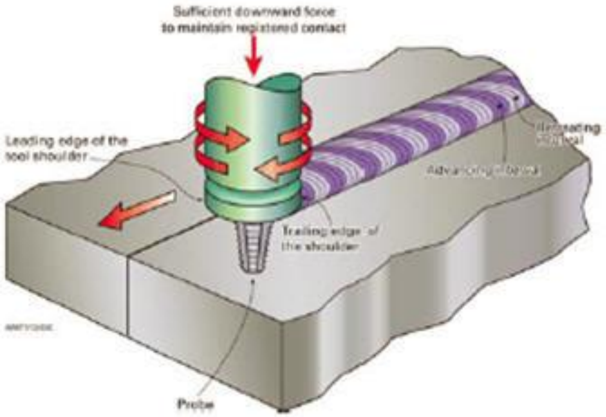
Offshore oil rig

Solid State Welding Processes

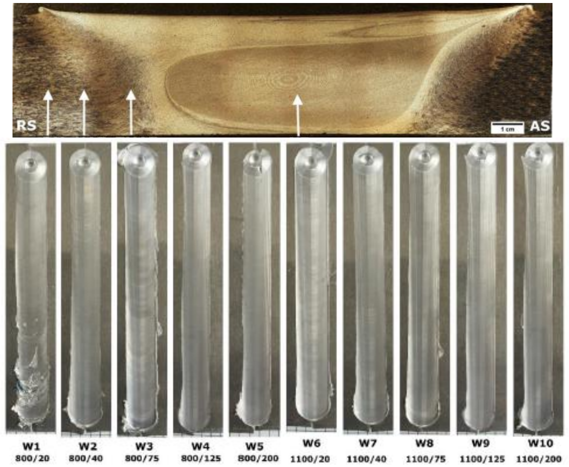
Bolted joint Part 1



Inertia friction welding for areoengine shaft

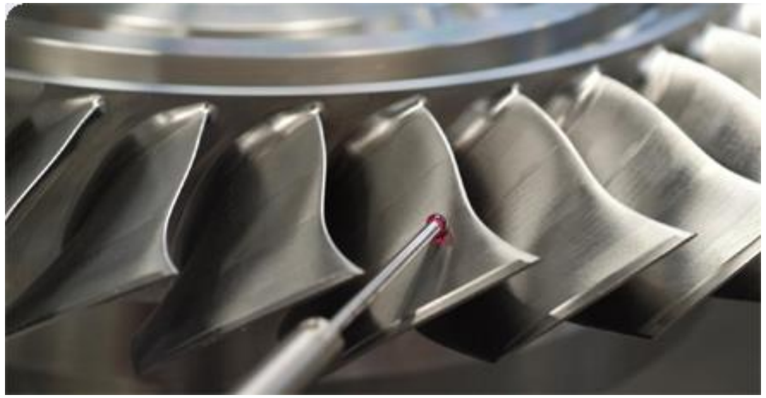


Friction stir welding (FSW), Invention by TWI
https://www.youtube.com/watch?v=y7rCTdxvGlg



O. Al-Jumaili, et al. FSW of Al6082, JMPT, vol.275,2020

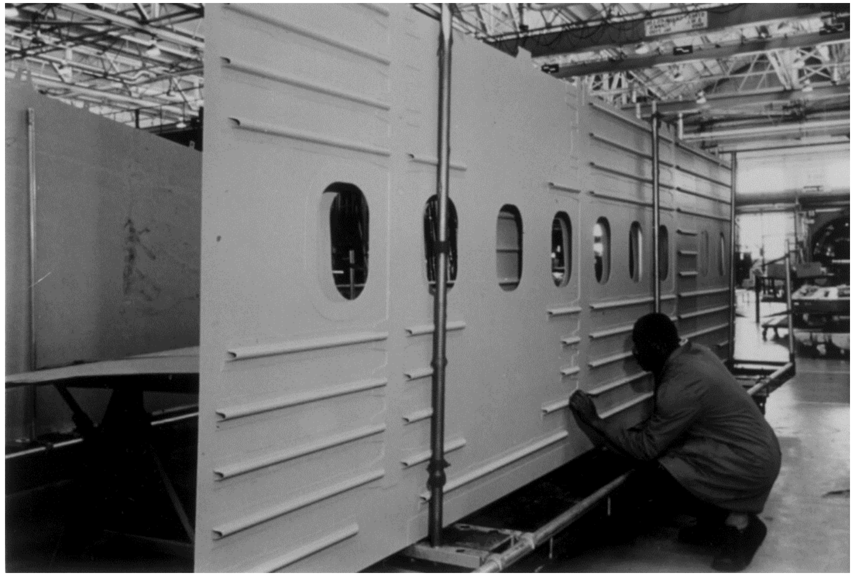
Inertia friction welding machine, MTC



Linear friction welding for manufacturing of blisk, MTU

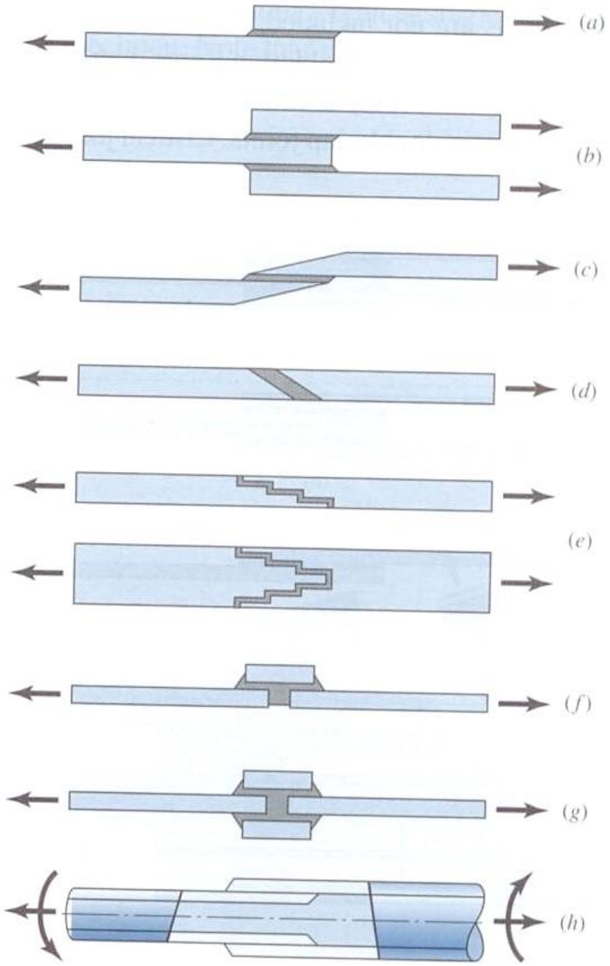
Bolted joint Part 1

Adhesive joints

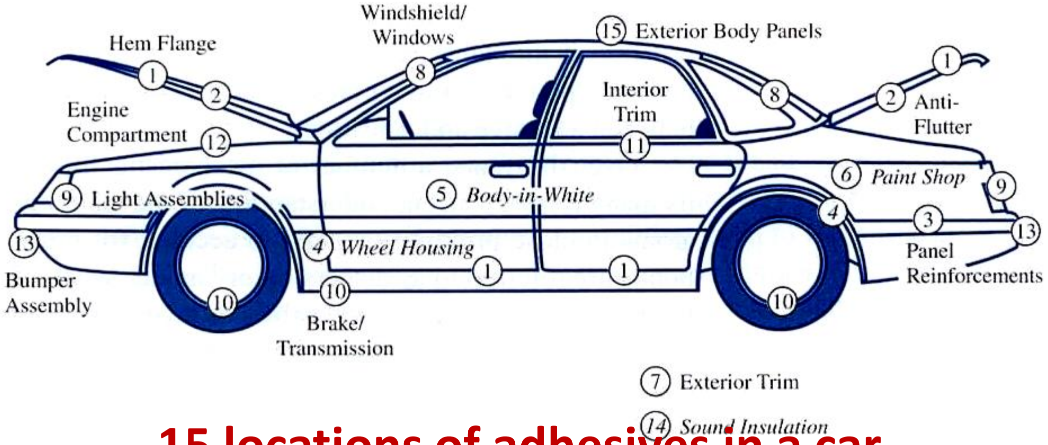


BAe RJ fuselage panel with stringers bonded to skin

Adhesive joint of engine component



Types of adhesive lap joints



15 locations of adhesives in a car

Quiz 1: **True or False** to each of the following statements **bolted joints**

A. **Bolted joint** may be used as both **"permanent"** and "**non-permanent**" joints.

B. **Bolted joint** can be used to **take both tensile and shear load**.

C. **Rivets** are good in taking **tensile but not shear load**.

D. Welding is commonly used to provide "**permanent**" joint.

Quiz 1: **True or False** to each of the following statements **bolted joints**

A. **Bolted joint** may be used as both **"permanent"** and "**non-permanent**" joints. **(true)**

B. **Bolted joint** can be used to **take both tensile and shear load**. **(true)**

C. **Rivets** are good in taking **tensile but not shear load**. **(false)**

D. Welding is commonly used to provide "**permanent**" joint. **(true)**

Design and analysis of bolted joints

End of Part 1

Design and analysis of bolted joints

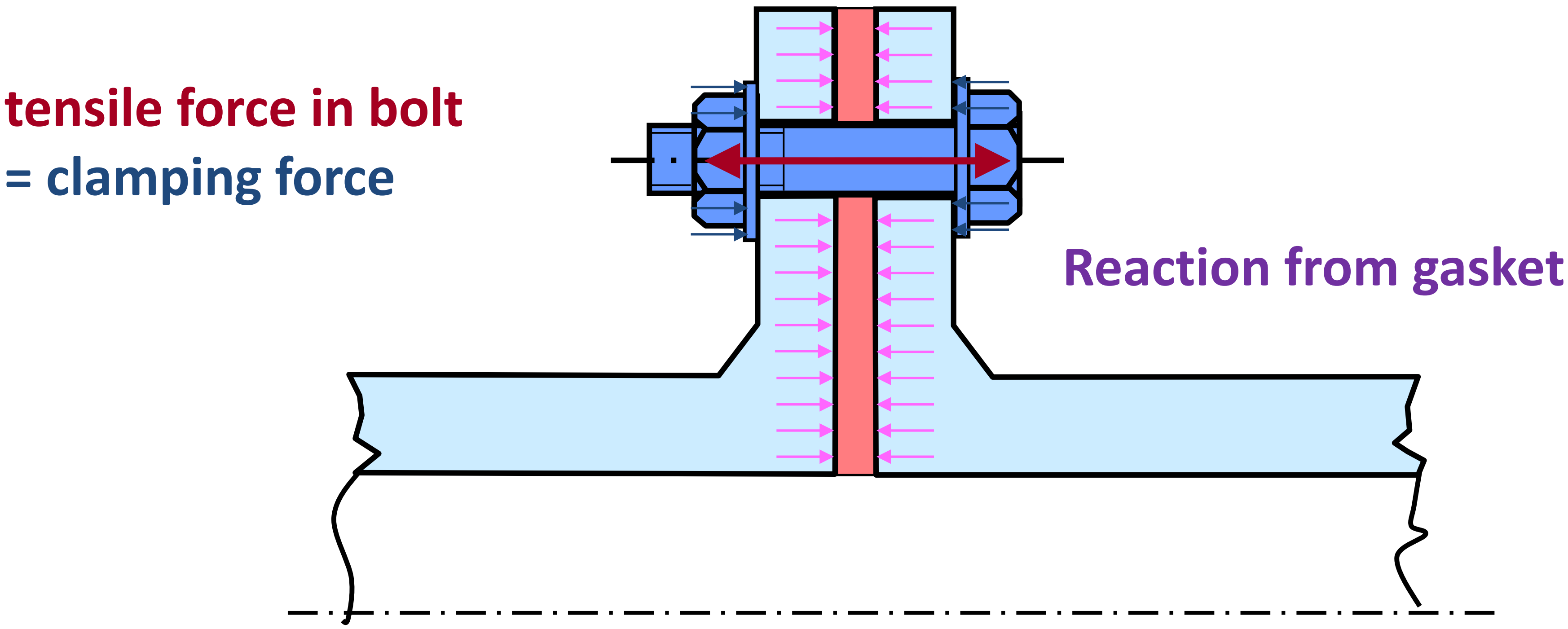
Part 2

Design Consideration of Bolted Joints

- Permanent or non-permanent joints
- Loads, stresses & strength
- Life
- Operational conditions
- Tooling & manufacturing efficiency

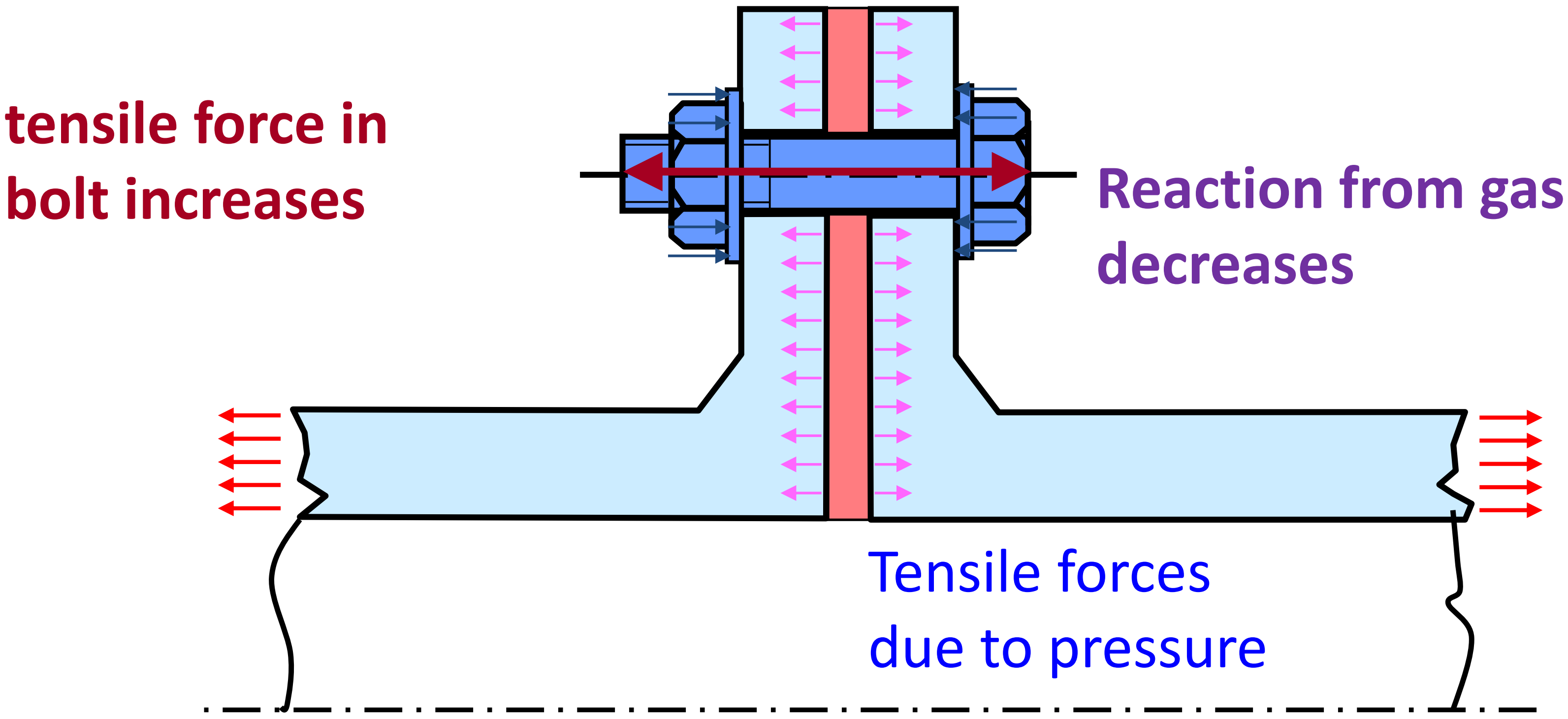
Loading of a bolted joint

- Initially, when the structure is unloaded, the tensile force in the bolt is equal to the compressive force in the clamped members.



Loading of a bolted joint

- When a load is applied to the joint (pressure, inertia, etc.), some of this load will stretch the bolt above its initial (pre-tensioned) length.



Pre-tensioned bolted joints

Bolts are pre-tensioned for two reasons

1. The bolt force must exceed the maximum force in the clamped members in service otherwise the joint faces will separate.

2. Pre-tensioning reduces the fluctuating stresses experienced by the bolt, thereby increasing the fatigue life.

Pre-tensioned bolted Joints

Bolted joint Part 2

- Recommended pre-load for **non-permanent** joints

$$F_i = 0.75 A_S \sigma_p$$

- Recommended pre-load for **permanent** joints

$$F_i = 0.9 A_S \sigma_p$$

where, A_s is the **tensile area** of the bolt,

σ_p is the **proof strength** of the bolt.

$$A_S = \frac{\pi}{16}\left(d_p + d_r\right)^2$$

$d_p = d - 0.6495p$ (pitch diameter)

$d_r = d - 1.0825p$ (minor diameter)

(***d*** is **nominal major diameter** and ***p*** is **pitch**)

If detailed information of the **proof strength**, $\boldsymbol{\sigma_p}$ is unavailable, an approximate value may be used

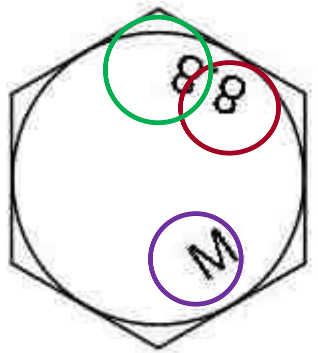
$$\boldsymbol{\sigma_p = 0.85\sigma_y}$$

$\boldsymbol{\sigma_y}$ is the **yield strength**

Grades & markings of bolts

- **BS 3692:2001 ISO metric hexagon bolts**, screws and nuts – Specification

Notes: ISO stands for International Organisation of Standards



M – ISO metric thread

1st **8** represents **100th** of the **tensile Strength**

2nd **8 or 0.8** indicates the **ratio** between the **yield strength** and the **tensile strength**

For example of an **8.8** bolt,

the **tensile strength**: $\sigma_{UTS} = 8 \times 100 = 800(MP_a)$

the **yield strength** $\sigma_y = 0.8 \times 800 = 640(MP_a)$

Therefore, $\sigma_p = 0.85\sigma_y = 0.85 \times 640 = 544(MP_a)$

Metric thread tensile area and mechanical properties

Bolted joint Part 2

- **BS EN ISO 898-1: 2013 Mechanical properties of fasteners made of carbon and alloy steels** (page 11)

Note: EN denotes European standards

Thread nominal dia, *d*

Tensile area, *As* (mm^2)

Thread property grade, (X.Y)

Proof load *Fp=As x σp* (N)

Table 5 — Proof loads — ISO metric coarse pitch thread

Thread [a] d	Nominal stress area $A_{s,nom}$ [b] mm^2	Property class								
		4.6	4.8	5.6	5.8	6.8	8.8	9.8	10.9	12.9/<u>12.9</u>
		Proof load, F_p ($A_{s,nom} \times S_{p,nom}$), N								
M3	5,03	1 130	1 560	1 410	1 910	2 210	2 920	3 270	4 180	4 880
M3,5	6,78	1 530	2 100	1 900	2 580	2 980	3 940	4 410	5 630	6 580
M4	8,78	1 980	2 720	2 460	3 340	3 860	5 100	5 710	7 290	8 520
M5	14,2	3 200	4 400	3 980	5 400	6 250	8 230	9 230	11 800	13 800
M6	20,1	4 520	6 230	5 630	7 640	8 840	11 600	13 100	16 700	19 500
M7	28,9	6 500	8 960	8 090	11 000	12 700	16 800	18 800	24 000	28 000
M8	36,6	8 240 [c]	11 400	10 200 [c]	13 900	16 100	21 200 [c]	23 800	30 400 [c]	35 500
M10	58	13 000 [c]	18 000	16 200 [c]	22 000	25 500	33 700 [c]	37 700	48 100 [c]	56 300
M12	84,3	19 000	26 100	23 600	32 000	37 100	48 900 [d]	54 800	70 000	81 800
M14	115	25 900	35 600	32 200	43 700	50 600	66 700 [d]	74 800	95 500	112 000
M16	157	35 300	48 700	44 000	59 700	69 100	91 000 [d]	102 000	130 000	152 000
M18	192	43 200	59 500	53 800	73 000	84 500	115 000	—	159 000	186 000
M20	245	55 100	76 000	68 600	93 100	108 000	147 000	—	203 000	238 000
M22	303	68 200	93 900	84 800	115 000	133 000	182 000	—	252 000	294 000
M24	353	79 400	109 000	98 800	134 000	155 000	212 000	—	293 000	342 000

Determining bolt torque to pre-tension

- **Recall torque equation for power screw**

$$T = \frac{F_i d_p\left(\mu\pi d_p + l\cos\alpha\right)}{2\left(\pi d_p \cos\alpha - \mu l\right)} + \mu_c F_i \frac{d_c}{2}$$

where d_p is pitch diameter, d_c is collar diameter, l is lead, *Fi* is pre-tension, μ and μ_c are friction coefficients.

rearrange and simplify the above equation

$$T = KF_i d$$

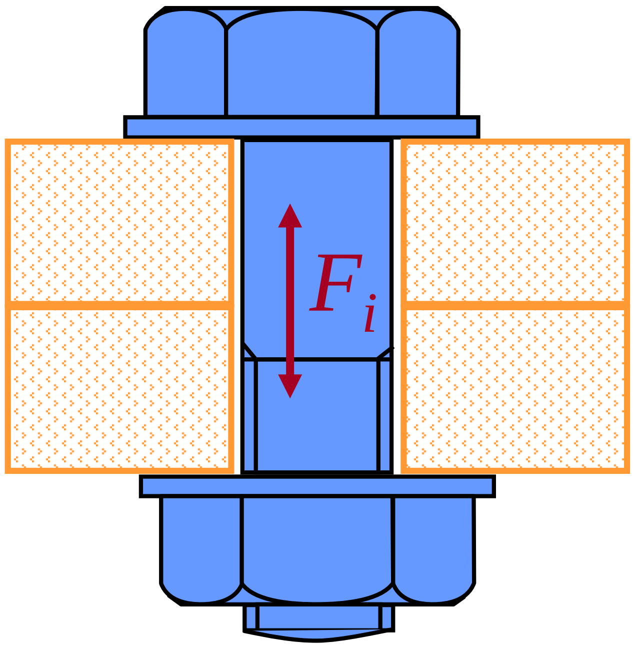
where *K* is torque coefficient dependent upon surface finish & lubrication, ***K* ≈ 0.2** for most cases; *d* is nominal diameter and *Fi* is pre-tension

Modelling of bolted joint

F_i = preload on bolt due to tightening

F_b = resultant load on bolt in **tension**

F_c = resultant load on components in **compression**



$$F_i = F_b = F_c$$

Modelling of bolted joint

F_i = preload on bolt due to tightening

P = external tensile load:

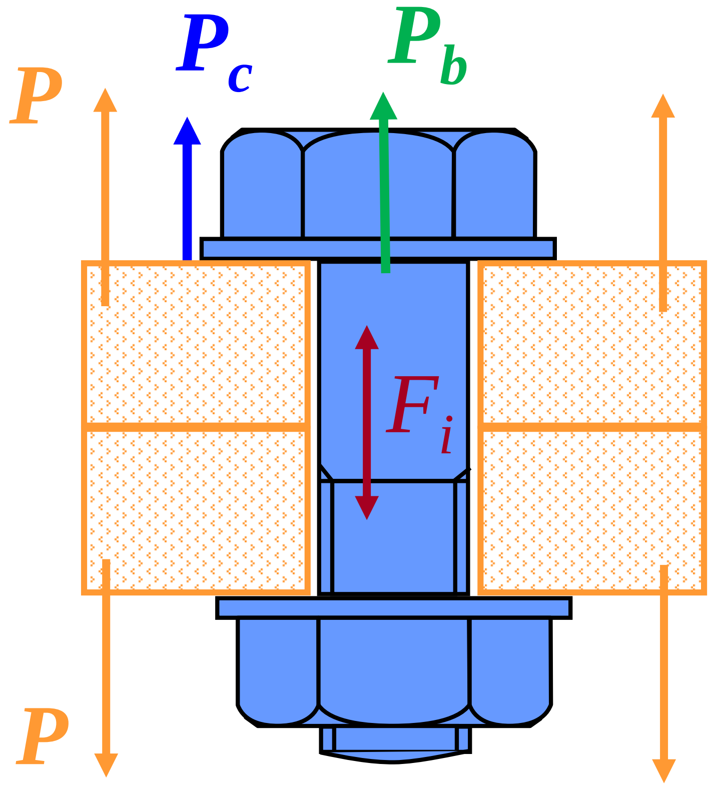
$$P = P_c + P_b$$

P_b = portion of P taken by bolt

P_c = portion of P taken by components

F_b = resultant load on bolt

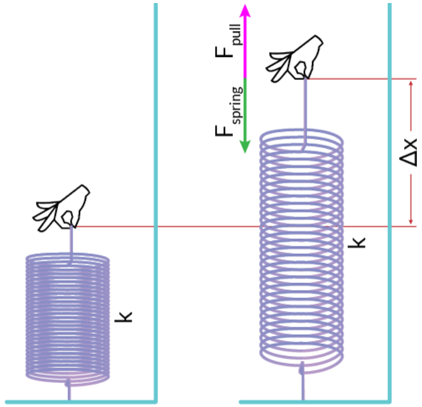
F_c = resultant load on components



$$F_b = P_b + F_i > 0$$

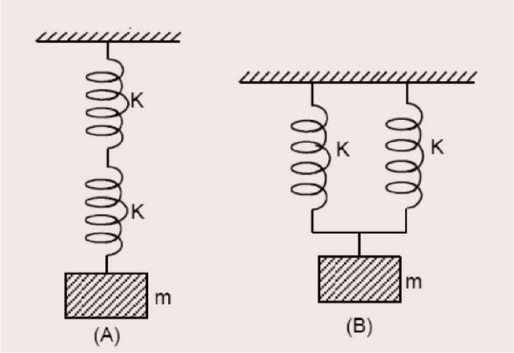
$$F_c = P_c - F_i \leq 0$$

Modelling of bolted joint



Hooke's law of a spring

$$F = k \times \Delta x$$



Springs in parallel & series

- Effectively, the bolted joint can be modelled as **two springs in parallel** with the components in compression & the bolt in tension.

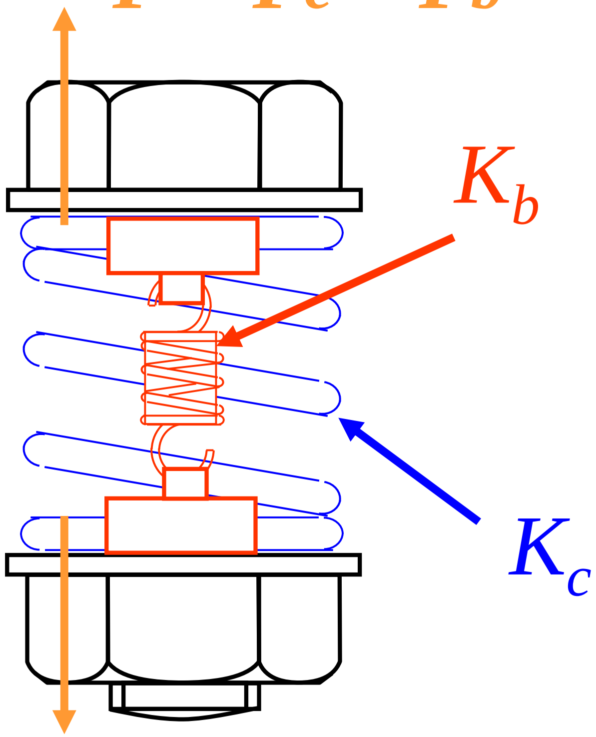
Change in bolt length $\delta_b = \dfrac{P_b}{K_b}$

=

Change in components length $\delta_c = \dfrac{P_c}{K_c}$

$$\delta_b = \delta_c$$

$$P = P_c + P_b$$



Modelling of bolted joint

as $\delta_b = \frac{P_b}{K_b} = \delta_c = \frac{P_c}{K_c}$, then $\frac{P_b}{K_b} = \frac{P_c}{K_c}$

but total external load is:

$$\mathrm{P} = P_b + P_c = P_b + P_b\left(\frac{K_c}{K_b}\right) = \left(1 + \frac{K_c}{K_b}\right)P_b$$

rearranging:

$$P_b = \frac{K_b}{K_b + K_c}P; \qquad P_c = \frac{K_c}{K_b + K_c}P$$

Modelling of bolted joint

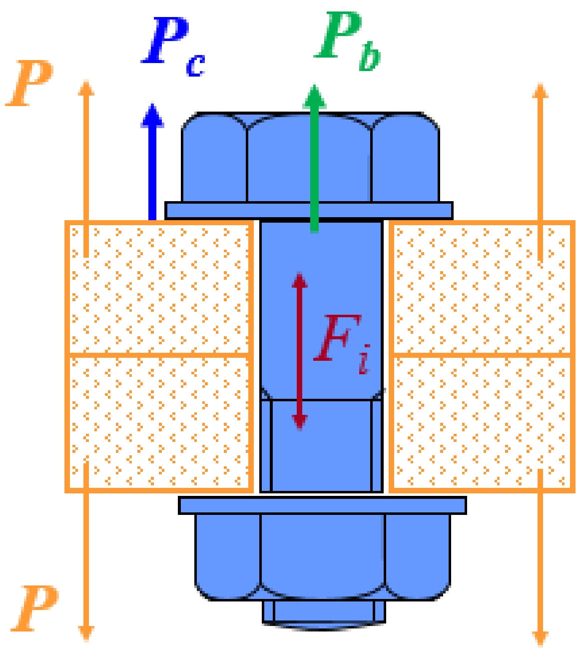
$$P_b = \frac{K_b}{K_b + K_c} P; \qquad P_c = \frac{K_c}{K_b + K_c} P$$

Resultant load on the bolt **in tension** is:

$$F_b = P_b + F_i = \frac{K_b}{K_b + K_c} P + F_i > 0$$

Similarly, for the component **in compression**:

$$F_c = P_c - F_i = \frac{K_c}{K_b + K_c} P - F_i \leq 0$$



$$F_b = P_b + F_i > 0$$

$$F_c = P_c - F_i \leq 0$$

Calculating K_b

- K_b is easy enough to calculate as $K_b = P/\delta$ if the bolt is threaded along its **length of grip**:

$$K_b = \frac{A_S E}{l_t}$$

where, A_s is tensile area, E is the Young's Modulus and l_t is the length

$$A_S = \frac{\pi}{16}\left(d_p + d_r\right)^2$$

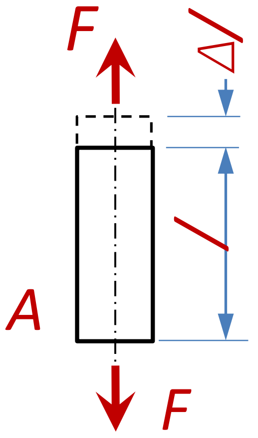
$$d_p = d - 0.6495p$$
$$d_r = d - 1.0825p$$

(for metric threads)

where d_p is pitch diameter, d_r is minor diameter, d is nominal major diameter and p is pitch, respectively.

Alternatively, A_s may be found from **Table 5, BS EN ISO 898-1: 2013**

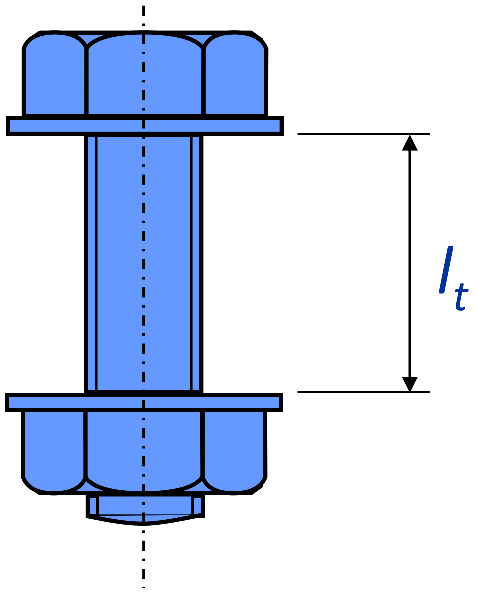
Stress-strain relation



$$\sigma = E\varepsilon$$

$$\frac{F}{A} = E\frac{\Delta l}{l}$$

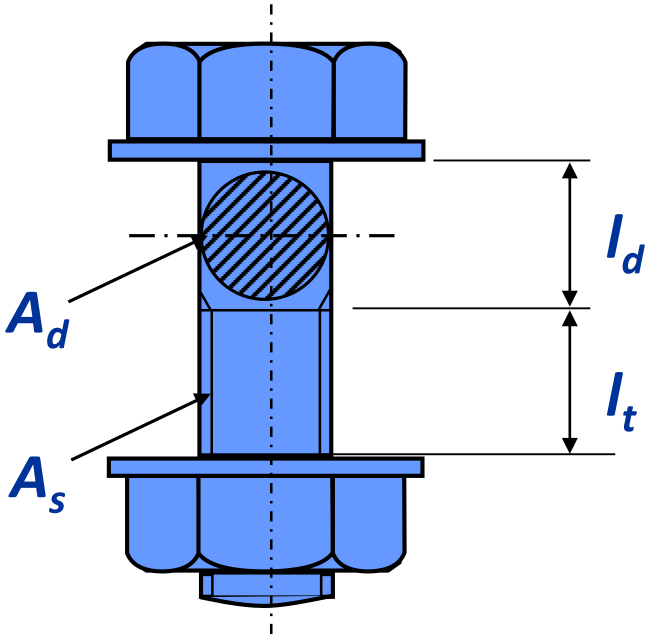
$$F = \frac{AE}{l}\Delta l$$



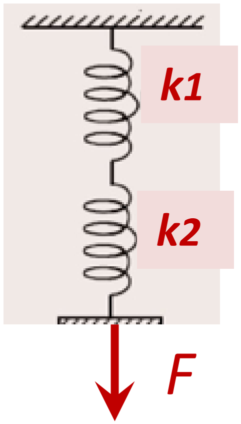
Calculating K_b

- If there is an unthreaded portion of cross-sectional area **A_d** and length **l_d** (including necked bolts) then use formula for **springs in series** to give:

$$K_b = \frac{A_d A_S E}{A_d l_t + A_S l_d}$$



- **How to derive the above formula?**



The derivation of the above equation is based on the concept of **Equivalent Stiffness of springs in series**

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} + \ \ldots$$

which may be derived by using the following relations

$$F = F_1 = F_2$$

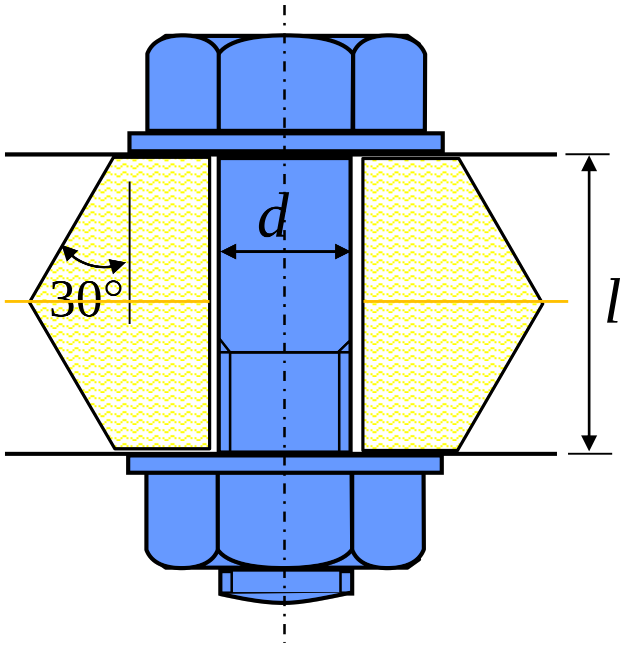
$$\Delta l = \Delta l_1 + \Delta l_2$$

$$F = k\,\Delta l \qquad F_1 = \frac{A_1 E}{l_1}\Delta l_1 \qquad F_2 = \frac{A_2 E}{l_2}\Delta l_2$$

Calculating K_c

- **K_c is harder**: requires us to consider hollow cone-shaped regions in compression
- It can be shown **(no need to know how to derive)**:

$$K_c = \frac{0.5774\pi E d}{2ln\left(5\,\frac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)}$$



(detailed derivations are given in Shigley et al., Mechanical Engineering Design, Ch8, TJ230 SHI)

Concept of "hard" joint

- A "good" joint would have **stiff components** and **elastic bolts**
- In a well engineered **'hard' joint**, the components are stiff $K_c >> K_b$:
  - K_c might be 1.7 GN/m or 2.2 GN/m
  - whereas K_b would be 250 MN/m or 500 MN/m
  - i.e. **$K_c > 3 K_b$**
- In **'soft' joint,** the bolt is stiffer than the components which results in:
  - High level of bolt fatigue loads

Recall,

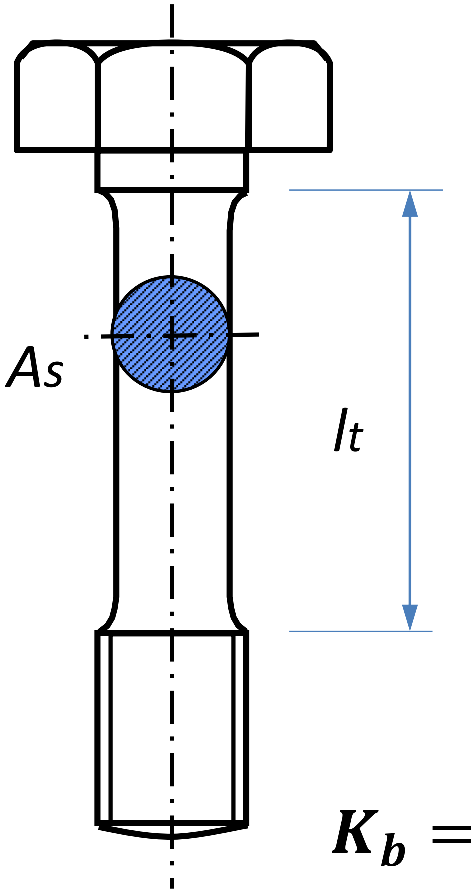
$$F_b = \frac{K_b}{K_b + K_c} P + F_i > 0$$

$$F_c = \frac{K_c}{K_b + K_c} P - F_i \leq 0$$

Reducing bolt stiffness: necked bolts

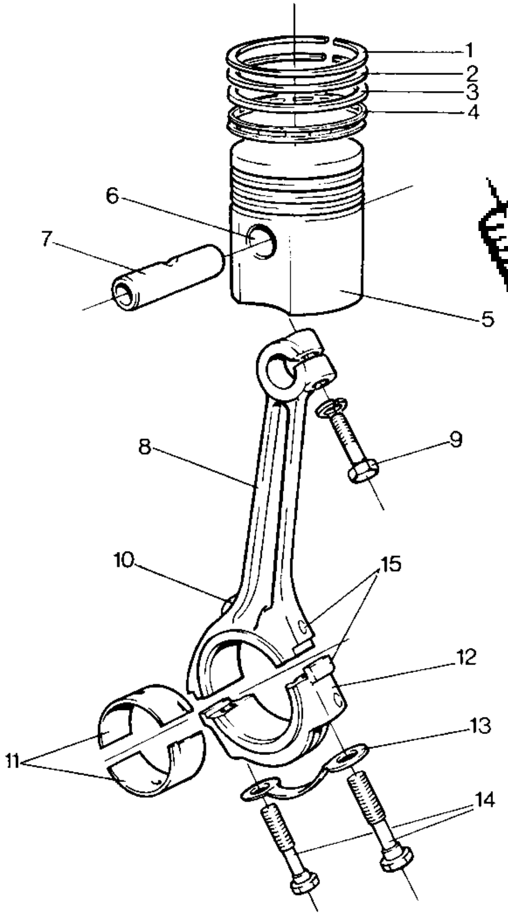
- For a bolt (*E* is dependent on material) we require
  (1) A **small** cross-sectional **area**
  (2) A **long length**

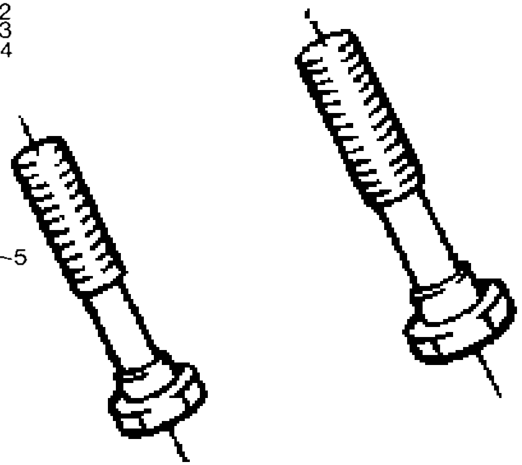
- A small cross-sectional area
  – Necked down to root diameter
  – Reduces stiffness without loss of strength



$$K_b = \frac{A_S E}{l_t}$$

Reducing bolt stiffness: necked bolts





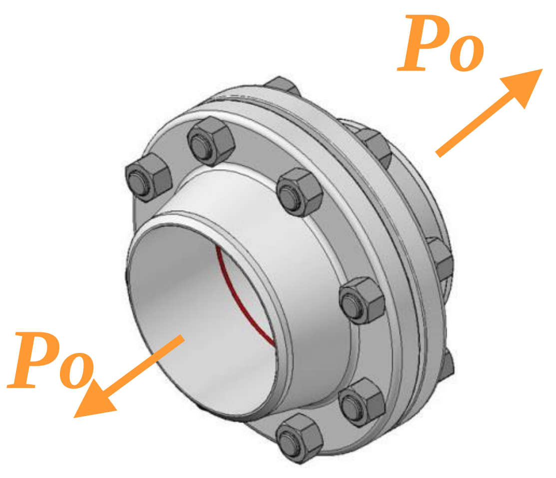
Necked bolts in connecting rod

Tie rods used in pneumatic cylinders

Reducing bolt stiffness: long bolts or tie rods to make a "hard" joint

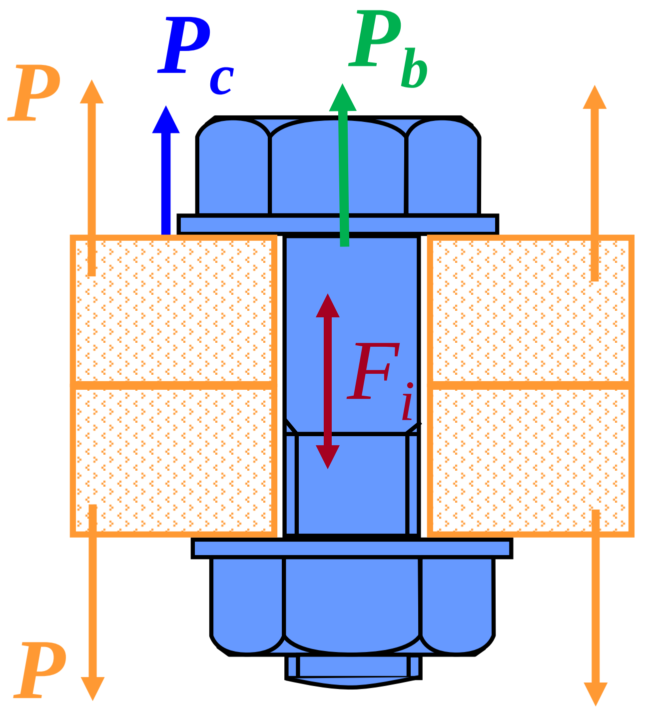
Joint failure & reserve factor

- In application, normally **multiple (N) bolts** need to be used



$$\frac{K_c}{K_b + K_c} P_0 - NF_i = 0$$

$$P_0 = NF_i \frac{K_b + K_c}{K_c}$$



Therefore, **reserve factor** $n_0 = \frac{P_0}{P} \geq 1.5 \sim 2$

$$F_b = \frac{K_b}{K_b + K_c} P + F_i > 0$$

$$\boldsymbol{F_c = \frac{K_c}{K_b + K_c} P - F_i \leq 0}$$

P_0 is the **maximum allowable external load** applied to **N** bolts at ***F_i*** preload,

P is the **actual external load** applied to the bolted joint

A general guide for reserve factor

Recommended reserve factor (n_0)	Operational and environmental conditions as well as use of materials	Uncertainty of material and working conditions
1.25 ~ 1.5	Reliable materials under controlled conditions subjected to loads and stresses known with certainty	
1.5 ~ 2	Well-known materials under reasonably constant environmental conditions subjected to known loads and stresses	
2 ~ 2.5	Average materials subjected to known loads and stresses	
2.5 ~3.0	Less well-known materials under average conditions of load, stress, and environment	
3 ~ 3.4	Untried materials under average conditions of load, stress, and environment or well-known materials under uncertain conditions of load, stress, and environment	

Design of pre-tensioned bolted joints

Bolted joint Part 2

1. **Consider** to use a non-permanent or permanent joint, **define** external load (***P***) and reserve factor (***no***), decide the number of bolts (***N***) **to ensure no joint separation**

2. **Estimate** preload (F^{est}_i) of bolt by **assuming** a **hard joint**, e.g. ***Kc ≈3Kb***, $NF_i \geq \frac{K_c}{K_b+K_c}P$

3. **Choose a suitable bolt size, e.g. M4, M6 or M10, … and grade, e.g. 6.8, 8.8** or **10.9** and **determine** the preload (***Fi***) by calculating or using **Table 5** of **BS EN ISO 898-1: 2009**

   $F_i = 0.75A_s\sigma_p$ (non-permanent) or $F_i = 0.9A_s\sigma_p$ (permanent joint) $\geq F^{est}_i$

4. **Calculate** the **stiffness** of the **bolts & components** (***Kb*** & ***Kc***):

$$K_b = \frac{A_d A_S E}{A_d l_t + A_S l_d} \quad \text{or} \quad K_b = \frac{A_S E}{l_t} \quad \text{and} \quad K_c = \frac{0.5774\pi E d}{2ln\left(5\frac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)}$$

6. **Calculate** the maximum allowable external load (***Po***):

$$P_0 = NF_i\frac{K_b + K_c}{K_c}$$

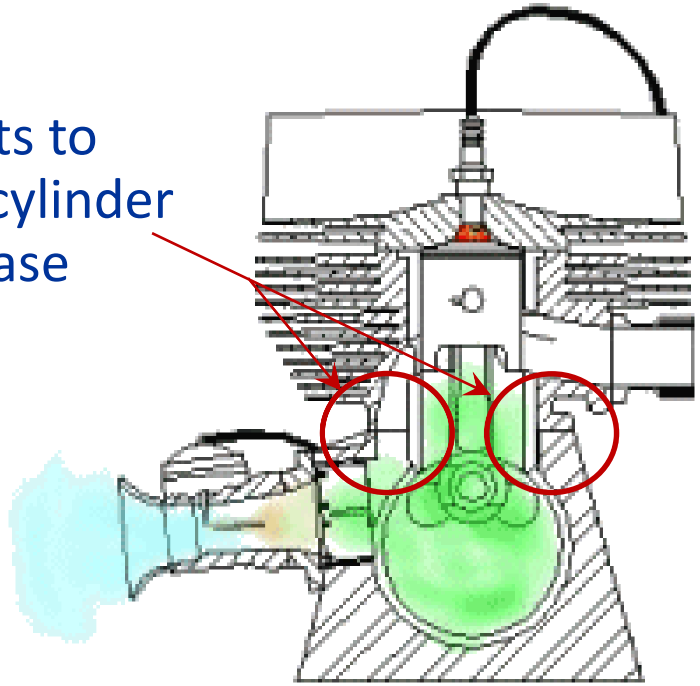
7. **Calculate** the **reserve factor**, ***no***$= \frac{P_0}{P} \geq 1.5{\sim}2$, **if NOT, go back to Step 3 & iterate**

Bolted joint under cyclic loading

Bolted joint Part 2

- Advantages of joints under cyclic loading, e.g. the bolted joint of the cylinder head & crank case of a **2-stroke engine**:

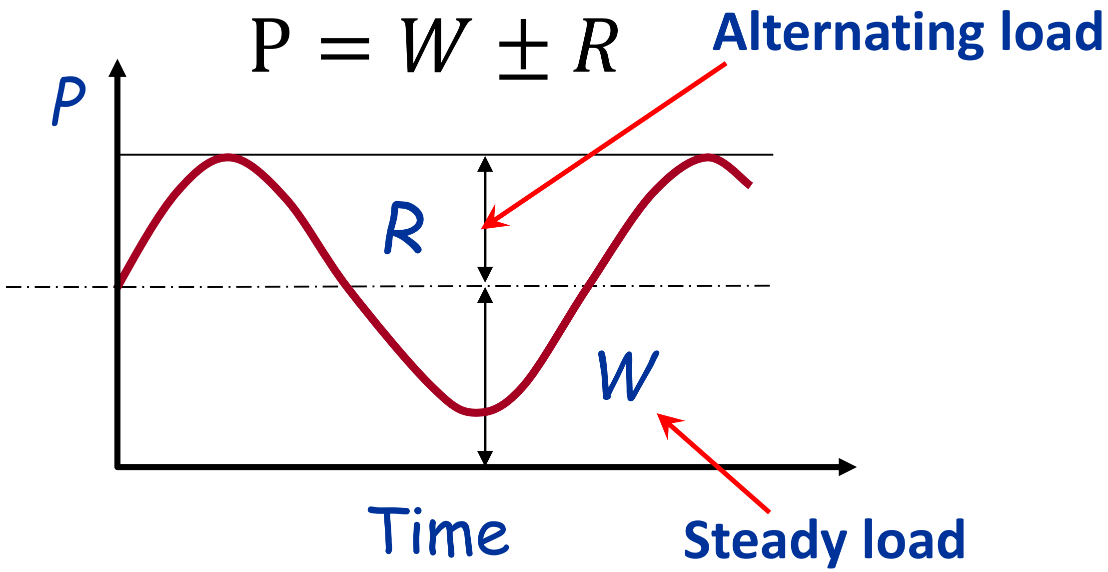
Bolted joints to clamp the cylinder and crankcase



Example of a small two-stroke engine

https://en.wikipedia.org/wiki/Two-stroke_engine

$$P = W \pm R$$



$$F_c = \frac{K_c}{K_b + K_c} P - F_i \leq 0$$

- For cylinder and crankcase faces to remain in contact:

$$F_i \geq (W + R) \frac{K_c}{K_b + K_c}$$

Bolted joint under cyclic loading

Bolted joint Part 2

Resultant force in bolt:

$$F_b = (W \pm R)\frac{K_b}{K_b + K_c} + (W + R)\frac{K_c}{K_b + K_c}$$

$$= \left(W + \frac{K_c}{K_b + K_c}R\right) \pm \frac{K_b}{K_b + K_c}R$$

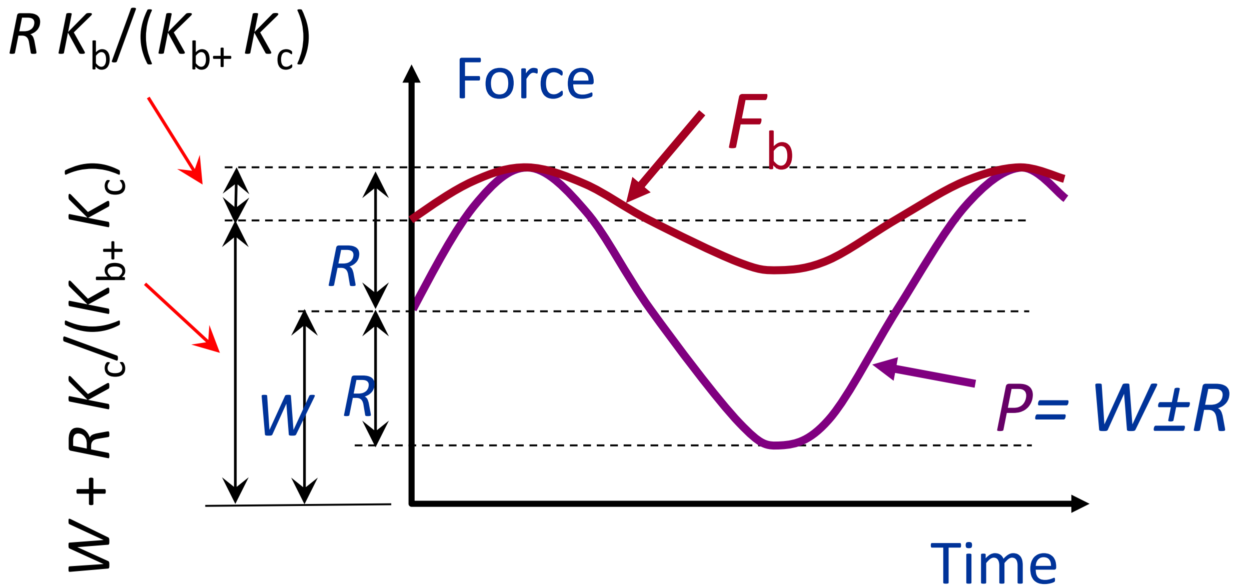
Steady load increased

Fluctuating load reduced

We know, $P = W \pm R$

$$F_i \geq (W + R)\frac{K_c}{K_b + K_c}$$

$$F_b = \frac{K_b}{K_b + K_c}P + F_i > 0$$



Summary

- Be familiar with various joining techniques and different types of joints;
- Be able to calculate pre-load for **non-permanent** and **permanent bolted joints** based on **BS/ISO bolt strength grading**, e.g. M10 8.8;
- Be able to determine the **stiffness** of the bolt & clamped members, **resultant loads**;
- Understand the behaviour of pre-tensioned joint under **cyclic loading**;
- Be able to use **reserve factor formula** and the **suggested steps** to design a pre-tensioned bolt joint.

Revision questions

- How to determine the recommended pre-tension load (***Fi***)?
- How much tightening torque (***T***) is needed to achieve the recommended pre-tension load (***Fi***)?
- By picking up any metric bolts/screws to BS 3692 or BS/EN/ISO 898, how do we know their **mechanical properties**, e.g. $\boldsymbol{\sigma_{UTS}}$ or $\boldsymbol{\sigma_p}$?
- How much external load (***P***) will be taken by the bolt (***Pb***) and the components (***Pc***)?
- In designing a bolted joint, why is a **"hard"** joint a preferred option?
- Why is a pre-tensioned bolted joint is beneficial for **cyclic loading**?
- What is the **threshold or critical requirement** used in the design of a bolted joint?

Bolted joint resources

- http://www.tribology-abc.com
  - Great calculators and other general information on threads (+ bearings)
- Childs, R.N., 2004. Mechanical Design, Elsevier
  - Chapter 12 discusses screw threads
- Shigley, J.E., Mischke, C.R., Budynas, R.G., 2003. Mechanical Engineering Design, 7th edition, McGraw-Hill, (TJ230 SHI)
  - Chapter 8 covers threads & joints

Design and analysis of bolted joints

End of Part 2

Design and analysis of bolted joints

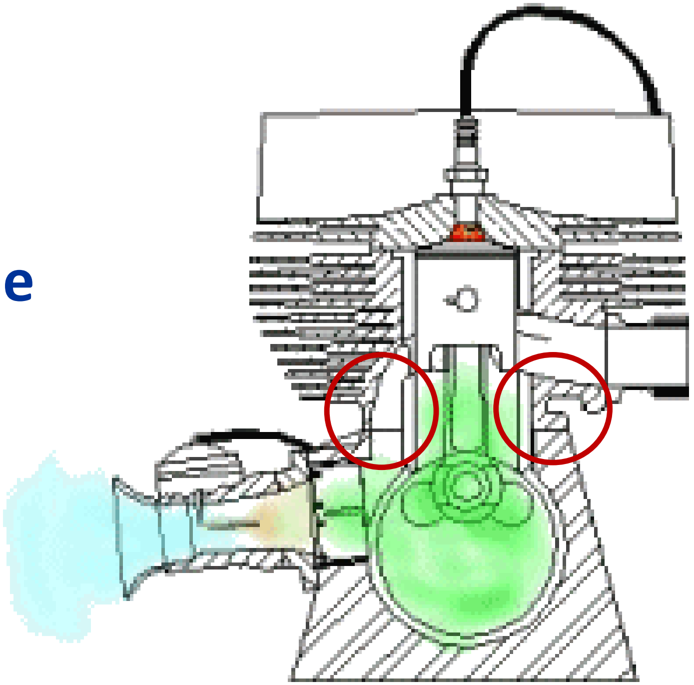
Part 3 Worked examples

Worked example 1: Joint design of a 2-stroke engine

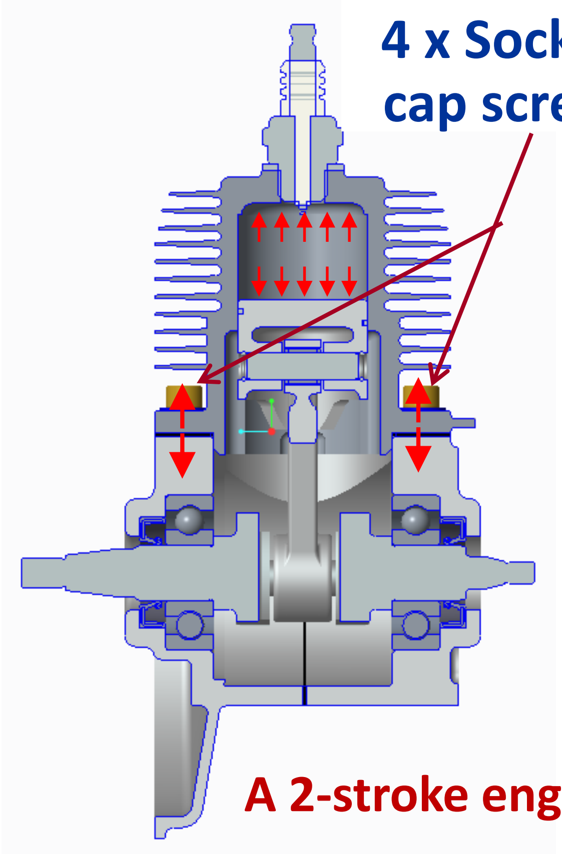
Bolted joint Part 3

- **Design bolted joint of a 2.4 kW** 2-stroke engine.
  - Peak force is ***P* = 6.5 kN**
  - A **permanent joint** with a threaded grip length ***l_t* = 25 mm**
  - Cylinder and crankcase are made of cast Al (**E=70 GPa**)
  - **4 x bolts (5.6 or similar,** carbon steel, **E=200 GPa)**
  - **Reserve factor** should be in the range of **1.5~2.0**.

- **Determine**
  - **a) Suitable size of socket cap screw**
  - **b) Right amount of tightening torque**



Otto cycle of two-stroke engine
https://en.wikipedia.org/wiki/Two-stroke_engine



4 x Socket cap screw

A 2-stroke engine

Worked example 1: Joint selection of a 2-stroke engine

Bolted joint Part 3

a) Selection of a suitable socket cap screw

1. **Joint design** specifies external load (***P=6.5 kN***) and reserve factor (***n_o=1.5~2***), a **permanent joint** of **4 x** socket cap screw (***N=4***).

2. **Estimate** pre-load (***Fi***) of bolt by assuming a hard joint, ***Kc ≈3Kb,*** $NF_i \geq \frac{K_c}{K_b+K_c}P$

$$F_i^{est} \geq \frac{1}{N}\frac{K_c}{K_b+K_c}P = \frac{1}{4} \times \frac{1}{1.3} \times 6,500 = 1,250(N)$$

3. **Choose** a suitable bolt size from **BS ISO 898-1: 2009 (Table 5)**

M4 seems to be the right size

$$F_i = 0.9 \times A_S\sigma_P = 0.9 \times F_P$$
$$= 0.9 \times 2,460 = 2,214(N)$$

$$F_i = 2,214 > F_i^{est} = 1,250\ (N)$$

Ok for detailed evaluation

Table 5 — Proof loads — ISO metric coarse pitch thread

Thread [a] d	Nominal stress area $A_{s,nom}$ [b] mm²	Property class 4.6	4.8	5.6	5.8	6.8	8.8	9.8
		Proof load, F_p ($A_{s,nom} \times S_{p,nom}$), N						
M3	5,03	1 130	1 560	1 410	1 910	2 210	2 920	3 270
M3,5	6,78	1 530	2 100	1 900	2 580	2 980	3 940	4 410
M4	8,78	1 980	2 720	2 460	3 340	3 860	5 100	5 710
M5	14,2	3 200	4 400	3 980	5 400	6 250	8 230	9 230
M6	20,1	4 520	6 230	5 630	7 640	8 840	11 600	13 100
M7	28,9	6 500	8 960	8 090	11 000	12 700	16 800	18 800

Worked example 1: Joint selection of a 2-stroke engine

Bolted joint Part 3

4. **Calculate** stiffness of the **bolts & components** (***Kb*** & ***Kc***):

$$K_b = \frac{A_S E}{l_t} = \frac{8.78 \times 200 \times 10^3}{25} = \mathbf{70.2 \times 10^3} \left({}^{N}\!/_{mm}\right)$$

$$K_c = \frac{0.5774\pi E d}{2ln\left(5\frac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)} = \frac{0.5774 \times 3.1416 \times 70 \times 10^3 \times 4}{2 \times ln\left(5 \times \frac{0.5774 \times 25 + 0.5 \times 4}{0.5774 \times 25 + 2.5 \times 4}\right)}$$

$$= \mathbf{211.3 \times 10^3} \left({}^{N}\!/_{mm}\right)$$

5. **Calculate** the maximum allowable external load (***Po***):

$$P_0 = NF_i \frac{K_b + K_c}{K_c} = 4 \times 2{,}214 \times \frac{(70.2 + 211.3) \times 10^3}{211.3 \times 10^3} = \mathbf{11{,}790}(N)$$

6. **Calculate** the reserve factor (***no***):

$$n_0 = \frac{P_0}{P} = \frac{11{,}790}{6{,}500} = \mathbf{1.8}$$

which is in the range of 1.5~2.0. Therefore, 5.6 M4 socket cap screw is a suitable choice.

Worked example 1: Joint selection of a 2-stroke engine

Bolted joint Part 3

b) Calculation of the required torque for tightening

Use simplified torque tightening equation

$$T = KF_i d$$

where ***K*** is torque coefficient, ***K* ≈ 0.2** for most cases;
d is nominal diameter and *Fi* is pre-tension

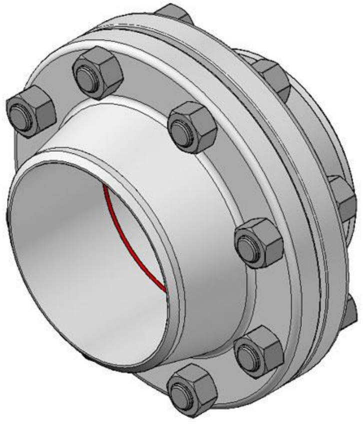
$$T = KF_i d = 0.2 \times 2{,}214 \times 4 \times 10^{-3} = \mathbf{1.8}\ (Nm)$$

Therefore, application of ***T=1.8 Nm* tightening torque** to each **M4** socket cap screw would generate the required ***Fi=2,214 N*** pre-tension load.

Worked example 2: Evaluation of a bolted joint

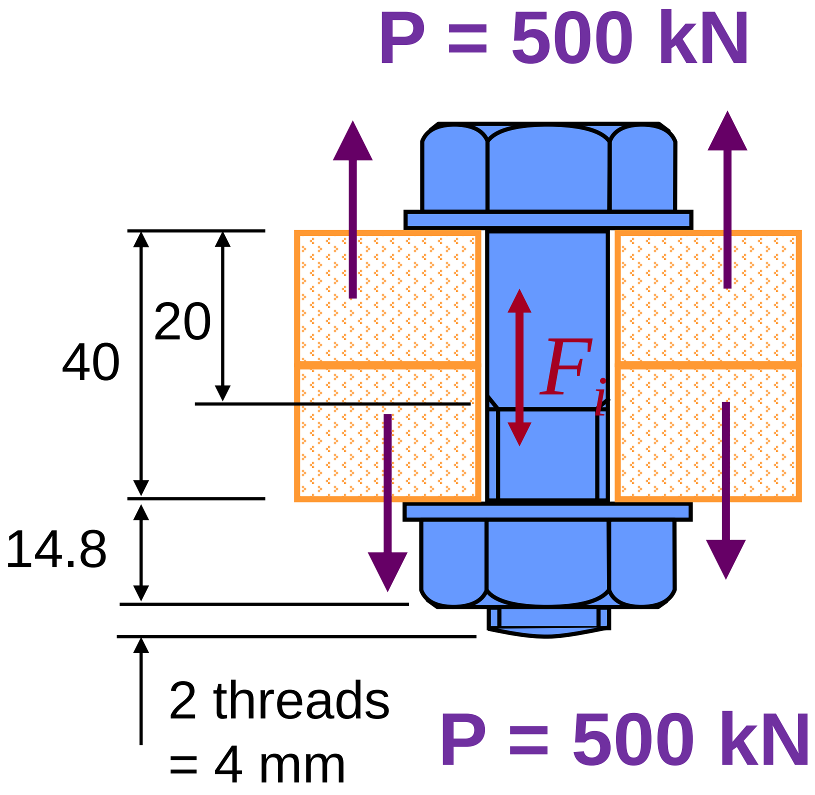
Bolted joint Part 3

Evaluation of a designed bolt joint



- **Eight non-permanent steel M16, 8.8** bolts (E = 200 GPa) are used to secure 40 mm of non-gasketed cast iron flanged coupling (E = 96 GPa) on which is imposed a separating force of **P=500 kN.**

- **Determine:**
  a) bolt stiffness K_b
  b) components stiffness K_c
  c) Reserve factor n_0



Bolt stiffness

- **Bolt stiffness:**

$$K_b = \frac{A_d A_S E}{A_d l_t + A_S l_d}$$

- Area of unthreaded section

$$A_d = \frac{\pi}{4} d^2 = \frac{\pi \times (16 \times 10^{-3})^2}{4} = 201.1 \times 10^{-6}\ m^2$$

- Area of threaded section

$$A_S = \frac{\pi}{16}\left(d_p + d_r\right)^2 = \frac{\pi}{16}(14.7 + 13.55)^2 \times 10^{-6}$$
$$= 156.7 \times 10^{-6} m^2$$

$$K_b = \frac{201.1 \times 156.7 \times 10^{-12} \times 200 \times 10^9}{(201.1 \times 20 + 156.7 \times 20) \times 10^{-9}} = 881.6 \times 10^6\ N/m$$

- **Component stiffness:**

$$K_c = \frac{0.5774\pi E d}{2ln\left(5\,\frac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)} = 1{,}513.4 \times 10^6\ N/m$$

Initial clamping force

- The recommended initial clamping force for **M16 8.8 non-permanent joints** is:

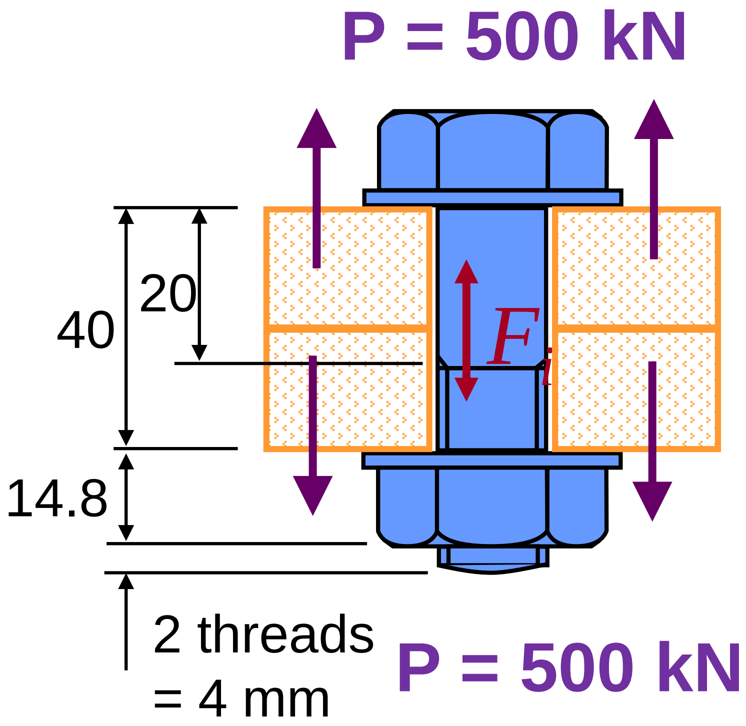
$$F_i = 0.75 A_S \sigma_p$$

$$\sigma_p = 0.85\sigma_y = 0.85 \times 640 = 544(MP_a)$$

$$A_S = \frac{\pi}{16}\left(d_p + d_r\right)^2 = 156.7 \times 10^{-6} m^2$$

Therefore,

$$F_i = 0.75 \times 544 \times 156.7 = 63.9 \times 10^3\ N$$



Reserve factor

- Maximum external load to separation for **8 x M16 bolts**:

$$P_0 = NF_i \frac{K_b + K_c}{K_c}$$

$$= 8 \times 63.9 \times 10^3 \frac{(881.6 + 1{,}513.4) \times 10^6}{1{,}513.4 \times 10^6}$$

$$= 809.2 \times 10^3\ N$$

- Therefore, the **reserve factor** of this bolted joint is:

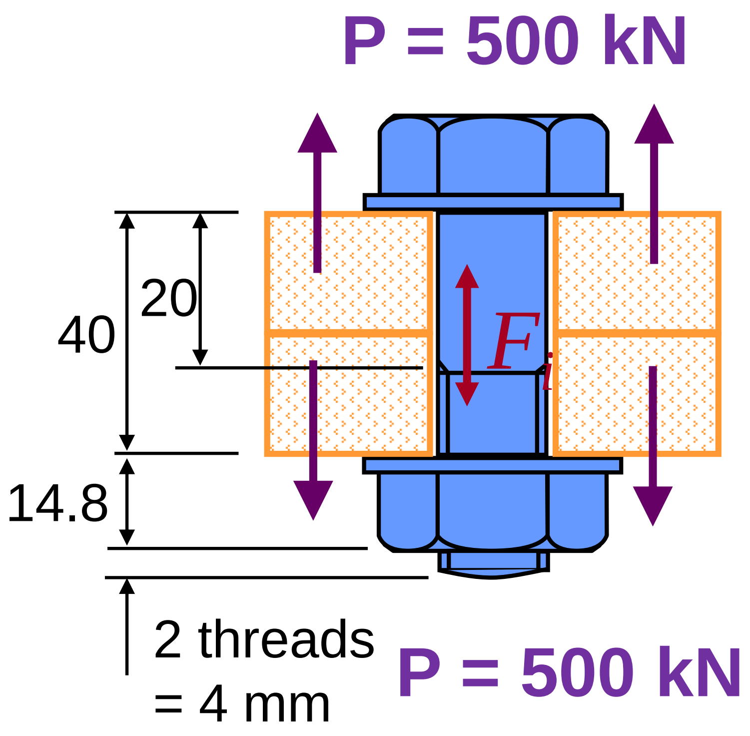
$$n_0 = \frac{P_0}{P} = \frac{809.2 \times 10^3}{500 \times 10^3} = 1.62$$

This is acceptable.

Recall,

$$\frac{K_c}{K_b + K_c} P_0 - NF_i = 0$$

$$P_0 = NF_i \frac{K_b + K_c}{K_c}$$



Answers to revision questions

Bolted joint Part 2

- The recommended **pre-tension load** (***Fi***): ***(slide 24)***

$$F_i = 0.75A_s\sigma_p \text{ (non-permanent) or } F_i = 0.9A_s\sigma_p \text{ (permanent joint)}$$

- **Tightening torque** (***T***): ***(slide 27)*** $T = KF_id$

- **Mechanical properties** of metric bolts are defined by **mark**s, e.g. **10.9** ***(slide 24)***

- External load (***P***) will be shared by the bolt (***Pb***) and components (***Pc***) ***(slides 29-32)***

$$F_b = \frac{K_b}{K_b + K_c}P + F_i > 0 \qquad F_c = \frac{K_c}{K_b + K_c}P - F_i \le 0$$

- A **"hard"** joint is preferred because components take a **larger load** ***(slides 36-38)***

- Pre-tensioned bolted joint reduces **cyclic loading in bolts** ***(slides 42-43)***

- **The requirement** is to ensure **no joint separation** ***(slide 40)***

$$P_0 = NF_i\frac{K_b + K_c}{K_c}$$

$$n_0 = \frac{P_0}{P} \ge 1.5\sim 2$$

Design and analysis of bolted joints

End of Session