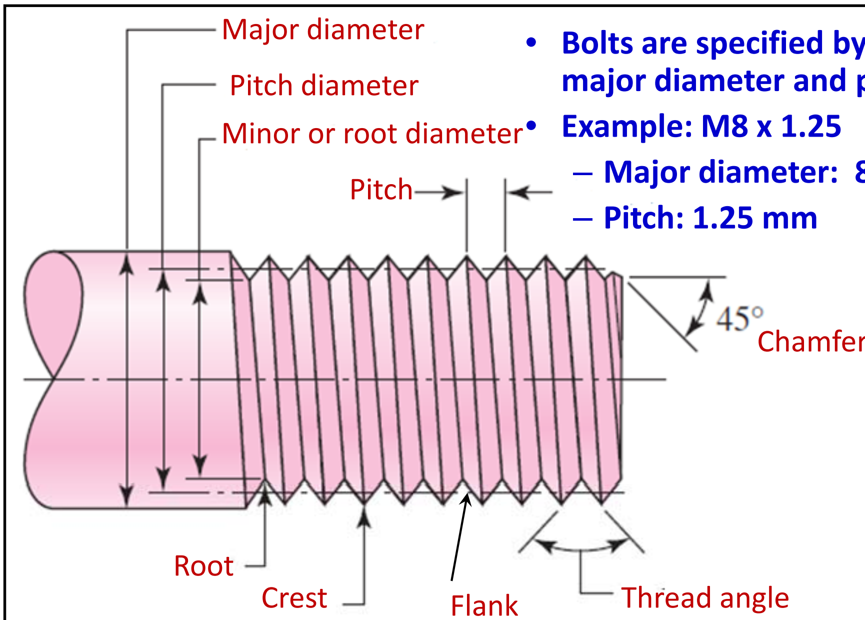
Fasteners

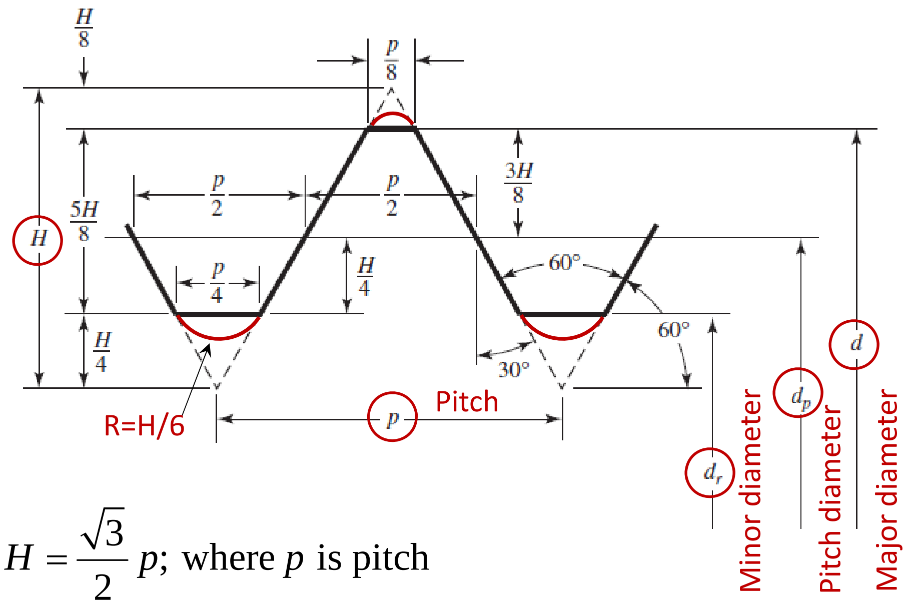
- General term for **elements** used to connect parts
  - o Bolts and nuts
  - o Rivets
  - o Screws
  - o Pins
  - o Locking clips
  - o Set screws
  - o Studs
- Boeing 747 has 2.5 million of them
- **Design of joints is very critical as any assembly is as weak as the weakest joint**



- **Bolts are specified by their major diameter and pitch**
- **Example: M8 x 1.25**
  - **Major diameter: 8 mm**
  - **Pitch: 1.25 mm**

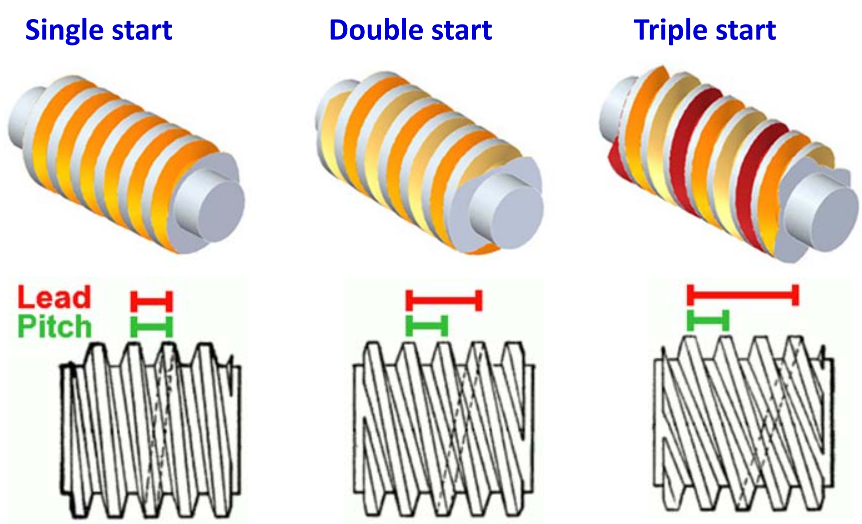
Pitch is distance measured parallel to axis between corresponding points on adjacent threads

Standard ISO Metric thread profile



$$H = \frac{\sqrt{3}}{2}p;\ \text{where } p \text{ is pitch}$$

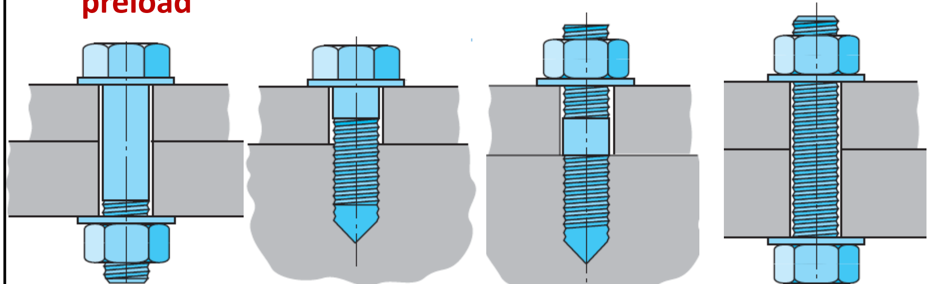
Single start and Multi-start threads



- Lead = number of starts x pitch
- Used when fast action is required (lead screws)

Bolted joints

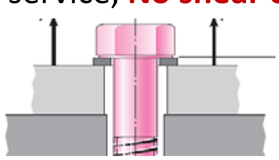
- The purpose of bolt and nut is to clamp parts together
- When assembled the bolt and nut together exerts a clamping force on the members of the joint
- The clamping force is obtained by turning the nut until the bolt has elongated almost to its elastic limit
- This induces a tensile stress in the bolt: **bolt pretension or preload**



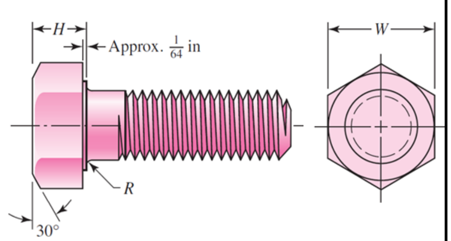
Bolt and Nut Cap screw Stud and nut Threaded rod and nut

Bolted joints

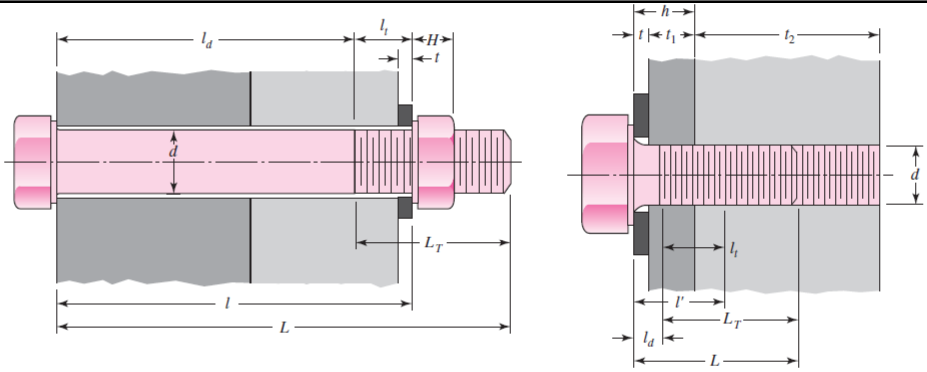
- ➢ Proper **design** and proper **assembly** are equally important
- ➢ Bolt hoes in members should be clearance holes
- ➢ Bolt should experience **only axial tensile load** during service, **No shear or bending**



- ➢ Points of stress concentration
  - o Fillet (radius R)
  - o Start of threads (run out)
  - o Thread root fillet in the plane of the nut when nut is present



- ➢ Use washers made of hardened steel to prevent the sharp edges of bolt holes biting into the fillet of the bolt



Metric series:

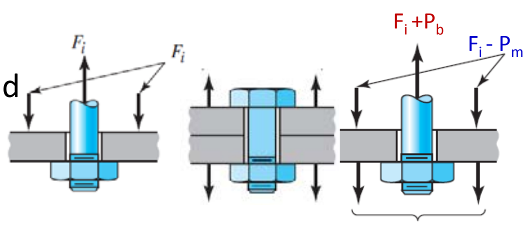
$$L_T = \begin{cases} 2d + 6 \text{ mm}, & L \le 125, d \le 48 \text{ mm} \\ 2d + 12 \text{ mm}, & 125 < L \le 200 \text{ mm} \\ 2d + 25 \text{ mm}, & L > 200 \text{ mm} \end{cases}$$

$$l' = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 \ge d \end{cases}$$

- The correct bolt length is when only couple of threads extend out of the nut after assembly

Tension joints

F_i – Preload; P – External load
k_m – stiffness of members,
k_b – bolt stiffness, $k_m >> k_b$

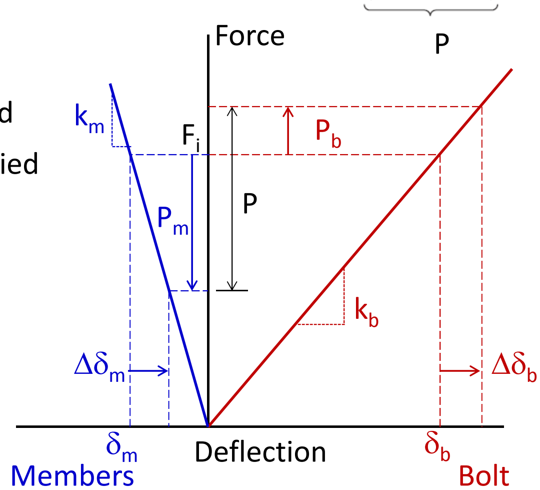


- During preloading
  - bolt elongates
  - members are compressed
- When external load is applied
  - bolt elongates further
  - members relax

$$\Delta\delta_m = \Delta\delta_b;\ \Delta\delta_m = \frac{P_m}{k_m};\ \Delta\delta_b = \frac{P_b}{k_b}$$

$$P = P_b + P_m$$

$$P_b = \frac{k_b P}{k_b + k_m};\ P_m = \frac{k_m P}{k_b + k_m}$$



$$C = \frac{k_b}{k_b + k_m}; \text{ Joint stiffness coefficient}$$

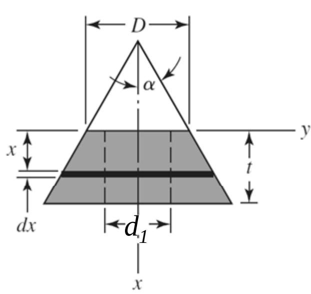
$$P_b = CP; \; P_m = (1-C)P$$

- Tensile Area (A_t): The area to be used in strength calculations
- Least area is at root diameter (d_r)

$$A_t = \frac{\pi}{4}\left[\frac{d_p + d_r}{2}\right]^2$$

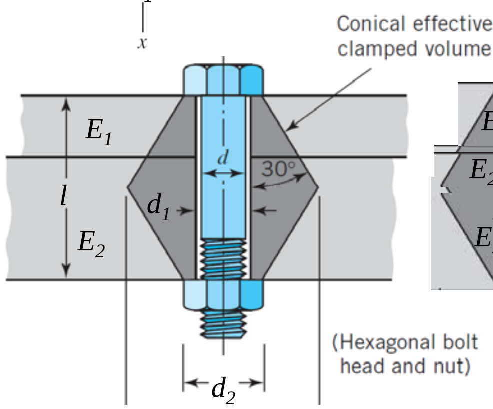
- Tests indicate that an **un-threaded rod** of cross-sectional area equal to A_t calculated above has the same tensile strength as a bolt of major diameter *d*

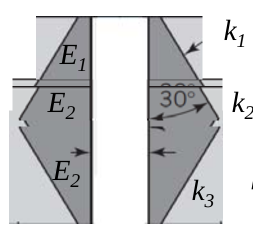
Joint stiffness



$$\frac{k}{\pi E d \tan\alpha} = \ln\left\{\frac{(2t\tan\alpha + D - d_1)(D + d_1)}{(2t\tan\alpha + D + d_1)(D - d_1)}\right\}$$

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \ldots.$$



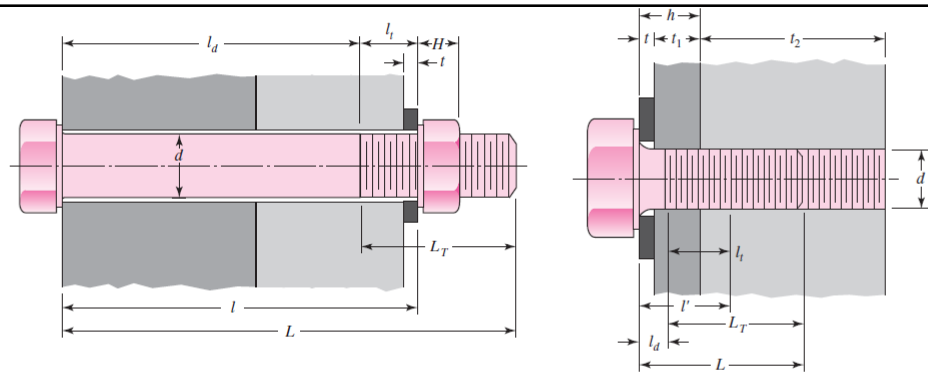


d_2 = 1.5d

d_1 = d

$$k_b = \frac{AE}{l} = \frac{\pi d^2 E}{4l}$$

l = Grip length



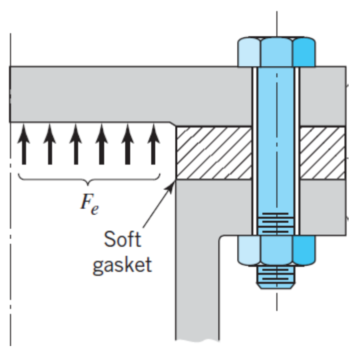
Metric series:

$$L_T = \begin{cases} 2d + 6 \text{ mm}, & L \le 125, d \le 48 \text{ mm} \\ 2d + 12 \text{ mm}, & 125 < L \le 200 \text{ mm} \\ 2d + 25 \text{ mm}, & L > 200 \text{ mm} \end{cases}$$

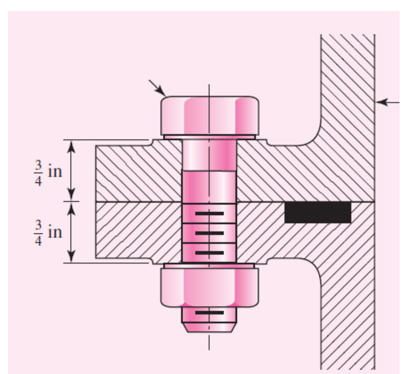
$$l' = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 \ge d \end{cases}$$

- Calculate the grip length (l or l' correctly)

Joint stiffness- Gasket joints



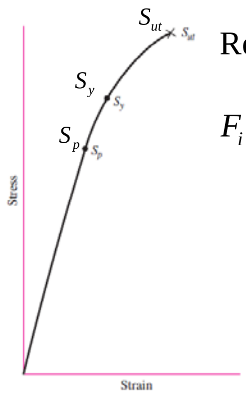
Unconfined Gasket



Confined Gasket

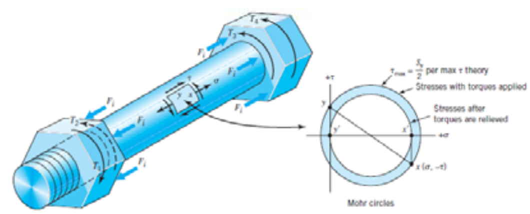
$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_g} \sim \frac{1}{k_g}$$

Preload



Recommended preload

$$F_i = \begin{cases} 0.75F_p \text{ ; Non-permanent connection} \\ 0.9F_p \text{ ; Permanent connection} \end{cases} \text{ ; } F_p = S_p A_t$$



Stresses during preloading

$$\sigma_i = \frac{F_i}{A_t}, \ \tau = 0.5\frac{16T}{\pi d^3}; \ T = KF_i d$$

If no conditions mentioned, $K = 0.2$

Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596

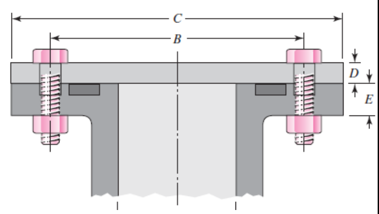
- By default coarse pitch is to be used
- Fine pitch
  - Reduction in cross section is less (thin walled pipes)
  - Better resistance to loosening
  - Better alignment
  - Easy to make in harder materials

Table 8–11

Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs*

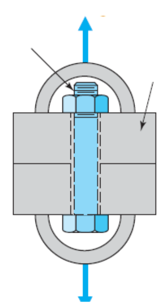
Property Class	Size Range, Inclusive	Minimum Proof Strength,† MPa	Minimum Tensile Strength,† MPa	Minimum Yield Strength,† MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9

Design for static load



Bolt failure during service

$$\sigma_b = n_d \frac{CP}{A_t} + \frac{F_i}{A_t} = S_p; \;\; n_f = \frac{S_p A_t - F_i}{CP}$$

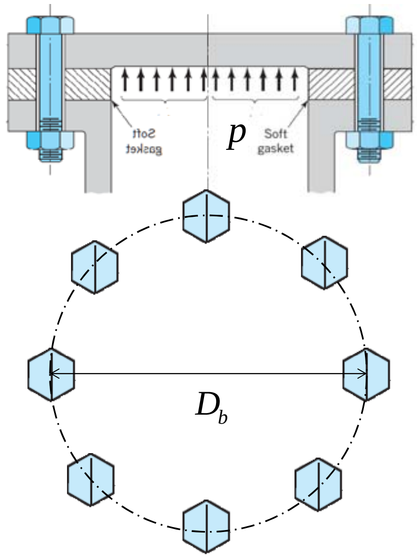


Separation of joint; loss of sealing

$(1-C)P_o = F_i$; P_o - Force at joint separation

Factor of safety, $n_o = \dfrac{P_o}{P} = \dfrac{F_i}{P(1-C)}$

Gasketed joints



$$P = \frac{p\pi D_s^2}{4};\ D_s - \text{ sealing diameter}$$

$$\text{Gasket pressure, } p_g = \frac{F_m}{A_g / N}$$

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

A_g – Total gasket area

N – Number of bots

$$S = \frac{\pi D_b}{N};\ 3d \le S \le 6d$$

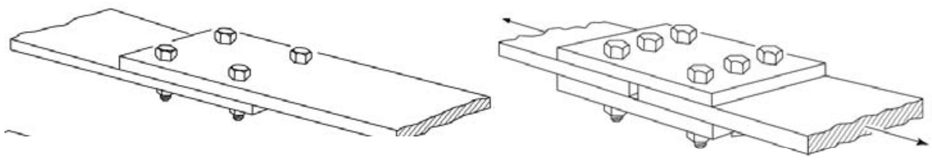
Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

SAE Grade	Metric Grade	Rolled Threads	Cut Threads	Fillet
0 to 2	3.6 to 5.8	2.2	2.8	2.1
4 to 8	6.6 to 10.9	3.0	3.8	2.3

Grade or Class	Size Range	Endurance Strength
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

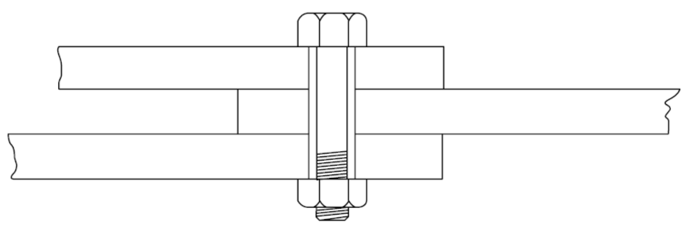
*Repeatedly-applied, axial loading, fully corrected.

Bolted ioints in shear



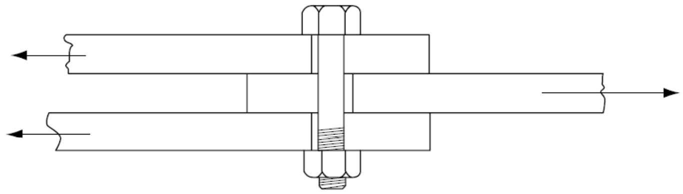
- The purpose of a bolt in a shear loaded joint is to prevent relative slipping of the members of the joint
- Slipping is prevented either by friction due to the clamping force or by the “shear pin” action of the bolt
  - Friction type joints: Bolt and bolt holes do not contact
  - Bearing type joints: Bolt and bolt hole bears against each other
- Reality: Neither of the above

Friction type joints



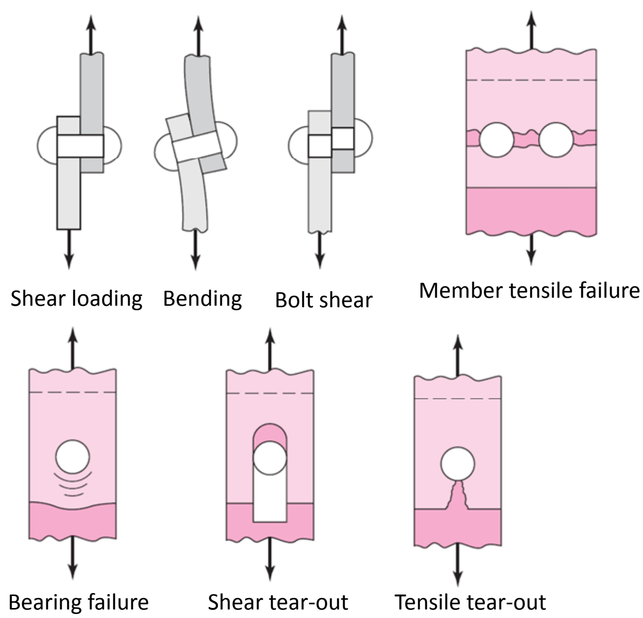
- The recommend pre-load is at least equal to the proof load
- As long as there is no slip
  - the stress in the bolt is only due to the pre-load
  - The entire assembly acts like a single solid

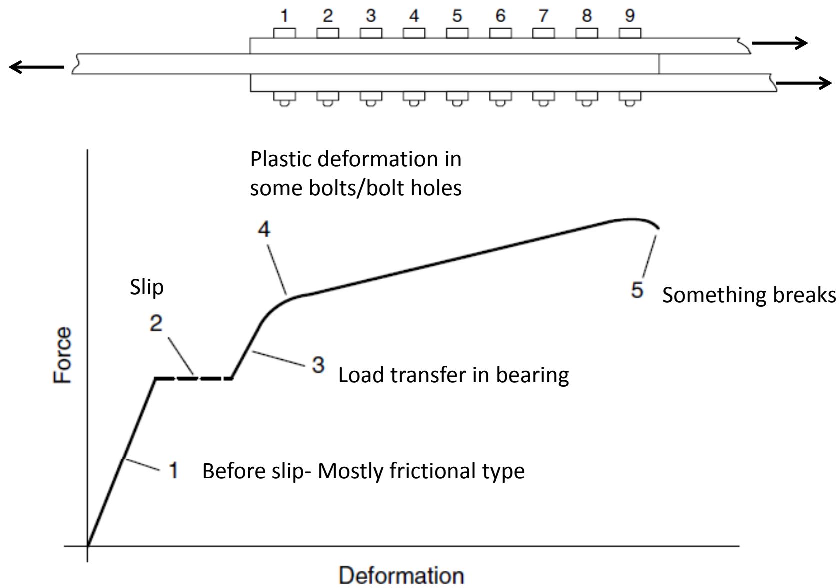
Bearing type joints



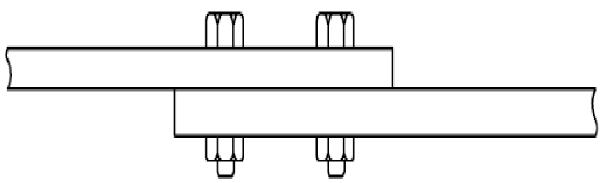
- The bolt acts like a shear-pin
- The bolts are subjected to shear stresses
- There is no clear recommendation on pre-load
- The external load does not alter the pre-load
- Pre-load prevents self loosening
- Apply maximum pre-load such that **nothing** in the joint yields

Failure modes





Design



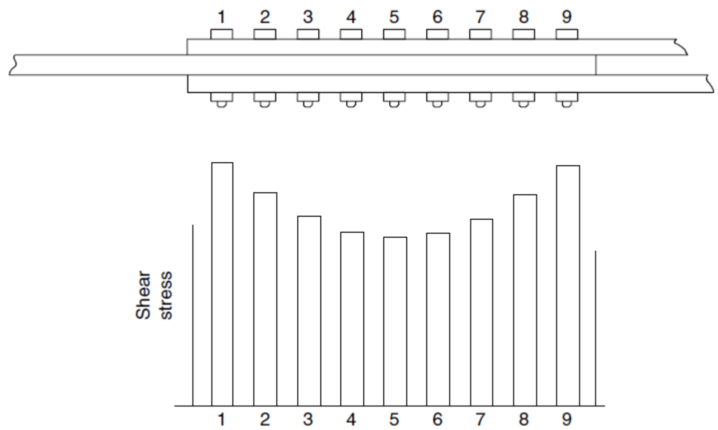
- Bolt-shear

$$\tau_s = \frac{F}{nA_r};\quad n_f = \frac{S_{sy}}{\tau_s}$$

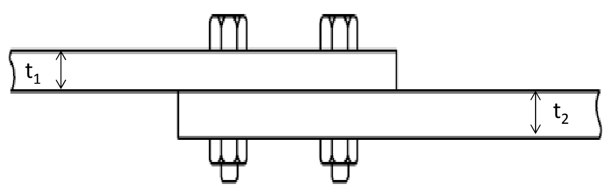
A_r - Area based on minor diameter (Table 8-1)

$$S_{sy} = 0.577 S_y$$

- What about stress due to pre-load?
- Tests indicate that shear strength of high strength bolts is not affected by the pre-load



- Shear stresses in bolts vary substantially



- Bearing on bolts: Failure by compressive contact stresses

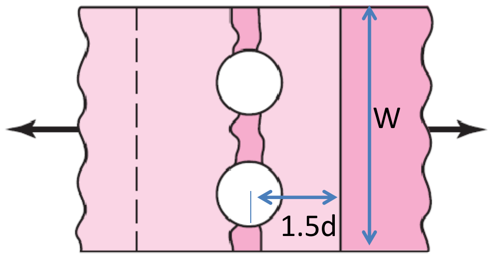
$$\sigma_b = \frac{F}{nA_b}; \quad n_f = \frac{S_{yc}}{\sigma_b}$$

A_b - $\min\{t_1, t_2\} \times d$: Projected area is used

S_{yc} = Yiled strength of bolt in compression

- Bearing on members:

$n_f = \frac{S_{yc}}{\sigma_b}$; S_{yc} = Yiled strength of member in compression

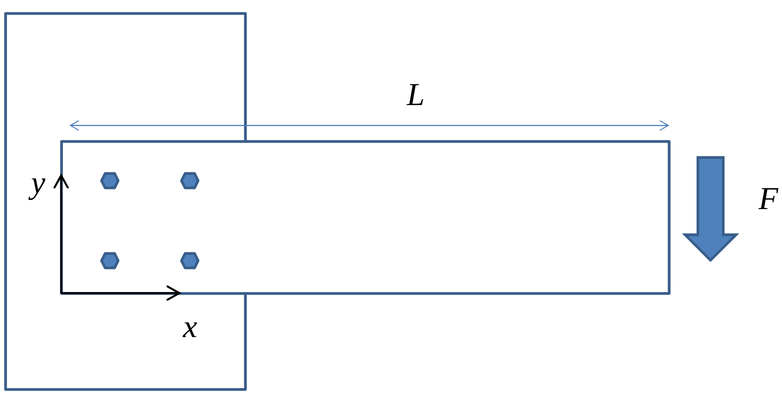


- Tensile failure of members

$$A_{tm} = \{W - nd\} t_{1or2};\ \sigma_t = \frac{F}{A_{tm}};\ n_f = \frac{S_y}{\sigma_t}$$

- Edge shearing and tensile tearing can be avoided by keeping the bolt hole center at a distance of 1.5d from the edge of the plate

Joints loaded in shear and torsion



- The number and arrangement pattern of the bolts is known

Joints loaded in shear and torsion

- The joint should resist the direct shear force and the moment due to this shear force
- Calculate the centroid of the bolt pattern

$$\bar{x} = \frac{\sum x_i A_i}{\sum A_i}; \ \bar{y} = \frac{\sum y_i A_i}{\sum A_i}; \quad A_i = \frac{\pi d^2}{4}$$

- The moment (M) of the force is calculated about this centroid

Force due to direct or primary shear: $F' = \dfrac{F}{n}$

Force due to moment or secondary shear: $F_i'' = \dfrac{M r_i}{\sum r_i^2}$

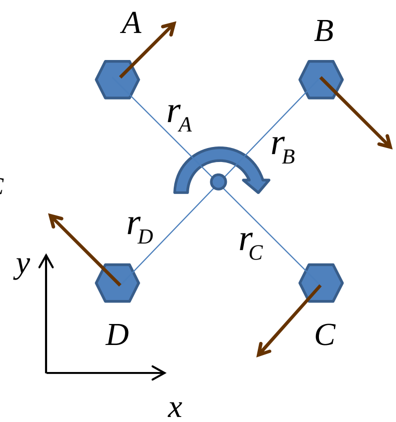
- The bolt farther away from centroid carries larger secondary shear. Why?
- Assumptions
  - Members are rigid and effect of moment is to rotate one member w.r.t other
  - Displacement (tangential) is more at location away from centroid
  - Shear strain on bolt is proportional to its radial distance from the centroid

$$\frac{F''_A}{r_A} = \frac{F''_B}{r_B} = \frac{F''_C}{r_C} = \frac{F''_D}{r_D} = k$$

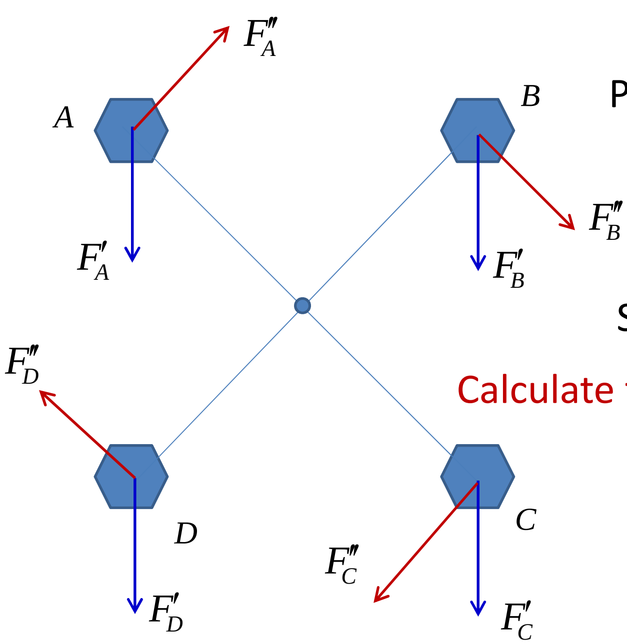
$$M = F''_A r_A + F''_B r_B + F''_C r_C + F''_C r_C$$

$$M = k\left\{r_A^2 + r_B^2 + r_C^2 + r_D^2\right\}$$

$$k = \frac{M}{\sum r_i^2} \qquad F''_i = \frac{M r_i}{\sum r_i^2}$$



- How to calculate the resultant force in each bolt?



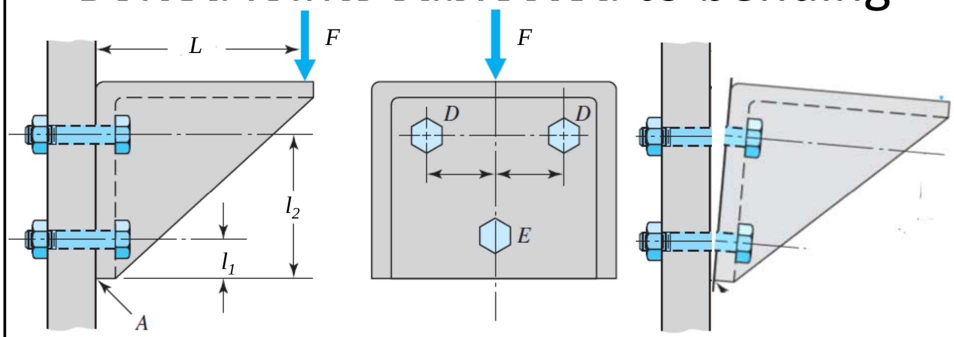
Primary Shear

Secondary Shear

Calculate the resultant, F_r

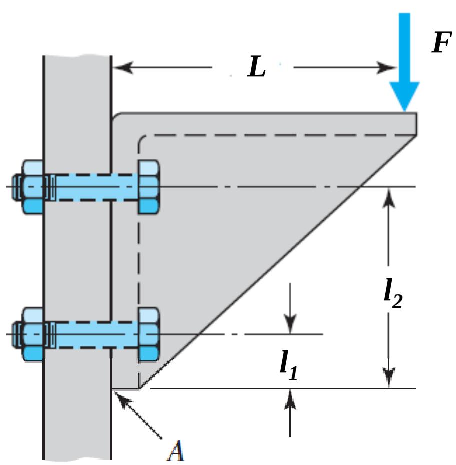
$$\tau = \frac{F_r}{A_r}$$

Bolted joints subjected to bending



- We will assume the members are rigid and the bracket will rotate about the lower edge
- The bolts farthest from the bottom edge as more elongation and hence experience more force

- Assume all bolts are of same type (i.e area of cross section and material the same)



$$F_i \propto l_i; \quad F_i = kl_i$$

$$M = k\sum_i l_i^2$$

$$k = \frac{M}{\sum_i l_i^2}$$

$$\sigma_i = \frac{F_i}{A_r} \qquad \tau = \frac{F}{nA_r}$$

Use von-Mises theory to calculate factor of safety