MIET2510

Mechanical Design

Week 8 – Fasteners and Connections – Part 1

School of Science and Technology, RMIT Vietnam

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Introduction

The manufacture of intricate products often requires assembly of components and several options exist, including threaded fasteners, rivets, welds, and adhesive joints.

Introduction

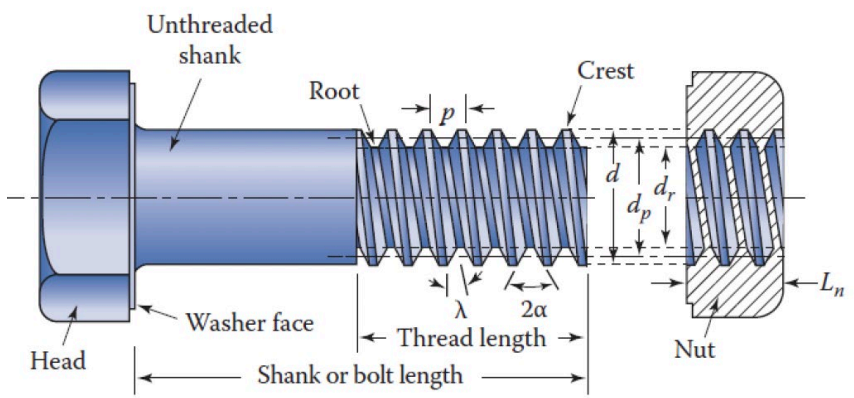
- We will review
  - Power Screws
  - Fasteners
    - Threaded connections
    - Rivets
    - Snap fasteners
  - Welding (Part 2)

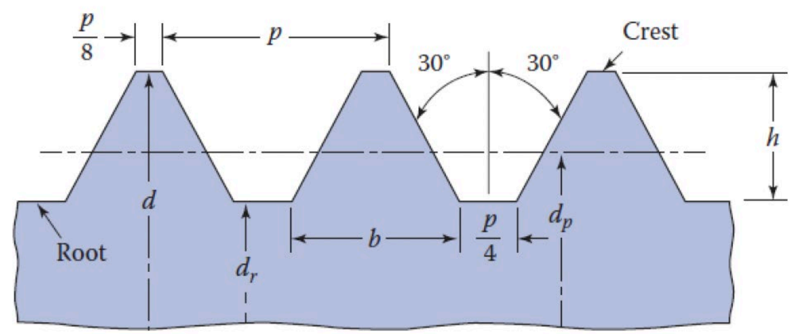
Thread Standard and Definition

Basic Dimensions of ISO (Metric) Screw Threads

	Coarse Threads		Fine Threads	
Nominal Diameter, d (mm)	Pitch, p (mm)	Tensile Stress Area, A_t (mm²)	Pitch, p (mm)	Tensile Stress Area, A_t, (mm²)
2	0.4	2.07		
3	0.5	5.03		
4	0.7	8.78		
5	0.8	14.2		
6	1	20.1		
7	1	28.9		
8	1.25	36.6	1.25	39.2
10	1.5	58.0	1.25	61.2
12	1.75	84.3	1.25	92.1
14	2	115	1.5	125
16	2	157	1.5	167
18	2.5	192	1.5	216
20	2.5	245	1.5	272
24	3	353	2	384
30	3.5	561	2	621
36	4	817	2	915
42	4.5	1120	9	1260
48	5	1470	2	1670
56	5.5	2030	2	2300
64	6	2680	2	3030

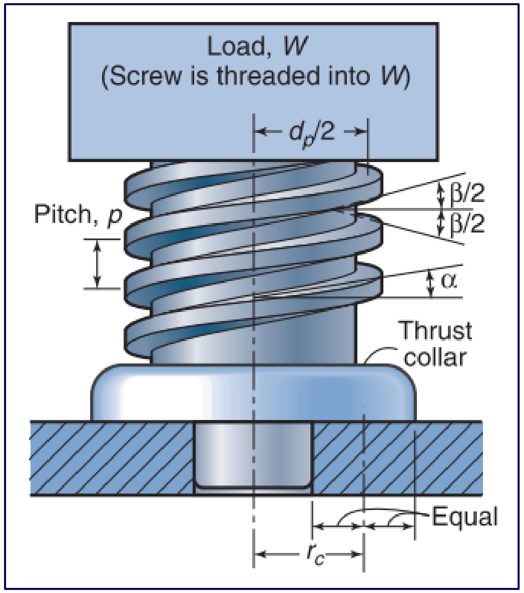
Source: *ANSI/ASME Standards*, B1.1–2014, B1.13–2005, New York, American Standards Institute, 2005.
Notes: Metric threads are specified by nominal diameter and pitch in millimeters, for example, $M10 \times 1.5$. The letter M, which proceeds the diameter, is the clue to the metric designation; root or minor diameter $d_r \approx d - 1.227p$.

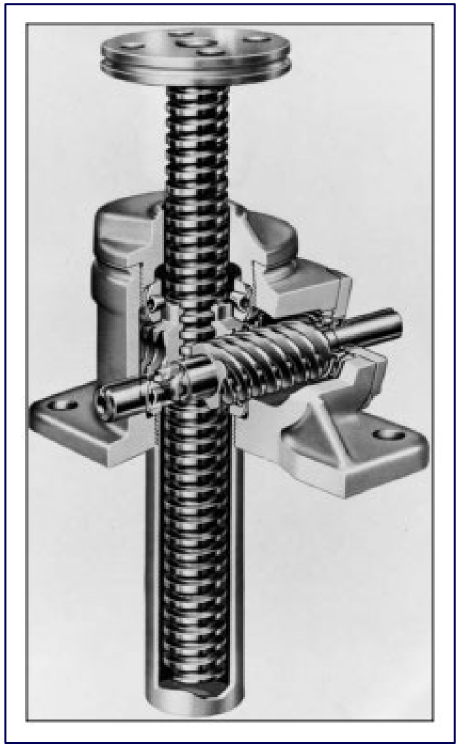




The Mechanics of Power Screws

Power screws are used to convert rotatory motion into linear motion, providing mechanical advantage.



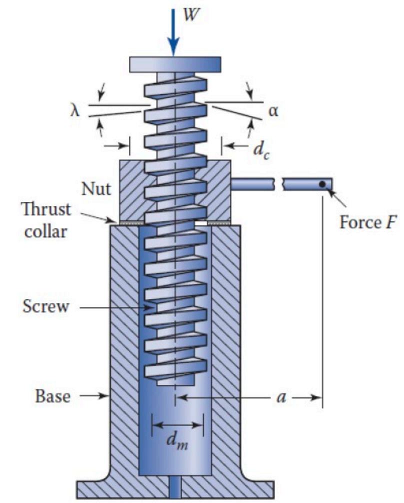


The Mechanics of Power Screws

- Some key expressions used to calculate the amount of torque needed to be applied power screws to lift a load

$$\boxed{\tan\lambda = \frac{L}{\pi d_m}}$$

λ = the helix or lead angle
L = the lead
d_m = the mean diameter of thread contact surface

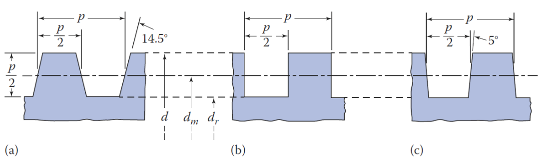


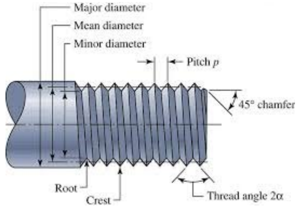
$$\boxed{d_m = d - \frac{p}{2}}$$

d: nominal diameter of the screw (mm)
p: pitch (mm)

$$\boxed{\tan\alpha_n = \cos\lambda\tan\alpha}$$

α_n: normal thread angle
α: thread angle
standard ACME for power screw angle: 14.5°





The Mechanics of Power Screws

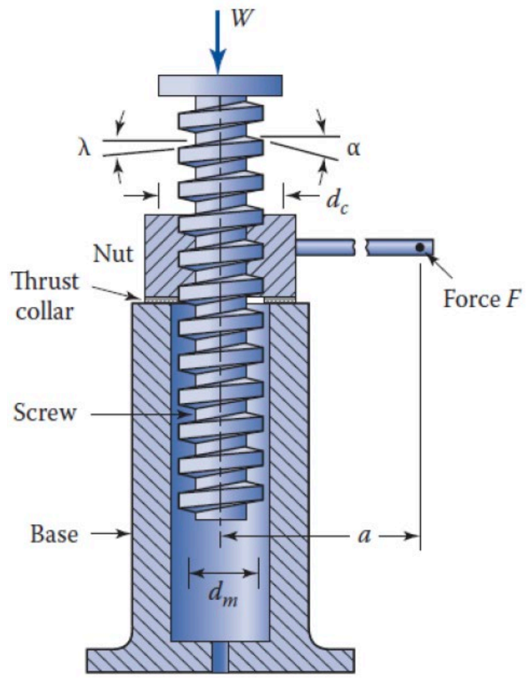
- The torque required to raise the load

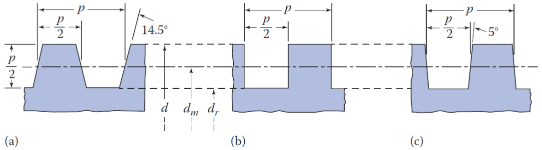
$$T_u = \frac{W d_m}{2} \frac{f + \cos\alpha_n \tan\lambda}{\cos\alpha_n - f \tan\lambda} + \frac{W f_c d_c}{2}$$

- The torque required to lower the load

$$T_d = \frac{W d_m}{2} \frac{f - \cos\alpha_n \tan\lambda}{\cos\alpha_n + f \tan\lambda} + \frac{W f_c d_c}{2}$$

f: friction coefficient (thread friction)
f_c: friction coefficient (collar friction)
W: weight
d_c: mean diameter of the collar





- Efficiency

$$e = \frac{\cos\alpha_n - f \tan\lambda}{\cos\alpha_n + f \cot\lambda}$$

Power Screw - Example

- A screw jack is used to lift a load W = 6kN
- The coefficients of friction are estimated as; $f = 0.12,$ $f_c = 0.09$
- The geometry of the power screw is (mm) $d = 30,$ $p = 4,$ $d_c = 40$ $\alpha = 14.5$
- The screw is quadruple threaded
- Determine
  - The screw lead, mean diameter and helix angle
  - The torque required for lifting and lowering the load
  - The efficiency of the jack when lifting the load
  - The length of a crank required, if F = 150N is exerted by an operator
- Determine the starting torque if; $f = 0.16, f_c = 0.12$

Solution:

$d_m = 28\ mm$

$L = 16\ mm$

$\lambda = 10.31°$

$\alpha_n = 14.28°$

$$\boxed{d_m = d - \frac{p}{2}}$$

$$\boxed{L = np}$$

$$\boxed{\tan\lambda = \frac{L}{\pi d_m}}$$

$$\boxed{\tan\alpha_n = \cos\lambda\tan\alpha}$$

$$\boxed{T_u = \frac{Wd_m}{2}\frac{f + \cos\alpha_n\tan\lambda}{\cos\alpha_n - f\tan\lambda} + \frac{Wf_c d_c}{2}}$$

$$\boxed{T_d = \frac{Wd_m}{2}\frac{f - \cos\alpha_n\tan\lambda}{\cos\alpha_n + f\tan\lambda} + \frac{Wf_c d_c}{2}}$$

$$\boxed{e = \frac{\cos\alpha_n - f\tan\lambda}{\cos\alpha_n + f\cot\lambda}}$$

Threaded Fasteners

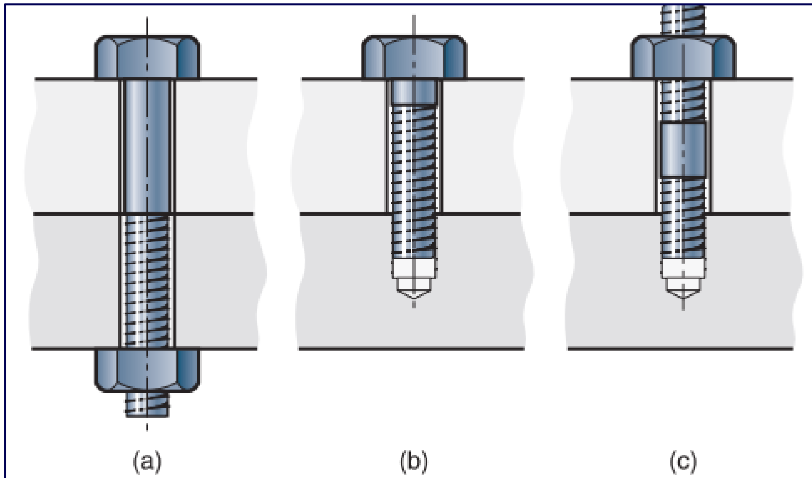


Figure 16.8: Three types of threaded fastener. (a) Bolt and nut; (b) cap screw; (c) stud.

- A fastener is a device to connect or join two or more members. Many fastener types and variations are commercially available.
- Threaded fasteners usually only carry tension, P.
- The axial stress may be calculated;

$$\sigma = \frac{P}{A}$$

Threaded Fasteners

- Bolts are typically used to hold parts together resisting shear and/or forces that pull them apart.
- The initial preload is important because it is this that resists the forces attempting to separate the parts
- Tightening torque is used to ‘load’ the fastener;

$$\boxed{T = KdF_i}$$

T = the tightening torque
d = the nominal bolt diameter
K = the torque coefficient
F_i = the initial tension or preload

Threaded Fasteners

- Care should be taken not to overtighten the fastener causing deformation
- The proof load, F_p is the load that the fastener can carry without deforming

$$\boxed{F_i = \begin{cases} 0.75F_p & \text{(reused connections)} \\ 0.9F_p & \text{(permanent connections)} \end{cases}}$$

$$\boxed{F_p = S_p A_t}$$

A_t: tensile stress area (ISO standard)
$S_p = 0.9S_y$: proof strength,
based on the yield strength S_y

Threaded Fasteners

Metric Specifications and Strengths for Steel Bolts

Class Number	Size Range Diameter, *d* (mm)	Proof Strength, S_p (MPa)	Yield Strength, S_y (MPa)	Tensile Strength, S_u (MPa)	Material Carbon Content
4.6	*M5–M36*	225	240	400	Low or medium
4.8	*M1.6–M16*	310	340	420	Low or medium
5.8	*M5–M24*	380	420	520	Low or medium
8.8	*M3–M36*	600	660	830	Medium, Q&T
9.8	*M1,6–M 16*	650	720	900	Medium, Q&T
10.9	*M5–M36*	830	940	1040	Low, martensite, Q&T
12.9	*M1.6–M36*	970	1100	1220	Alloy, Q&T

Source: Society of Automotive Engineers Standard J429k, 2011.

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Threaded Fasteners

- In fastening parts together;

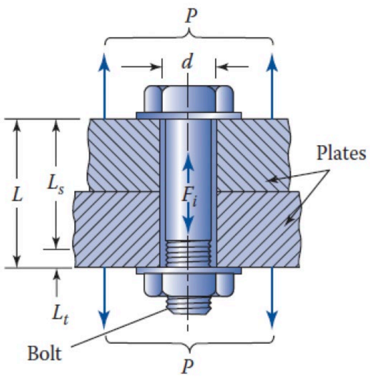
$$P = F_b + F_p$$

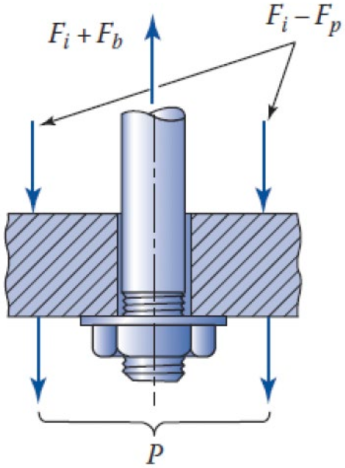
F_b: tensile force in the bolt
F_p: clamping force between parts

- The deformation of the bolt and the parts are defined by

$$\delta_b = \frac{F_b}{k_b}, \quad \delta_p = \frac{F_p}{k_p} \qquad \text{and} \qquad \frac{F_b}{k_b} = \frac{F_p}{k_p}$$

- Where the stiffness of the bolt and parts, k_b and k_p are determined by their respective mechanical properties





Threaded Fasteners

- The forces are given

$$F_b = \frac{k_b}{k_b + k_p} P = CP, \quad F_p = \frac{k_p}{k_b + k_p} P = (1-C)P$$

- Where C is the joint’s stiffness factor

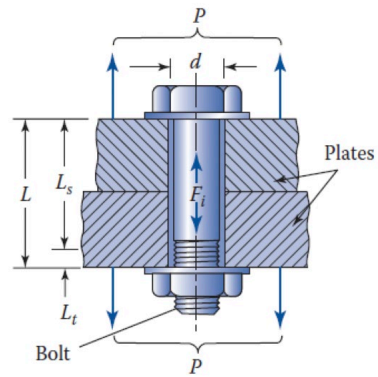
$$C = \frac{k_b}{k_b + k_p}$$

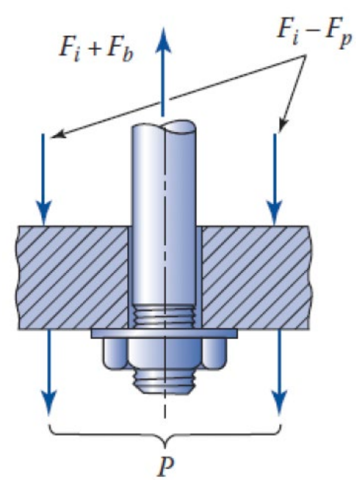
- The total forces in the bolt and in the parts considers the initial pre-tension

$$F_b = CP + F_i$$

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

F_b = the bolt axial tensile force
F_p = the lamping force on the two parts
F_i = the initial tension or preload





Threaded Fasteners

- When the bolt is not fully threaded, the stiffness of the bolt must take account of the threaded and unthreaded parts.

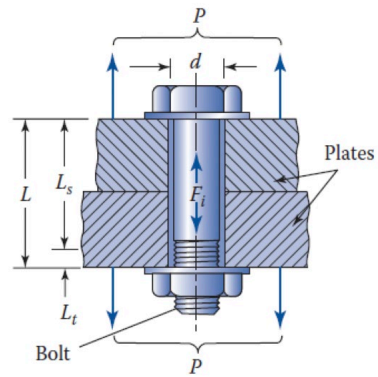
$$\frac{1}{k_b} = \frac{L_t}{A_t E_b} + \frac{L_s}{A_b E_b}$$

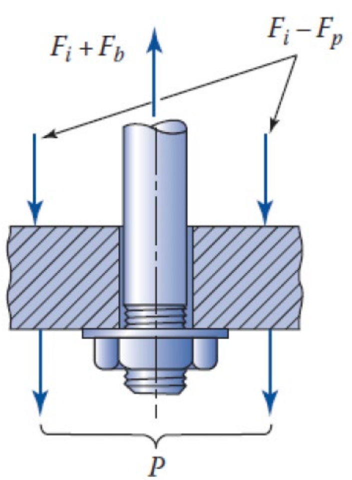
A_b: gross cross-sectional area
A_t: the tensile stress area of the bolt.

$$L_t = \begin{cases} 2d+6 & L \le 125 \\ 2d+12 & 125 < L \le 200 \\ 2d+25 & L > 200 \end{cases}$$

L: Total bolt length
d: bolt diameter

$$L_s = L - L_t$$





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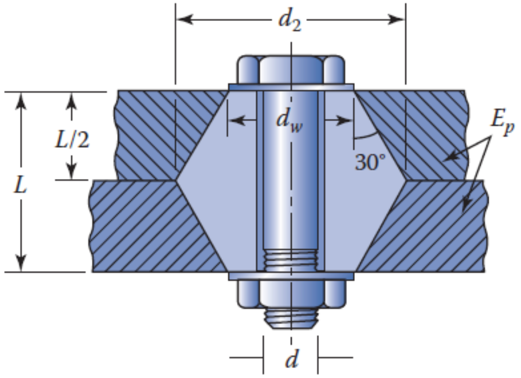
Threaded Fasteners

- The clamped parts stiffness can be estimated

$$A_p = \frac{\pi}{4}\left[\left(\frac{d_w + d_2}{2}\right)^2 - d^2\right] \qquad k_p = \frac{A_p E_p}{L}$$

d = the bolt diameter
L = the grip
E_p = the modulus of elasticity of the single or two identical parts

$d_w = 1.5d \qquad d_2 = d_w + L \tan 30°$



$$k_p = \frac{0.58\pi E_p d}{2\ln\left(5\dfrac{0.58L + 0.5d}{0.58L + 2.5d}\right)}$$

Threaded Fasteners

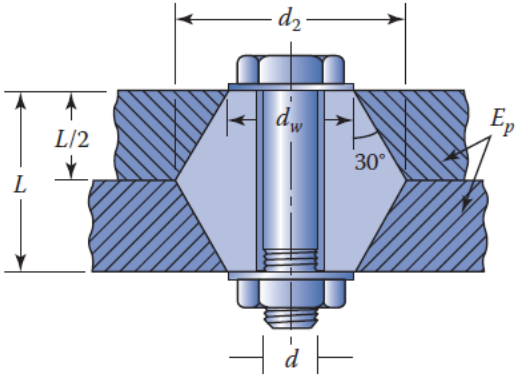
- In designing a joint the external load must be smaller than that load required to separate the joint.
- A bolt safety factor can be calculated

$$n = \frac{S_p A_t - F_i}{CP} = \frac{S_y}{\sigma_b}$$

- As well as a load safety factor

$$n_s = \frac{P_s}{P} = \frac{F_i}{P(1-C)} \qquad P_s = \frac{F_i}{(1-C)}$$

- Where P_s is the load required to separate the joint



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Threaded Fasteners - Example

- A steel bolt-and-nut clamps a steel cylinder of known cross section and length subjected to an external load P.

$D = 20$ mm, $L = 65$ mm, $d = 10$ mm, $E = E_b = E_p = 200$ GPa

$P = 8$ kN $\quad A_t = 58$ mm² (from Table 15.2)

$S_p = 380$ MPa and $S_y = 420$ MPa (by Table 15.5)

a. Preload and bolt tightening torque
b. Joint stiffness factor
c. Maximum tensile stress in the bolt
d. Factors of safety against yielding and separation

- Cross-sectional area of the parts:

$$A_p = \pi(D^2 - d^2)/4 = 235.6\ mm^2$$

- Preloading is:

$$F_i = 0.75F_p = 0.75S_pA_t = 16.53\ kN$$

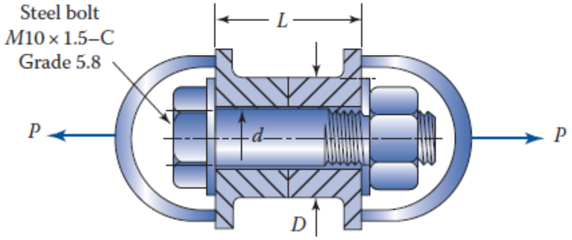
- Bolt tightening torque:

$$T = 0.2F_id = 33.06\ Nm$$

- Length of thread L_t and shank L_s:

$$L_t = 2d + 6 = 2(10) + 6 = 26 \text{ mm}$$

$$L_s = L - L_t = 65 - 26 = 39 \text{ mm}$$



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Threaded Fasteners - Example

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Threaded Fasteners - Example

- Stiffness constant for the bolt:

$$\frac{1}{k_b} = \frac{L_t}{A_t E} + \frac{L_s}{A_s E} = \frac{1}{200\left(10^6\right)}\left[\frac{26}{58} + \frac{39(4)}{\pi(10)^2}\right], \quad k_b = 2.117\left(10^8\right)\text{ N/m}$$

- Stiffness constant for the parts:

$$k_p = \frac{A_p E}{L} = \frac{235.6\times10^{-6}\left(200\times10^9\right)}{65\times10^{-3}} = 7.249\left(10^8\right)\text{ N/m}$$

- The joint stiffness factor:

$$C = \frac{k_b}{k_p + k_b} = \frac{2.117}{7.249 + 2.117} = 0.226$$

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Threaded Fasteners - Example

- The forces in the bolt and parts are:

$$F_b = F_i + CP = 16.53 + 0.226(8) = 18.34 \text{ kN}$$

$$F_p = F_i - (1 - C)P = 16.53 - (1 - 0.226)(8) = 10.34 \text{ kN}$$

- The tensile stress in the bolt is:

$$\sigma_b = \frac{F_b}{A_t} = \frac{18.34\left(10^3\right)}{58\left(10^{-6}\right)} = 316 \text{ MPa}$$

- The safety factor against yielding is:

$$n = \frac{S_y}{\sigma_b} = \frac{420}{316} = 1.33$$

- The safety factor against separation is:

$$P_s = \frac{F_i}{(1-C)} = \frac{16.53}{(1-0.226)} = 21.36 \text{ kN} \qquad n_s = \frac{P_s}{P} = \frac{21.36}{8} = 2.67$$

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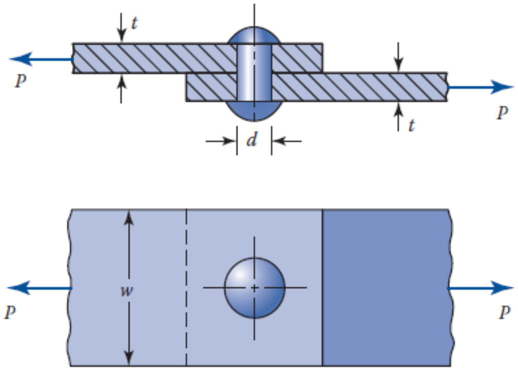
Riveted Fasteners



Riveted Fasteners

- Primary failure modes are;
  - Bending of a member
  - Shear of rivet
  - Tensile failure of member
  - Bearing of member on rivet

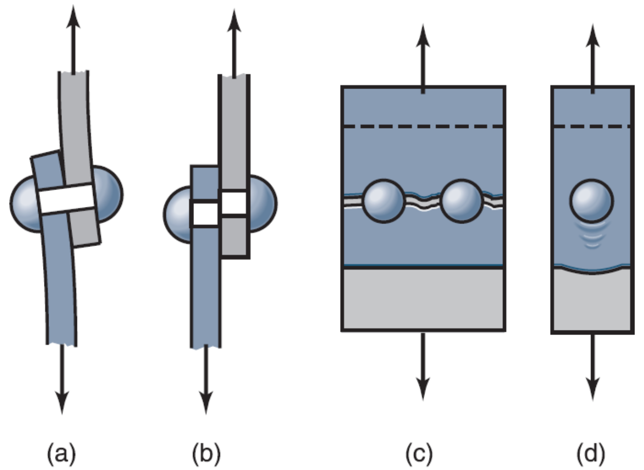


Figure 16.19: Failure modes due to shear loading of riveted fasteners. (a) Bending of member; (b) shear of rivet; (c) tensile failure of member; (d) bearing of member on rivet.

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Riveted Fasteners

- To avoid failure of the joint due to the bending of a member the following condition should be met;

$$\frac{S_{yj}}{n_s} = \frac{PL}{2Z_m},$$

where L is the grip length, Z_m is the section modulus of the weakest member, where $Z_m = I/c$, and S_{yj} is the yield strength of weakest member.

- To avoid failure due to shearing of the rivets;

$$\frac{S_{sy}}{n_s} = \frac{4P}{\pi d_c^2},$$

where d_c is the crest diameter and S_{sy} is the rivet yield strength in shear. In Eq. (16.49), the diameter of the rivet rather than the diameter of the hole is used. Normally,

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Riveted Fasteners

- To avoid tensile failure of a member

$$\frac{S_{yj}}{n_s} = \frac{P}{(b - N_r d_c) t_m},$$

where b is the width of the member, N_r is the number of rivets across the member's width, and t_m is the thickness of the member.

- To avoid compressive bearing failure

$$\frac{S_{yj}}{n_s} = \frac{P}{d_c t_m}$$

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Snap Fasteners

- Snap fasteners are popular because they can make assembly very easy.
- When used with plastic housings they can be moulded directly on to the assembled parts.

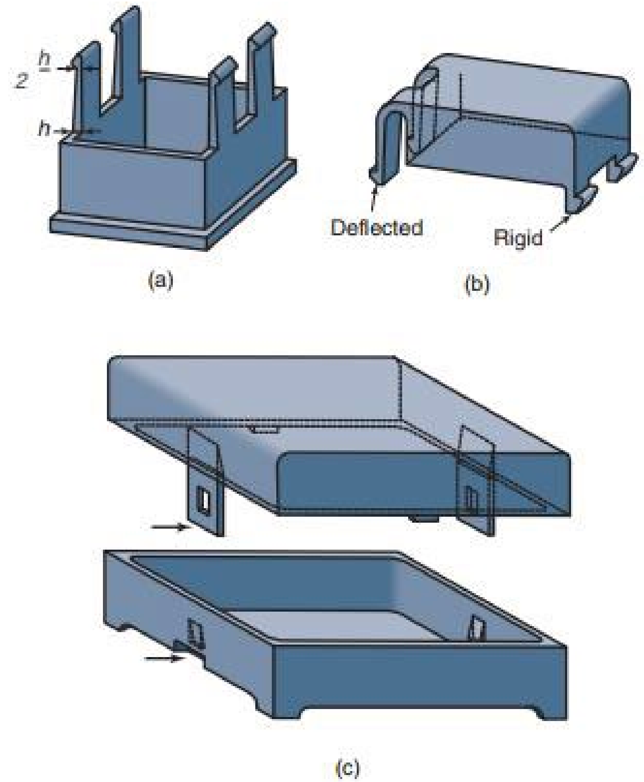

Figure 16.31: Common examples of integrated fasteners. (a) Module with four cantilever lugs; (b) cover with two cantilever and two rigid lugs; (c) separable snap joints for chassis cover.

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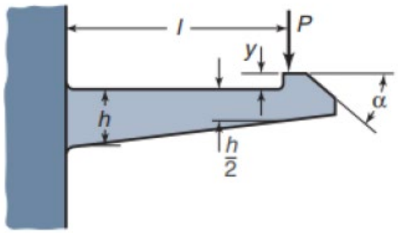
Snap Fasteners

- The analysis of snap fasteners can be complicated.
- The required mating force for a cantilever snap fastener can be calculated

$$W = P\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha},$$

$$P = \frac{bh^2}{6}\frac{E_s\epsilon}{l},$$

where b is the cantilever width and E_s is the secant modulus of elasticity.



	Cross section: Rectangle	Cross section: Trapezoid
Constant cross section	0.67	$\frac{a+b}{2a+b}$
Tapered thickness ($\frac{h}{2}$)	1.09	$1.64\,\frac{a+b}{2a+b}$
Tapered width ($\frac{b}{4}$)	0.86	$1.28\,\frac{a+b}{2a+b}$

Figure 16.33: Shape constant, A, used to obtain deflection of snap fastener cantilevers.

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Thank you for your attendance :D

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Reference

- *Mechanical Design of Machine Components (2nd) by Ansel C.Ugural.*
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