- by giving the number of threads per inch N for the Unified sizes.

Size designation	nominal major diameter in	threads per inch N	tensile stress area A_t in^2	minor diameter area A_r in^2
12	0.2160	24	0.0242	0.0206

(Table 8-2, page 328) → $\underbrace{0.216''}_{\text{the nominal major diameter}} - \underbrace{24}_{\text{number of threads per inch}}\ \underbrace{UNRF}_{\text{thread series}}$

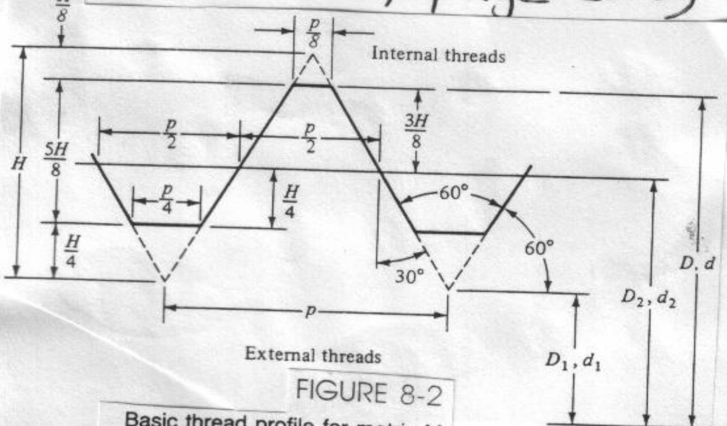


FIGURE 8-2
Basic thread profile for metric M and MJ threads. D (d) = basic major diameter of internal (external) thread; D_1 (d_1) = basic minor diameter of internal (external) thread; D_2 (d_2) = basic pitch diameter of internal (external) thread; p = pitch; $H = 0.5(3)^{1/2} p$.

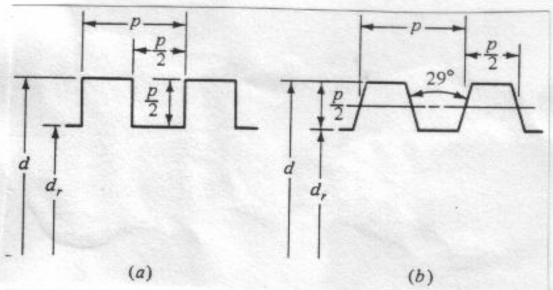


FIGURE 8-3
(*a*) Square thread; (*b*) Acme thread.

- Square and Acme threads are used on screws when power is to be transmitted

The mechanics of power screws

- A power screw is a device used in machinery to change angular motion into linear motion, and, usually, to transmit power.

d_m : mean diameter

p : pitch

λ : lead angle

ψ : helix angle

- find an expression for the torque required to raise this load
- find an expression for the torque required to lower the load

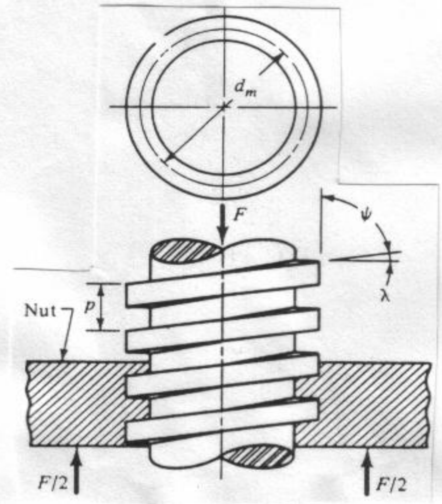


FIGURE 8-5
Portion of a power screw.

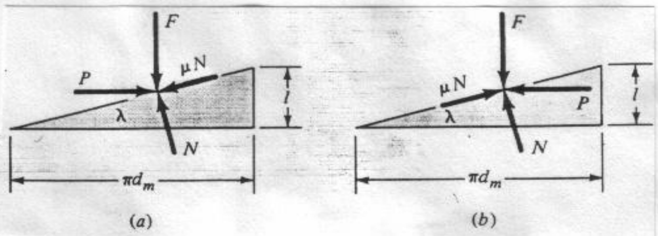


FIGURE 8-6

Force diagrams: (a) lifting the load; (b) lowering the load.

- for raising the load

$$\left.\begin{aligned}\sum F_H &= P - N\sin\lambda - \mu N\cos\lambda = 0\\ \sum F_V &= F + \mu N\sin\lambda - N\cos\lambda = 0\end{aligned}\right\}\quad P = \frac{F(\sin\lambda + \mu\cos\lambda)}{\cos\lambda - \mu\sin\lambda}$$

$$\tan\lambda = l/\pi d_m \Rightarrow P = \frac{F\left[(l/\pi d_m) + \mu\right]}{1 - (\mu l/\pi d_m)}$$

$$T = P \cdot d_m/2 = \frac{F d_m}{2}\left(\frac{l + \pi\mu d_m}{\pi d_m - \mu l}\right)$$

- for lowering the load

$$\left.\begin{aligned}\sum F_H &= -P - N\sin\lambda + \mu N\cos\lambda = 0\\ \sum F_V &= F - \mu N\sin\lambda - N\cos\lambda = 0\end{aligned}\right\}\quad T = \frac{F d_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right)$$

if $T > 0 \Rightarrow$ self-locking $\Rightarrow \mu > \tan\lambda$

- efficiency

assume $\mu = 0 \Rightarrow T_0 = \dfrac{Fl}{2\pi} \Rightarrow e = \dfrac{T_0}{T} = \dfrac{Fl}{2\pi T}$

- for square threads

raising the load $\Rightarrow T = \dfrac{F d_m}{2}\left(\dfrac{l + \pi\mu d_m \sec\alpha}{\pi d_m - \mu l \sec\alpha}\right)$

<u>Threaded fasteners</u>

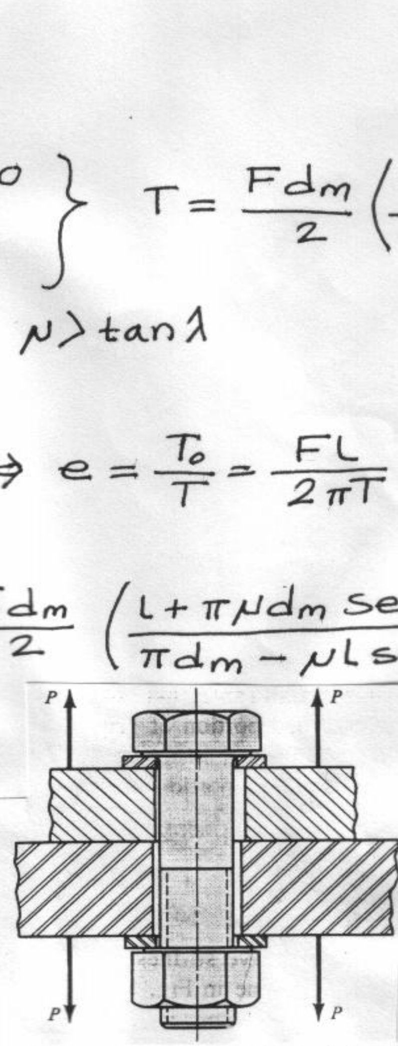


FIGURE 8-12

A bolted connection loaded in tension by the forces P. Note the use of two washers. A simplified conventional method is used here to represent the screw threads. Note how the threads extend into the body of the connection. This is usual and is desired.

- thread length
  - inch series

$$L_T = \begin{cases} 2D + \frac{1}{4}\text{ in} & L \le 6\text{ in}\\ 2D + \frac{1}{2}\text{ in} & L > 6\text{ in}\end{cases}$$

  - metric bolts

$$L_T = \begin{cases} 2D + 6 & L \le 125 \quad D \le 48\\ 2D + 12 & 125 < L \le 200\\ 2D + 25 & L > 200\end{cases}$$

- The purpose of a bolt is to clamp two or more parts together.
- Never reuse nuts, it can be dangerous.

a thrust or collar bearing between the rotating and stationary members

$$T_c = \frac{F \cdot f_c \cdot d_c}{2}$$

stresses in power screws (at the top of the root "plane")

$$\tau = \frac{16T}{\pi \cdot d_r^3} = \tau_{xy} \qquad \sigma_b = \frac{6F}{\pi d_r n_t p} = \sigma_x \qquad \tau_{yz} = 0$$

$$\sigma = \frac{4F}{\pi d_r^2} = -\sigma_z \qquad \sigma_y = 0 \qquad \tau_{zx} = 0$$

$$\frac{k_m}{E \cdot d} = A \cdot \exp(B \cdot d / l)$$ A and B are from Table 8-7

Dynamic loading (Fatigue loading)

$$\sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min}) = \frac{1}{2}\left(\frac{CP}{A_t} + \frac{F_i}{A_t} - \frac{F_i}{A_t}\right) = \frac{CP}{2A_t}$$

$$\sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min}) = \frac{1}{2}\left(\frac{CP}{A_t} + \frac{F_i}{A_t} + \frac{F_i}{A_t}\right) = \frac{CP}{2A_t} + \frac{F_i}{A_t}$$

Using Goodman criterion, $\frac{S_a}{S_e} + \frac{S_m}{S_{ut}} = 1$ and solving for n_f we can obtain

$$n_f = \frac{S_{ut} A_t - F_i}{(CP/2)(1 + S_{ut}/S_e)}$$

Shear Joints

$$M = F_A'' \cdot r_A + F_B'' \cdot r_B + F_C'' \cdot r_C + \cdots$$

$$\frac{F_A''}{r_A} = \frac{F_B''}{r_B} = \frac{F_C''}{r_C}$$

$$F_n'' = \frac{M r_n}{r_A^2 + r_B^2 + r_C^2 + \cdots}$$

Pins

$$\left.\begin{array}{ll} \sigma_x = -p & \tau_{xy} = 0 \\ \sigma_y = -p & \tau_{yz} = 0 \\ \sigma_z = 0 & \tau_{xz} = 0 \end{array}\right\} \text{use von Mises}$$

$$\sigma' = (p^2 + 3\tau_{xz}^2)^{1/2} = \frac{S_y}{n_d} = \frac{\sqrt{3} S_{sy}}{n_d}$$

$$(\tau_{xz})_{all} = \frac{1}{\sqrt{3}}\sqrt{3\left(\frac{S_{sy}}{n_d}\right)^2 - p^2} = \frac{1}{\sqrt{3}}\sqrt{\left(\frac{S_y}{n_d}\right)^2 - p^2}$$

Tension connections - the fastener

- Twisting the nut stretches the bolt to produce the clamping force. This clamping force is called the pre-tension or bolt preload.
- The clamping force produces tension in the bolt and compression in the members.
- A stud is a road threaded on both ends. The stud is screwed into the lower member first; then the top member is positioned and fastened down with hardened washers and nuts.

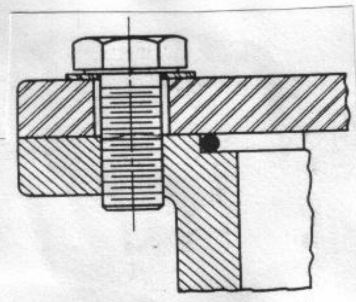


FIGURE 8-13

Section of a cylindrical pressure vessel. Hexagon-head cap screws are used to fasten the cylinder head to the body. Note the use of the O-ring seal.

- The spring constant, or stiffness constant of a bolt

$$\frac{\text{force}}{\text{deflection}} \Rightarrow \frac{AE}{L}$$

tensile stress area; major diameter area

$$k_T = \frac{A_t E}{L_T} \qquad k_d = \frac{A_d E}{L_d} \quad \xrightarrow{\text{two springs}} \implies \underset{\text{in series}}{\implies} \quad k_b = \frac{A_d A_t E}{A_d L_t + A_t L_d}$$

L_T: length of threaded portion

L_d: Length of unthreaded portion

k_b: effective stiffness of the bolt

Tension connections - the members

$$k_m = \frac{\pi E d \tan\alpha}{2\ln\dfrac{(L\tan\alpha + d_w - d)(d_w + d)}{(L\tan\alpha + d_w + d)(d_w - d)}}$$

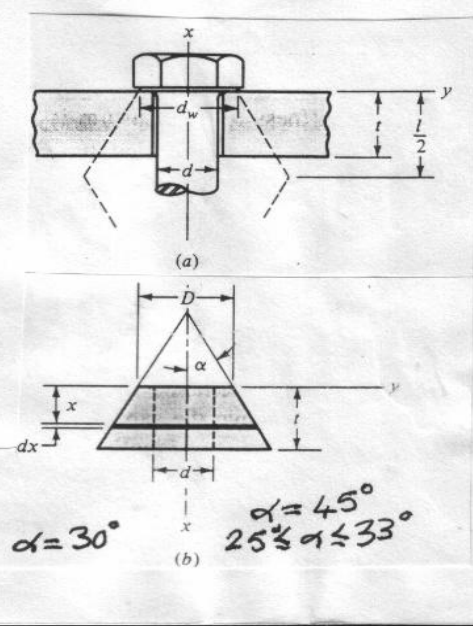


FIGURE 8-14

Compression of a member assumed to be confined to the frustum of a hollow cone.

$\alpha = 30°$ $\quad$ $\alpha = 45°$ $\quad$ $25 \le \alpha \le 33°$

Bolt strength

- Minimum proof strength
- Minimum tensile strength

(Tables 8-4, 8-5, 8-6; pages 341, 342, 343)

Tension connections- the external load

- The tension load P (Fig. 8-12) causes the connection to stretch.

$$\delta = \frac{P_b}{k_b} = \frac{P_m}{k_m} \Rightarrow P_b = P_m \frac{k_b}{k_m}$$

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

$$\underbrace{F_b}_{\text{resultant bolt load}} = \underbrace{P_b}_{\text{portion of } P \text{ taken by bolt}} + \underbrace{F_i}_{\text{preload}} = \frac{k_b P}{k_b + k_m} + F_i \qquad F_m < 0$$

$$\underbrace{F_m}_{\text{resultant load on members}} = \underbrace{P_m}_{\text{portion of } P \text{ taken by members}} - F_i = \frac{k_m P}{k_b + k_m} - F_i \qquad F_m < 0$$

- The members take over 80% of the external load

$$C = \frac{k_b}{k_b + k_m} \cong \begin{cases} 0.168 \\ 0.136 \\ 0.114 \end{cases} \quad \text{(Table 8-7, page 345)}$$

Torque requirements

- We need to estimate the wrench torque required to develop the specified preload.
- Wrenching methods
  - torque wrenching
  - pneumatic-impact wrenching
  - the turn-of-the-nut method (requires to define the meaning of snug-tight)

  "snug-tight condition is the tightness attained by a few impacts of an impact wrench, or the full effort of a person using an ordinary wrench"

  "all additional turning develops useful tension in the bolt"

  "turn-of-the-nut method requires to compute the fractional number of turns necessary to develop the required preload from the snug-tight condition"

- An expression for the torque required

$$T = \underbrace{\left[\left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda + \mu\sec\alpha}{1-\mu\tan\lambda\sec\alpha}\right) + 0.625\,\mu_c\right]}_{K} F_i d$$

$K \cong 0.20$ (look at Table 8-10, page 347 for K values)

Bolt preload - static loading

- $F_m = (1-C)P - F_i$
- P_0 is the external load that would cause joint separation (at separation $F_m = 0$)

$n = P_0/P \quad \Rightarrow \quad n = \dfrac{F_i}{P(1-C)}$

$\hookrightarrow$ the factor of safety guarding against joint separation

- In general (no joint separation)

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

"$n > 1$ ensures that the bolt stress is less than the proof strength"

- Recommendation for both static and fatigue loading,

$$F_i = \begin{cases} 0.75\,F_p & \text{for reused connections} \\ 0.90\,F_p & \text{for permanent connections} \end{cases}$$

$F_p = A_t S_p$ (for S_p Tables 8-4 to 8-6, pages 341-343)

"for other materials, an approximate value for $S_p \Rightarrow S_p \cong 0.85\,S_y$"

Fatigue loading

- Average fatigue-strength-reduction factors (Table 8-11, page 351)
- $n = S_a/\sigma_a$

factor of safety guarding against a fatigue failure

σ_a $\hookrightarrow$ alternating stress component

$$S_a = \frac{S_{ut} - F_i/A_t}{1 + S_{ut}/S_e}$$

- Check the possibility of yielding

$$n = \frac{S_y}{\sigma_m + \sigma_a}$$

Bolted and riveted joints loaded in shear

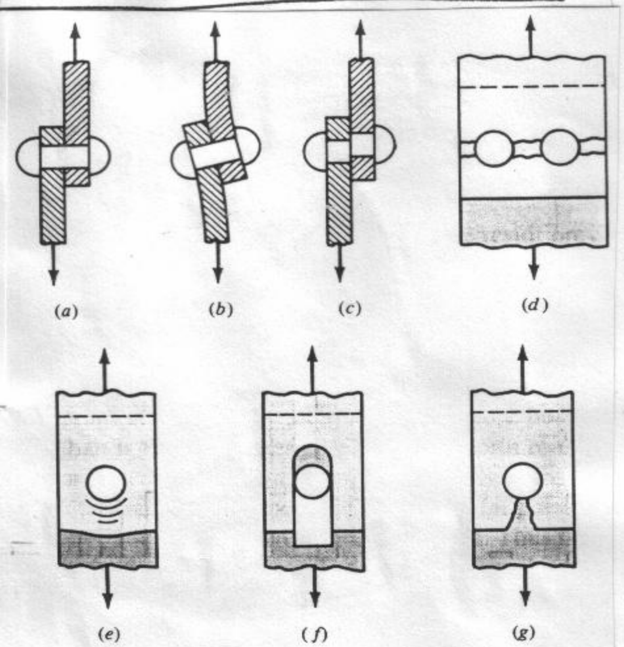


FIGURE 8-21
Modes of failure in shear loading of a bolted or riveted connection: (a) shear loading; (b) bending of rivet; (c) shear of rivet; (d) tensile failure of members; (e) bearing of rivet on members or bearing of members on rivet; (f) shear tear-out; (g) tensile tear-out.

- Failure by bending (b)

$$\sigma = \frac{M}{I/c} \qquad M = \frac{Ft}{2}$$

- Rupture of one of the (d) connected members by pure tension

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

A → net area of the plate

- Failure by pure shear (c)

$$\tau = \frac{F}{A}$$

A → cross-sectional area of all the rivets

- Crushing of the rivet or plate (e)

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

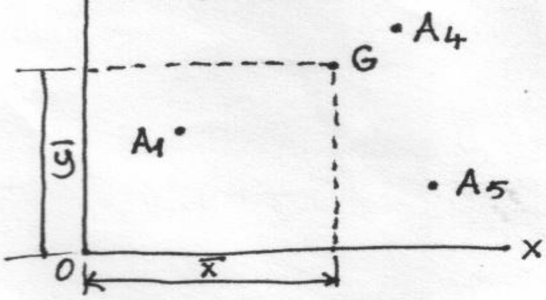
$A = td$

d → rivet or bolt diameter

t → thickness of the thinnest plate

Centroids of bolt groups

- In order to determine the shear forces which act upon each bolt, it is necessary to know the location of the centroid of the bolt group.



$$\bar{x} = \frac{\sum_{1}^{n} A_i x_i}{\sum_{1}^{n} A_i}$$

$$\bar{y} = \frac{\sum_{1}^{n} A_i y_i}{\sum_{1}^{n} A_i}$$

Setscrews

- Compression to develop a clamping force
- The resistance to axial motion or rotary motion of the collar or hub relative to the shaft is called the holding power.
- The length of a setscrew is about half of the shaft diameter.

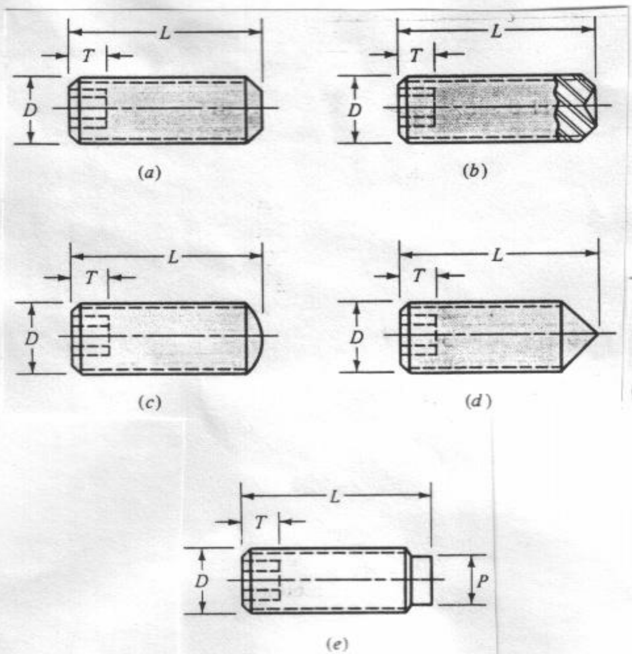


FIGURE 8-26

Socket setscrews: (a) flat point; (b) cup point; (c) oval point; (d) cone point; (e) half-dog point.

Keys and pins

- Keys and pins are used on shafts to secure rotating elements, such as gears, pulleys, or other wheels.
  - Keys are used to enable the transmission of torque from the shaft to the shaft-supported element.
  - Pins are used for axial positioning and for the transfer of torque or thrust or both.
- Tables 8-14 to 8-17 (pages 367 — 369) for dimensions

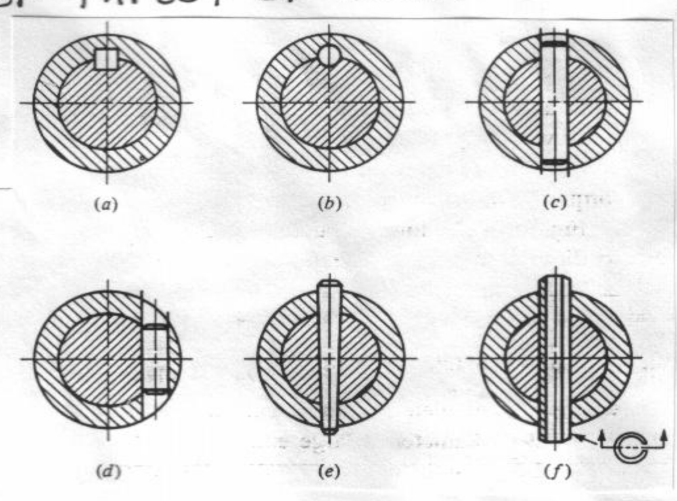


FIGURE 8-27

(a) Square key; (b) round key; (c and d) round pins; (e) taper pin; (f) split tubular spring pin. The pins in parts (e) and (f) are shown longer than necessary, to illustrate the chamfer on the ends, but their lengths should be kept smaller than the hub diameters to prevent injuries due to projections on rotating parts.