University of Technology
Mechanical Engineering Department

Machine Design I

Third Class for All Branches

LECTURES TWENTY &TWENTY ONE

FASTENERS

Reference: "Machine Elements in Mechanical Design" 4th Edition in SI units, By: Robert L. Mott, Chapter 18 & 20.

"A Textbook of Machine Design" 14th Edition in SI units, By: R. S. Khurmi & J. K. Gupta.

Introduction:

A fastener: is any device used to connect or join two or more components. Literature hundreds of fastener types and variation are available. The most common are threaded fastener referred to by many names, among them bolts, screws, nut, and studs ….etc. see figures below.

A bolt: is a threaded fastener designed to pass through holes in the mating members and to be second by tightening a nut from the end opposite threaded of the bolt, see figure (18-1- a).

A screw: is a threaded fastener designed to be inserted through a hole in one member and to be joined and in to a threaded hole in the mating member, see figure (18-1- b).

FIGURE 18–1
Comparison of a bolt with a screw (R. P. Hoelscher et al., *Graphics for Engineers*, New York: John Wiley & Sons, 1968)

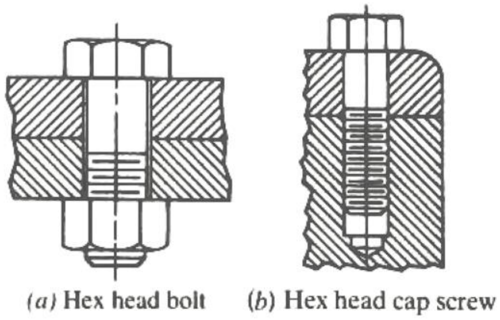


FIGURE 18–2 Bolt styles. See also the hex head bolt in Figure 18–1. (R. P Hoelscher et al., *Graphics for Engineers*, New York: John Wiley & Sons, 1968)

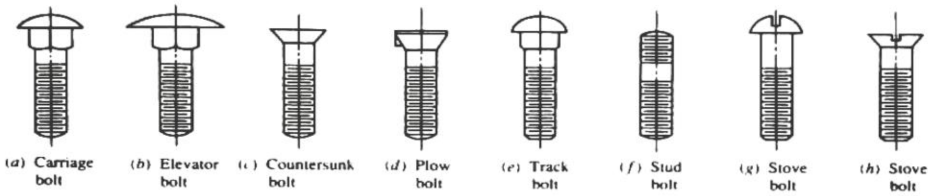


FIGURE 18–3 Cap screws or machine screws. See also the hex head cap screw in Figure 18–1. (R. P. Hoelscher et al., *Graphics for Engineers*, New York: John Wiley & Sons, 1968)

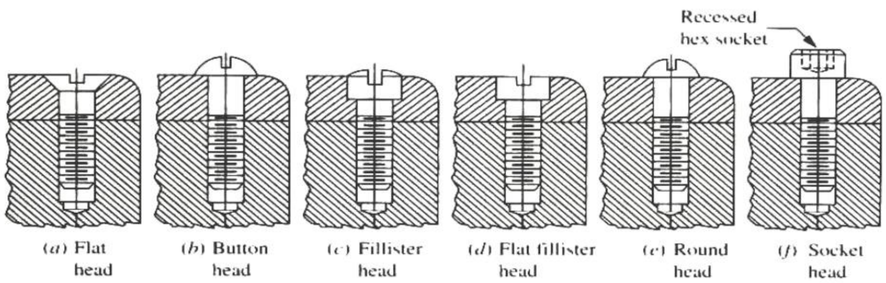


FIGURE 18–4
Sheet-metal and lag screws (R. P. Hoelscher et al., *Graphics for Engineers*, New York: John Wiley & Sons, 1968)

Bolt Materials and Strength (Section 18-2, Page 714)

Table 18-1 SAE grades of steels for fasteners

Grade No.	Bolt size		Tensile Strength		Yield Strength		Proof Strength		Head marking
	(in)	(mm)	(Ksi)	(GPa)	(Ksi)	(GPa)	(Ksi)	(GPa)	
1	$\frac{1}{4}$ - $1\frac{1}{2}$	6.3-38.1	60	0.41	36	0.25	33	0.23	None
2	$\frac{1}{4}$ - $\frac{3}{4}$	6.3-19.1	74	0.51	57	0.39	55	0.38	None
	$> \frac{3}{4}$ - $1\frac{1}{2}$	> 19.1-39.1	60	0.41	36	0.25	33	0.23	
4	$\frac{1}{4}$ - $1\frac{1}{2}$	6.3-38.1	115	0.79	100	0.69	65	0.45	None
5	$\frac{1}{4}$ - 1	6.3-25.4	120	0.83	92	0.63	85	0.59	
	> 1 - $1\frac{1}{2}$	>25.4-38.1	105	0.72	81	0.56	74	0.51	
7	$\frac{1}{4}$ - $1\frac{1}{2}$	6.3-38.1	133	0.92	115	0.79	105	0.72	
8	$\frac{1}{4}$ - $1\frac{1}{2}$	6.3-38.1	150	1.03	130	0.5	120	0.83	

Notes:

1. In machine design, most fasteners are made from steel because of its high strength, high stiffness, good ductility, and good machinability and formability.
2. Aluminum, Brass, Copper, bronze, Nickel and its alloys, plastic and stainless steel are also used for fasteners.
3. Proof strength is similar to elastic limit $\cong (\mathbf{0.9 - 0.95})\boldsymbol{S_y}$

Thread designations and stress area (Section 18.3, Page 717)

See table (18-4) for American standard thread dimensions. Below is table (18-5) for metric thread dimensions for coarse and fine thread (Page 718).

TABLE 18–4 American Standard thread dimensions

A. Numbered sizes

Size	Basic major diameter (in)	Coarse threads: UNC Threads per in	Coarse threads: UNC Tensile stress area (in^2)	Fine threads: UNF Threads per in	Fine threads: UNF Tensile stress area (in^2)
0	0.0600			80	0.001 80
1	0.0730	64	0.00263	72	0.002 78
2	0.0860	56	0.00370	64	0.003 94
3	0.0990	48	0.00487	56	0.005 23
4	0.1120	40	0.00604	48	0.006 61
5	0.1250	40	0.00796	44	0.008 30
6	0.1380	32	0.00909	40	0.010 15
8	0.1640	32	0.0140	36	0.014 74
10	0.1900	24	0.0175	32	0.0200
12	0.2160	24	0.0242	28	0.0258

B. Fractional sizes

Size	Basic major diameter (in)	Coarse threads: UNC Threads per in	Coarse threads: UNC Tensile stress area (in^2)	Fine threads: UNF Threads per in	Fine threads: UNF Tensile stress area (in^2)
1/4	0.2500	20	0.0318	28	0.0364
5/16	0.3125	18	0.0524	24	0.0580
3/8	0.3750	16	0.0775	24	0.0878
7/16	0.4375	14	0.1063	20	0.1187
1/2	0.5000	13	0.1419	20	0.1599
9/16	0.5625	12	0.182	18	0.203
5/8	0.6250	11	0.226	18	0.256
3/4	0.7500	10	0.334	16	0.373
7/8	0.8750	9	0.462	14	0.509
1	1.000	8	0.606	12	0.663
$1\frac{1}{8}$	1.125	7	0.763	12	0.856
$1\frac{1}{4}$	1.250	7	0.969	12	1.073
$1\frac{3}{8}$	1.375	6	1.155	12	1.315
$1\frac{1}{2}$	1.500	6	1.405	12	1.581
$1\frac{3}{4}$	1.750	5	1.90		
2	2.000	$4\frac{1}{2}$	2.50		

American standard designation

10-24 UNC ; 1/2-13 UNC 10-32 UNF ; 1/2-20 UNF

$10 \rightarrow size$; $24 \rightarrow No. of\ thread\ per\ inch$

$10 \rightarrow size$; $32 \rightarrow No. of\ thread\ per\ inch$

UNC : Coarse thread ; UNF: Fine thread

TABLE 18–5 Metric thread dimensions

Basic major diameter (mm)	Coarse threads Pitch (mm)	Coarse threads Tensile stress area (mm^2)	Fine threads Pitch (mm)	Fine threads Tensile stress area (mm^2)
1	0.25	0.460		
1.6	0.35	1.27	0.20	1.57
2	0.4	2.07	0.25	2.45
2.5	0.45	3.39	0.35	3.70
3	0.5	5.03	0.35	5.61
4	0.7	8.78	0.5	9.79
5	0.8	14.2	0.5	16.1
6	1	20.1	0.75	22.0
8	1.25	36.6	1	39.2
10	1.5	58.0	1.25	61.2
12	1.75	84.3	1.25	92.1
16	2	157	1.5	167
20	2.5	245	1.5	272
24	3	353	2	384
30	3.5	561	2	621
36	4	817	3	865
42	4.5	1121		
48	5	1473		

Matric designation

M3*0.5 M3*0.35

M→ $Metric$ $3 \rightarrow basic\ major\ diameter$ $0.5\ or\ 0.35 \rightarrow Pitch\ in\ mm$

Clamping Load (Section 18.4, Page 719)

$$P = \frac{F}{n} \qquad \ldots\ldots\ldots(1)$$

$$\sigma_a = K\,[\sigma] \qquad \ldots\ldots\ldots(2)$$

$$A_t = \frac{P}{\sigma_a} \qquad \ldots\ldots\ldots(3)$$

$$T = K_1 * D * P \qquad \ldots\ldots\ldots(4)$$

Where:

P : Clamping load on one bolt

F : Overall clamping load on bolts

n : No. of bolts

$\boldsymbol{\sigma_a}$: Allowable stress

$[\boldsymbol{\sigma}]$: Proof strength

$\mathbf{A_t}$ **:** Required tensile stress area

T: Required tightening torque

D : Nominal outside diameter of threads

K = 0.75 { The clamping load is often taken to be 0.75 times the proof load. Where: the proof load is = $[\boldsymbol{\sigma}] * \text{tensile stress area } (A_t)$} . Also this factor called demand factor in MDesign.

K_1 = Constant depends on lubricant present

= 0.15 Lubricant at all is present

= 0.2 If thread well cleaned and dried.

<u>Example (18-1) Page 719</u>

A set of three bolts is to be used to provide a clamping force of (12000 Ib) 53370 N between two components of a machine. The load is shared equally among the three bolts. Specify suitable bolts, including the grade of the material, if each is to be stressed to *75%* of its proof strength. Then compute the required tightening torque.

Solution:

Choose SAE grade from table (18-1) say grade no.5

$[\boldsymbol{\sigma}] = 85000\ Psi = 590\ MPa$

P = clamping load = $\frac{F}{n} = \frac{53370}{3} = 17.79\ KN$

$\sigma_a = K\ [\sigma]\ = 0.75 * 586 = 439.6\ MPa$

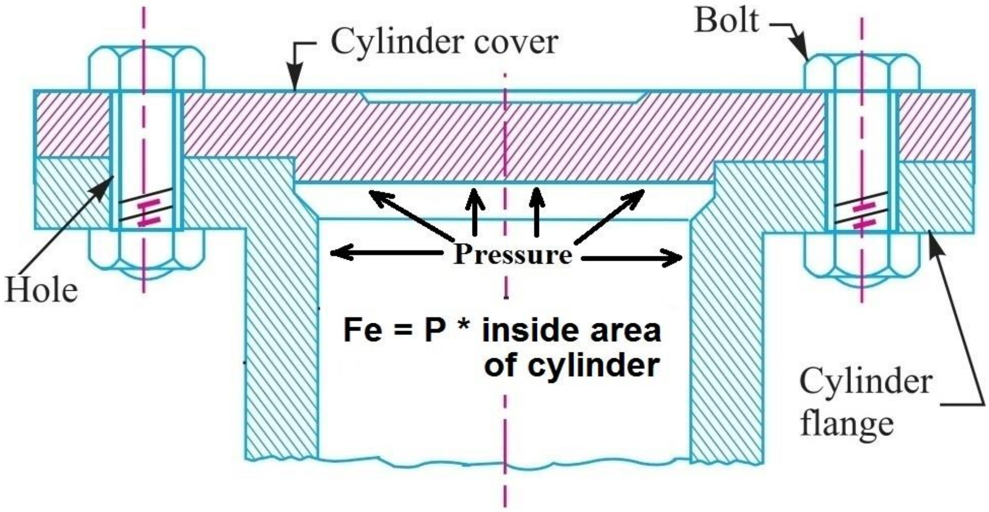
$$A_t = \frac{P}{\sigma_a} = \frac{17.79 * 10^3 N}{439.6\ MPa} = 40.45\ mm^2$$

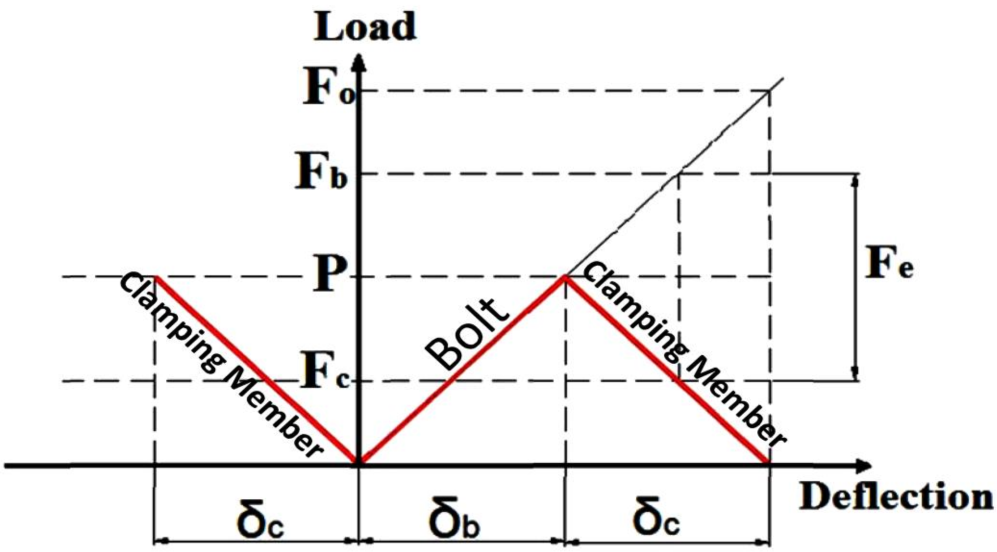
From table (18-5) choose M10*1.5 (A_t = 58 mm^2)

$T = K_t * D * P\ = 0.15 * 10 * 10^{-3} * 17.79 * 10^3 = 26.7\ N.m$

Externally Applied force on a bolted joint (Section 18.5, Page 722)

The above previous example (18-1) is for bolts under static conditions, now if there is an external load on the bolts (Fe) as shown below, when the cover of pressure vessels is fixed by bolts.





$$F_b = P + \frac{K_b}{K_b + K_c} F_e \qquad \dots\dots\dots (18-8)$$

$$F_c = P - \frac{K_c}{K_b + K_c} F_e \qquad \dots\dots\dots (18-9)$$

For more than two clamping members:

$$\frac{1}{K_c} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \quad \dots\dots\dots\dots$$

Where:

F_e : externally applied load
P : initial clamping load
F_b : Final force on bolt
F_c : Final force on clamping member
K_b : Stiffness of bolt = $\frac{P}{\delta_b}$
K_c : Stiffness of clamping member = $\frac{P}{\delta_c}$
F_o : Load to open the connection.

<u>Example (18-2) Page 722</u>

Assume that the joint described in Example Problem 18-1 was subjected to an additional external load of 3000 Ib (13.344 KN) after the initial clamping load of 4000 lb (17.792 KN) was applied. Also assume that the stiffness of the clamped members is three times that of the bolt. Compute the force in the bolt, the force in the clamped members, and the final stress in the bolt after the external load is applied.

Solution: $\boldsymbol{K_c = 3\, K_b}$

$$\mathrm{F_b} = \mathrm{P} + \frac{\mathrm{K_b}}{\mathrm{K_b} + \mathrm{K_c}} \mathrm{F_e} = \mathrm{P} + \frac{\mathrm{K_b}}{4\mathrm{K_b}} \mathrm{F_e} = \mathrm{P} + \frac{1}{4} \mathrm{F_e} = 17.792 + \frac{13.344}{4} = 21.128 \text{ KN}$$

$$\mathrm{F_c} = \mathrm{P} - \frac{\mathrm{K_c}}{\mathrm{K_b} + \mathrm{K_c}} \mathrm{F_e} = \mathrm{P} - \frac{3}{4} \mathrm{F_e} = 17.79 - \frac{3}{4}(13.34) = 7.784 \text{ KN}$$

$Fc > 0 \rightarrow$ the joint is still $\rightarrow$ $tight$

Now for M10*1.5 ; $A_t = 58\ mm^2$

$$A_t = \frac{P}{\sigma_a} \rightarrow \sigma_a = \frac{P}{A_t} = \frac{21128}{58} = 364.3\ MPa$$

$$now\ [\sigma] = proof\ stress = 590\ MPa$$

$$\text{Demand factor} = \frac{\sigma_a}{[\sigma]} = \frac{364.3}{590} = 61.7\% < 75\% \quad \{\text{the select bolt is still safe}\}$$

Example (18-3) Page 723

Solve Example Problem 18-2 again, but assume that the joint has a flexible elastomeric gasket separating the clamping members and that the stiffness of the bolt is then 10 times that of the joint.

Solution: $\boldsymbol{K_b = 10\, K_c}$

$$F_b = P + \frac{K_b}{K_b + K_c} F_e = P + \frac{10}{11} F_e = 17.79 + \frac{10}{11} * 13.34 = 29.922 \text{ KN}$$

$$F_c = P - \frac{K_c}{K_b + K_c} F_e = P - \frac{1}{11} F_e = 17.79 - \frac{1}{11} * 13.34 = 16.578 \text{ KN}$$

$$\sigma = \frac{29.922 \text{ KN} * 1000}{58 \text{ mm}^2} = 517.1 \text{ MPa}$$

Demand factor = $\frac{517.1}{590} = 87.6\% \quad > \quad 75\%$

{The selected bolt is dangerous close to proof strength and yield strength}.