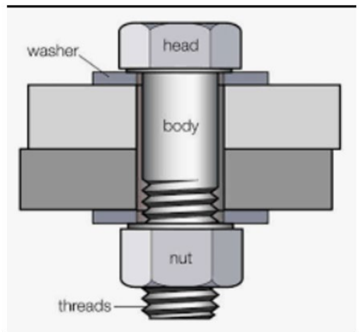
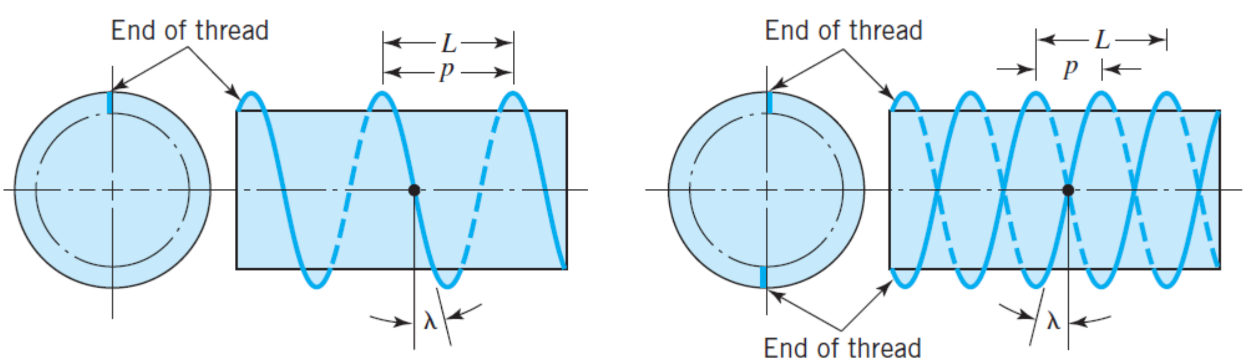
Lesson 03- Fasteners

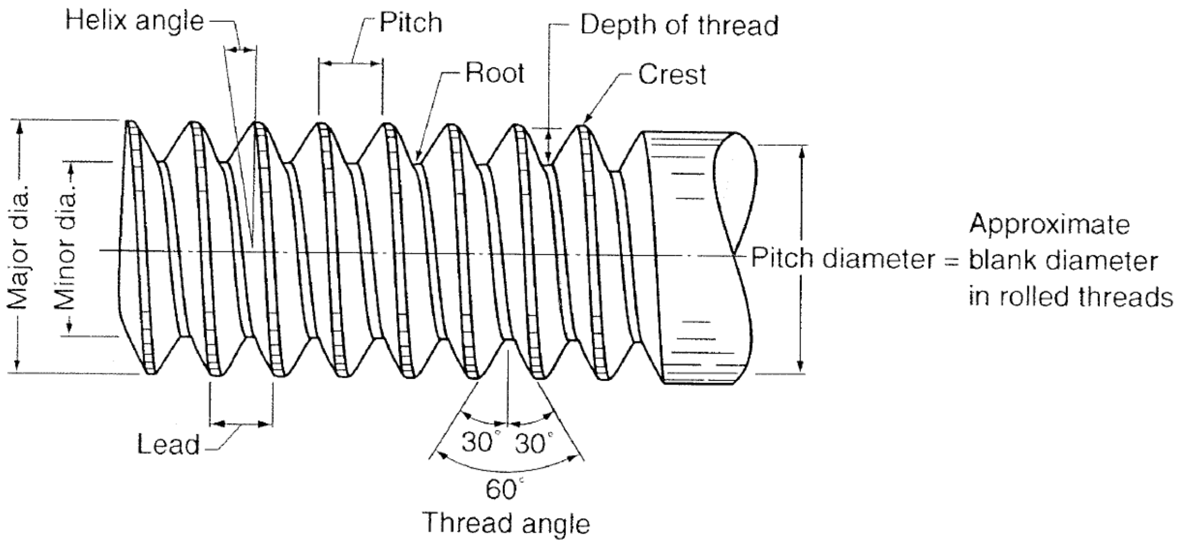


- Threads are the basic arrangement of a helical thread wound around a cylinder,
- Threads could be defined by : Pitch, lead, lead angle, and hand-of-thread
- In general all bolts and screws have a single thread, however power screws sometimes have double, triple, and even quadruple threads.
- Unless otherwise noted, all threads are assumed to be right-hand.



(a)
Single thread–right hand

(b)
Double thread–left hand



Major (nominal) diameter: This is the largest diameter of a screw thread, touching the crests on an external thread or the roots of an internal thread.

Minor (core) diameter: This is the smallest diameter of a screw thread, touching the roots or core of an external thread (root or core diameter) or the crests of an internal thread.

Pitch diameter: This is the diameter of an imaginary cylinder, passing through the threads at the points where the thread width is equal to the space between the threads.

Pitch: It is the distance measured parallel to the axis, between corresponding points on adjacent screw threads.

Lead: It is the distance a screw advances axially in one turn.

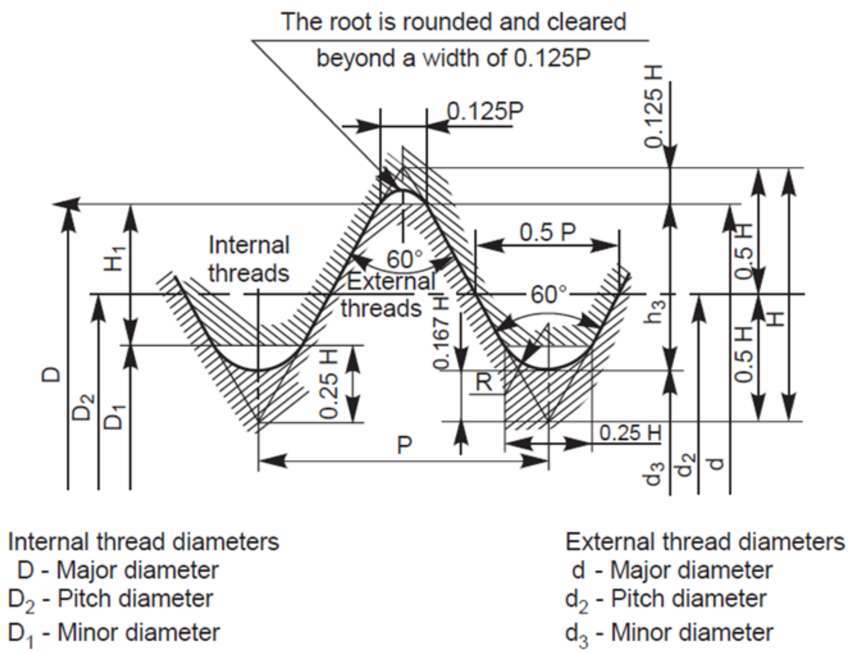
Crest: It is the peak edge of a screw thread, that connects the adjacent flanks at the top.

Root: It is the bottom edge of the thread that connects the adjacent flanks at the bottom.

Thread angle: This is the angle included between the flanks of the thread, measured in an axial plane.

Forms of thread:

The ISO (International Organization for Standards) threads adapted the metric threads.



Where:

P = Pitch

$H = 0.86\,P$

$D = d$ = Major diameter

$D_2 = d_2 = d - 0.75H$

$D_1 = d_2 - 2(H/2 - H/4) = d - 2H_1$

$d_3 = d_2 - 2\,(H/2 - H/6)$

$\quad = d - 1.22P$

$H_1 = (D - D_1)/2 = 5H/8 = 0.54P$

$h_3 = (d - d_3)/2 = 17/24H = 0.61P$

$R = H/6 = 0.14P$

Other types of threads:

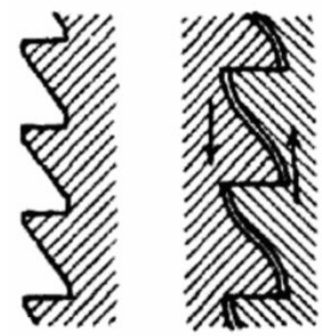
The V thread (sharp):

- This thread profile has a larger contact area, providing more frictional resistance to motion
- Hence, it is used where effective positioning is required. It is also used in brass pipe work.

The British Standard Thread:

- This thread form is adopted in Britain in inch units.
- The profile has rounded ends, making it less liable to damage than sharp V-thread.

The Buttress thread:



- This thread is a combination of V-and square threads.
- It exhibits the advantages of square thread, the ability to transmit power and low frictional resistance,
- It is used where power transmission takes place in one direction only such as screw press,
- Quick acting carpenter's vice, etc

The square thread:

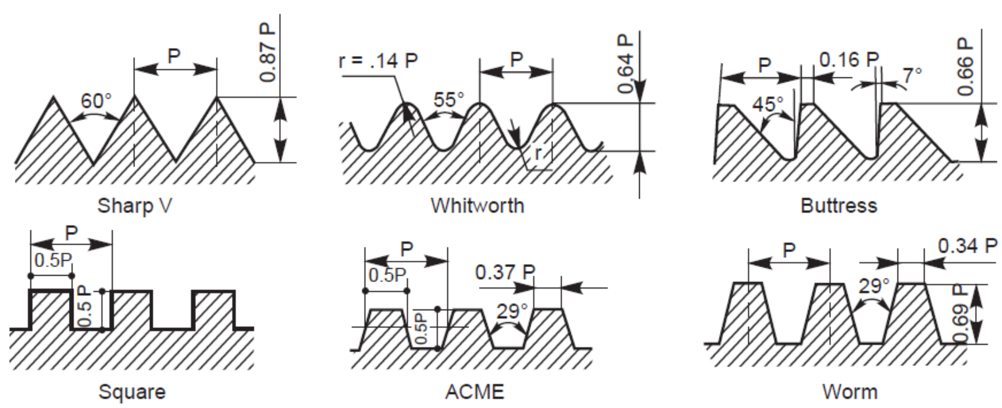
- Square thread is an ideal thread form for power transmission.
- As the thread flank is at right angle to the axis, the normal force between the threads acts parallel to the axis, with zero radial component.
- This enables the nut to transmit very high pressures, as in the case of a screw jack and other similar applications.

Acme thread:

- It is a modified form of square thread. It is much stronger than square thread because of the wider base and it is easy to cut.
- The inclined sides of the thread facilitate quick and easy engagement and disengagement as for example, the split nut with the lead screw of a lathe.

Worm thread:

- Worm thread is like the ACME thread but is deeper. It is used on shafts to carry power to worm wheels.



Thread series:

The Table below gives the nominal diameter and pitch combinations for coarse and fine series of ISO metric screw threads

Nominal diameter		Pitch			
First choice	Second choice	Coarse	Fine		
			1	2	3
2	—	0.4	0.25	—	
—	2.2	0.45	0.25	—	—
2.5	—	0.45	0.35	—	—
3	—	0.5	0.35	—	—
—	3.5	0.6	0.35	—	—
4	—	0.7	0.5	—	—
—	4.5	0.75	0.5	—	—
5	—	0.8	0.5	—	—
6	—	1	0.75	0.5	—
8	—	1.25	1	0.75	—
10	—	1.5	1.25	1	0.75
12	—	1.75	1.5	1.25	—
16	14	2	1.5	1	—
20	18,22	2.5	2	1.5	1
24	27	3	2	1.5	1
30	33	3.5	2	1.5	1
36	39	4	3	2	1.5
42	45	4.5	4	3	2
48	52	5	4	3	2
56	60	5.5	4	3	2
64	68	6	4	3	2
72	76	6	4	3	2
80	85	6	4	3	2
90	95	6	4	3	2
100	—	6	4	3	2
105 to 300	—	—	6	4	3

The diameter-pitch combination of an ISO metric screw thread is designated by the letter 'M' followed by the value of the nominal diameter and pitch, the two values being separated by the sign '×'.

Example: a diameter pitch combination of nominal diameter 10 mm and pitch 1.25 mm is designated as M10 × 1.25

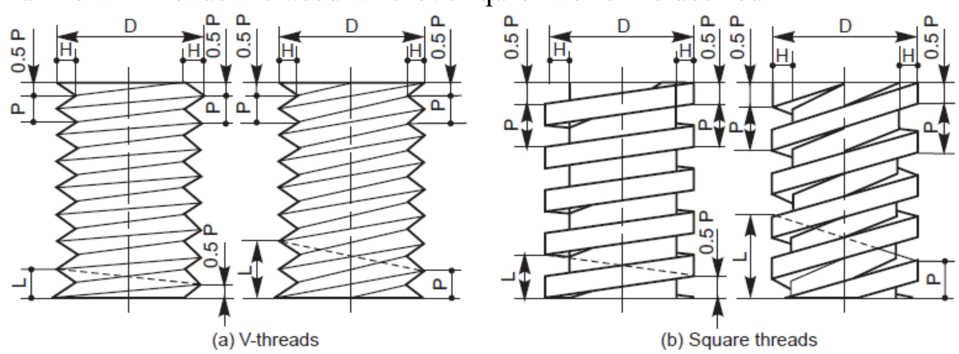
Size	Major Diameter d (in.)	Coarse Threads—UNC: Threads per Inch	Coarse Threads—UNC: Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC: Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF: Threads per Inch	Fine Threads—UNF: Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF: Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599
$\frac{9}{16}$	0.5625	12	0.4603	0.182	18	0.4943	0.203
$\frac{5}{8}$	0.6250	11	0.5135	0.226	18	0.5568	0.256
$\frac{3}{4}$	0.7500	10	0.6273	0.334	16	0.6733	0.373
$\frac{7}{8}$	0.8750	9	0.7387	0.462	14	0.7874	0.509
1	1.0000	8	0.8466	0.606	12	0.8978	0.663
$1\frac{1}{8}$	1.1250	7	0.9497	0.763	12	1.0228	0.856
$1\frac{1}{4}$	1.2500	7	1.0747	0.969	12	1.1478	1.073
$1\frac{3}{8}$	1.3750	6	1.1705	1.155	12	1.2728	1.315
$1\frac{1}{2}$	1.5000	6	1.2955	1.405	12	1.3978	1.581
$1\frac{3}{4}$	1.7500	5	1.5046	1.90			
2	2.0000	$4\frac{1}{2}$	1.7274	2.50			
$2\frac{1}{4}$	2.2500	$4\frac{1}{2}$	1.9774	3.25			
$2\frac{1}{2}$	2.5000	4	2.1933	4.00			
$2\frac{3}{4}$	2.7500	4	2.4433	4.93			
3	3.0000	4	2.6933	5.97			
$3\frac{1}{4}$	3.2500	4	2.9433	7.10			
$3\frac{1}{2}$	3.5000	4	3.1933	8.33			
$3\frac{3}{4}$	3.7500	4	3.4433	9.66			
4	4.0000	4	3.6933	11.08			

We specify standard bolting as follows:

¼-20UNC-2A

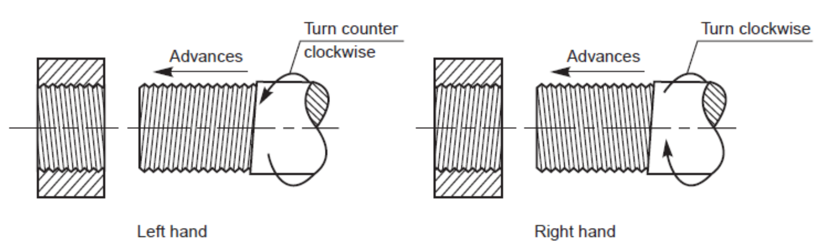
Multi Start thread:

- A single-start thread consists of a single, continuous helical groove for which the lead is equal to the pitch
- As the depth of the thread depends on the pitch, greater the lead desired, greater will be the pitch and hence smaller will be the core diameter, reducing the strength of the fastener. To overcome this drawback, multi-start threads are recommended
- Multi-start threads are used wherever quick action is desired



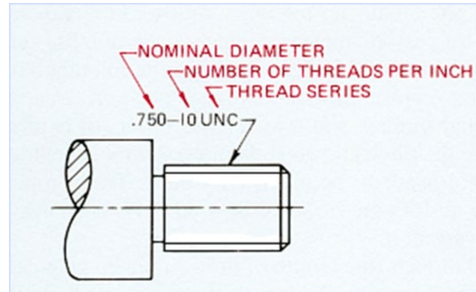
Right hand or left-hand threads:

- Screw threads may be right hand or left hand, depending on the direction of the helix.
- A right-hand thread is one which advances into the nut, when turned in a clockwise direction and a left is the opposite.
- Unless otherwise stated, a thread should be considered as a right hand one.



MENG 3050-Power Transmission Components

Inch Threads

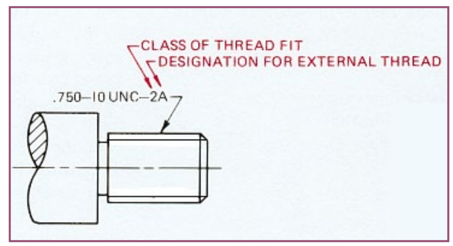


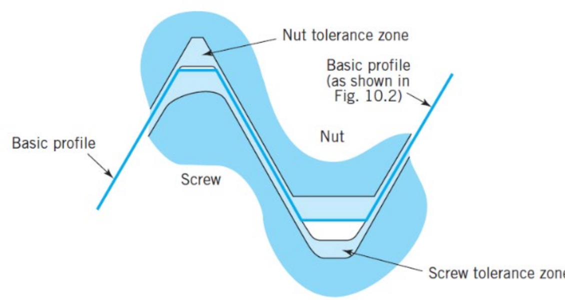
- Nominal size = screw or bolt diameter = major diameter
- Number of thread per inch
- Thread series: coarse, fine, extra fine

Thread Class:

Class of thread fit:

- 1 – Loose fit
- 2 – General commercial grade
- 3 – Close fit, for high grade commercial products



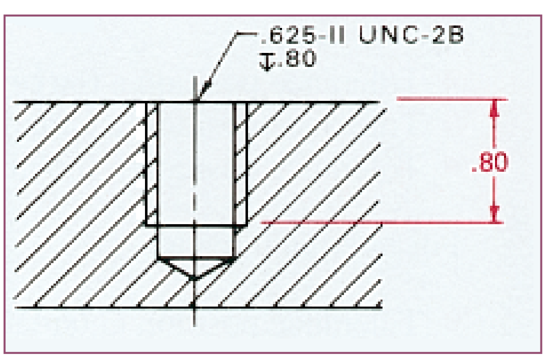


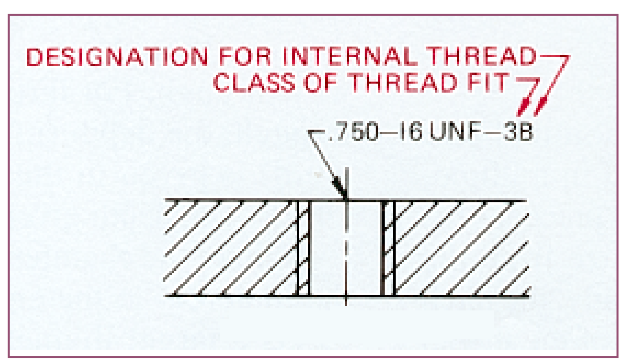
Tolerance zones for various classes of unified threads. Note: Each class—1, 2, and 3—uses a portion of the zones shown

Designation for thread:

- A – External thread
- B – Internal thread

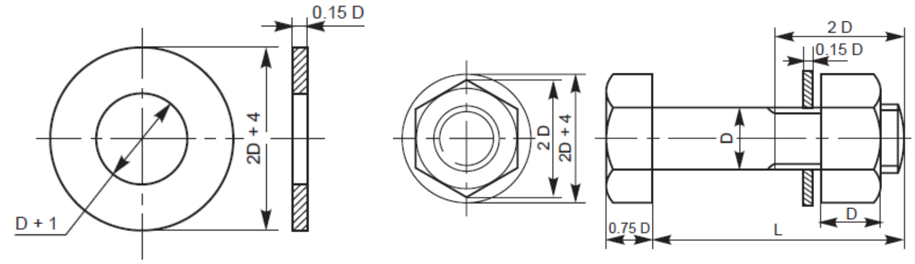
Internal thread examples.





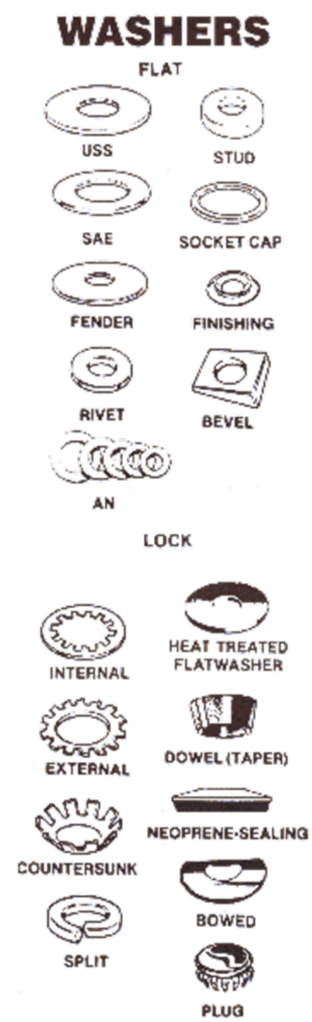
Washers:

- A washer is a cylindrical piece of metal with a hole to receive the bolt.
- It is used to give a perfect seating for the nut
- and to distribute the tightening force uniformly to the parts under the joint.
- It also prevents the nut from damaging the metal surface under the joint.
- Common washer metric sizes shown in the figure below.



Bolt Size	Flat Washer			Lockwasher			Spring Lockwasher		
	ID	OD	Thickness	ID	OD	Thickness	ID	OD	Thickness
2	2.2	5.	0.5	2.1	3.3	0.5			
3	3.2	7	0.5	3.1	5.7	0.8			
4	4.3	9	0.8	4.1	7.1	0.9	4.2	8	0.3 0.4
5	5.3	11	1	5.1	8.7	1.2	5.2	10	0.4 0.5
6	6.4	12	1.5	6.1	11.1	1.6	6.2	12.5	0.5 0.7
7	7.4	14	1.5	7.1	12.1	1.6	7.2	14	0.5 0.8
8	8.4	17	2	8.2	14.2	2	8.2	16	0.6 0.9
10	10.5	21	2.5	10.2	17.2	2.2	10.2	20	0.8 1.1
12	13	24	2.5	12.3	20.2	2.5	12.2	25	0.9 1.5
14	15	28	2.5	14.2	23.2	3	14.2	28	1 1.5
16	17	30	3	16.2	26.2	3.5	16.3	31.5	1.2 1.7
18	19	34	3	18.2	28.2	3.5	18.3	35.5	1.2 2
20	21	36	3	20.2	32.2	4	20.4	40	1.5 2.25
22	23	39	4	22.5	34.5	4	22.4	45	1.75 2.5
24	25	44	4	24.5	38.5	5			
27	28	50	4	27.5	41.5	5			
30	31	56	4	30.5	46.5	6			

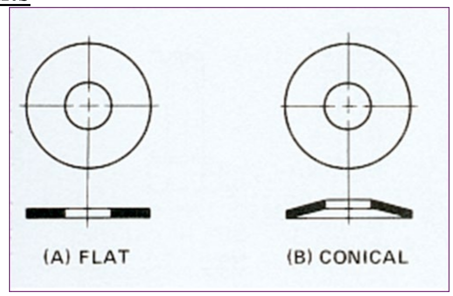
MILLIMETRE SIZES

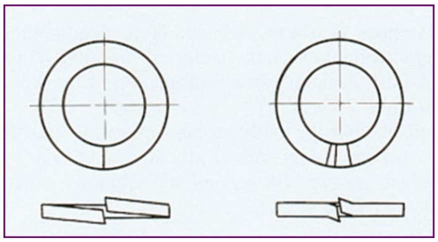


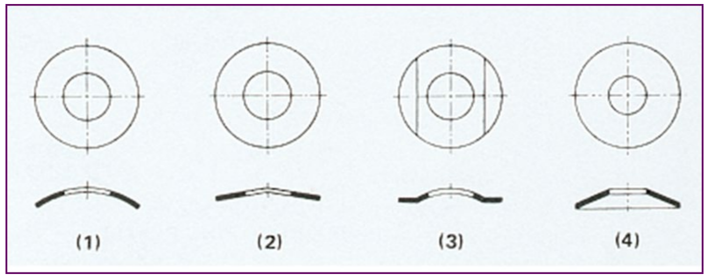
MENG 3050-Power Transmission Components

Classification of Washers:

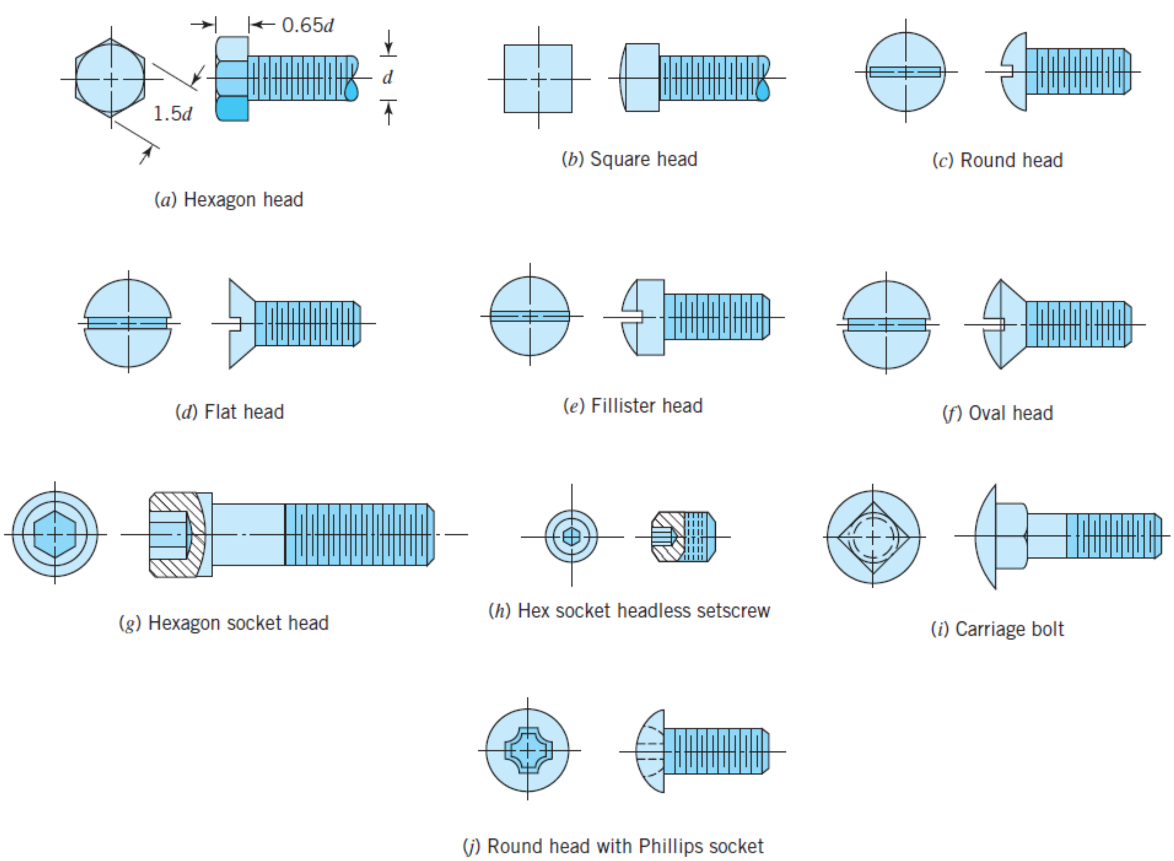
- Flat washer
- Conical washer
- Helical Spring washer
- Spring washer
- Tooth Lock washer

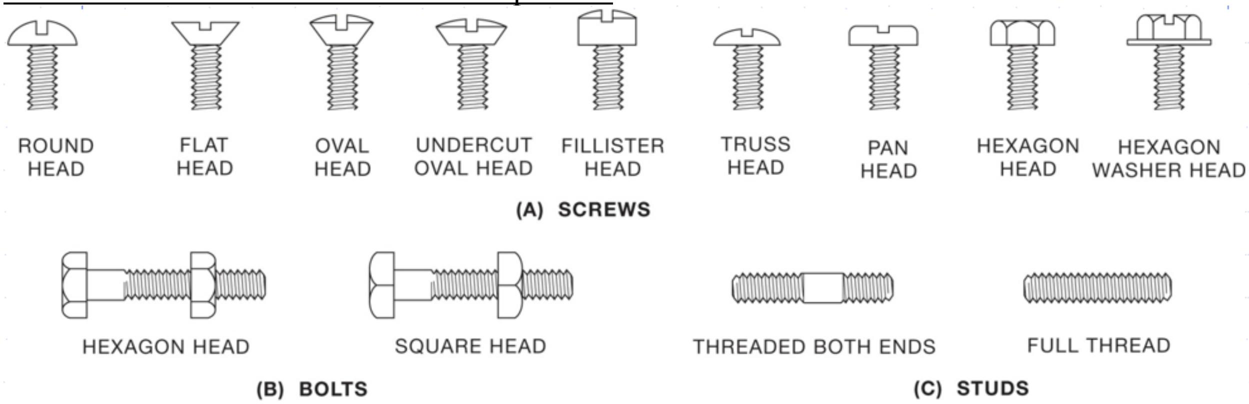


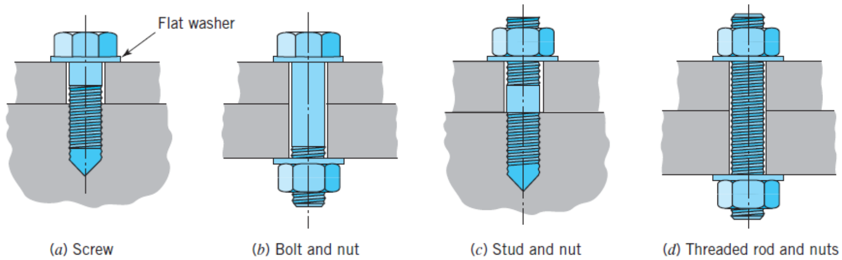




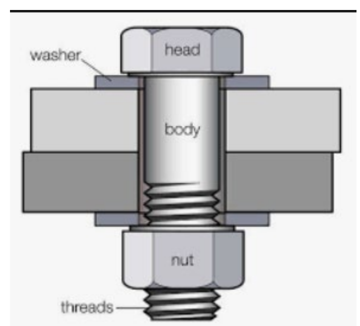
Threaded Fasteners





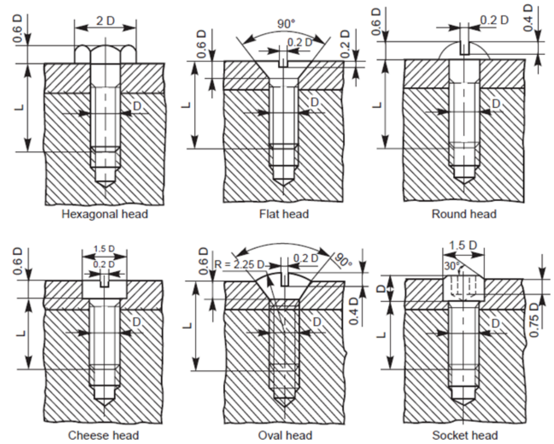


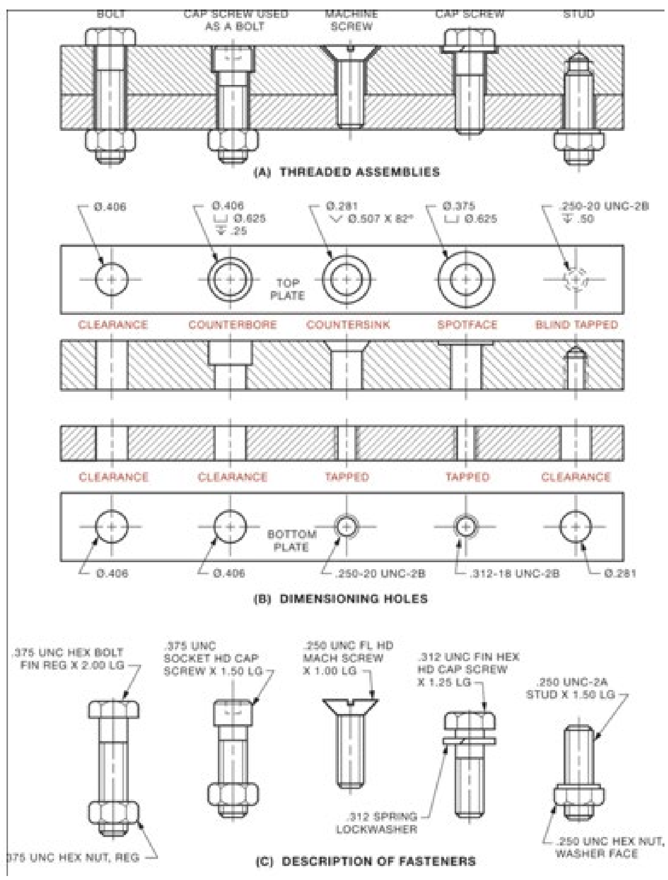
Cap screws



- Used for fastening two parts,
- The clearance of the unthreaded hole need not be shown on the drawing as its presence is obvious.

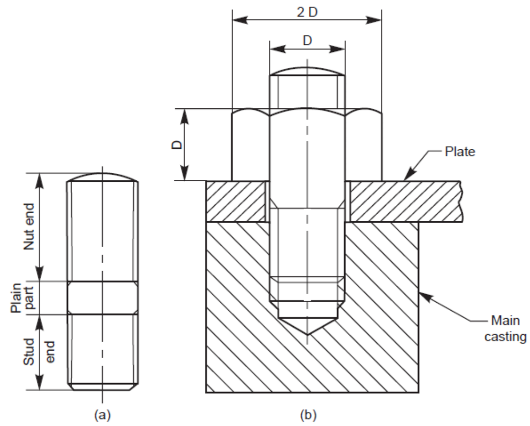
- Cap screws are:
  - Produced in finish form and are used on machines where accuracy and appearance are important.
  - They are used only on machines requiring few adjustments and are not suitable where frequent removal is necessary.
  - These are produced in different diameters, up to a maximum of 100 mm and lengths 250 mm.



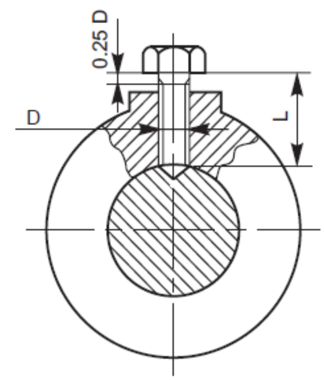


<u>Stud screw:</u>

- It consists of cylindrical shank with threads cut on both the ends.
- It is used where there is no place for accommodating the bolt head
- or when one of the parts to be joined is too thick to use an ordinary bolt.

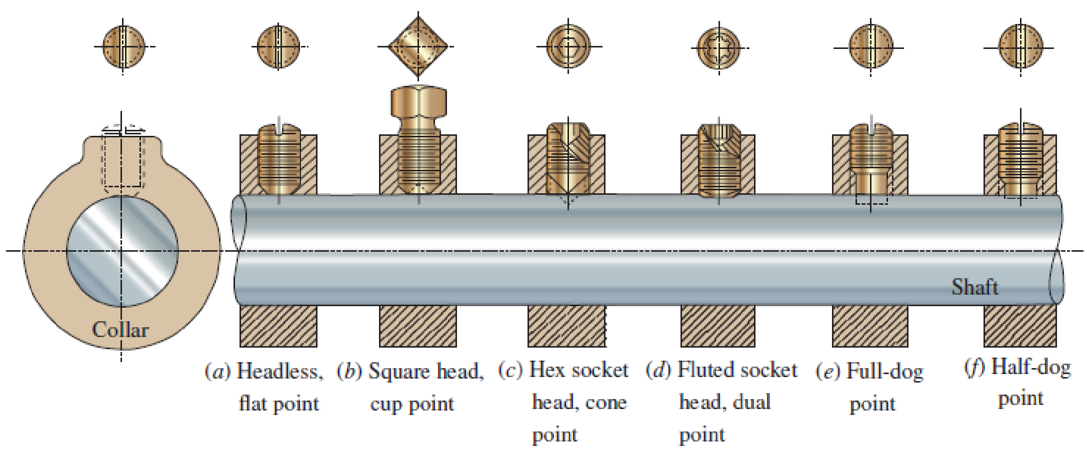


(*a*)–Stud, (*b*)–Stud joint

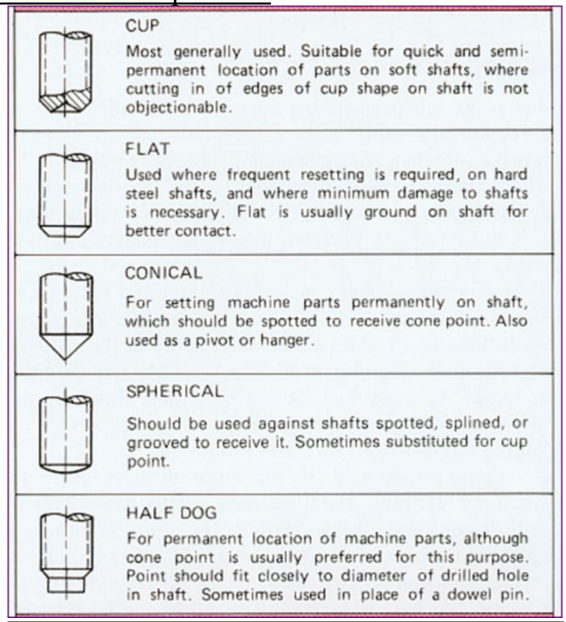


Set Screws:

- These are used to prevent relative motion between two rotating parts, such as the movement of pulley on shaft.
- a set screw is screwed into the pulley hub so that its end-point bears firmly against the shaft.
- Set screws are not efficient and so are used only for transmitting very light loads.
- For longer life, set screws are made of steel and case hardened.
- for better results, the shaft surface is suitably machined for providing more grip, eliminating any slipping tendency.

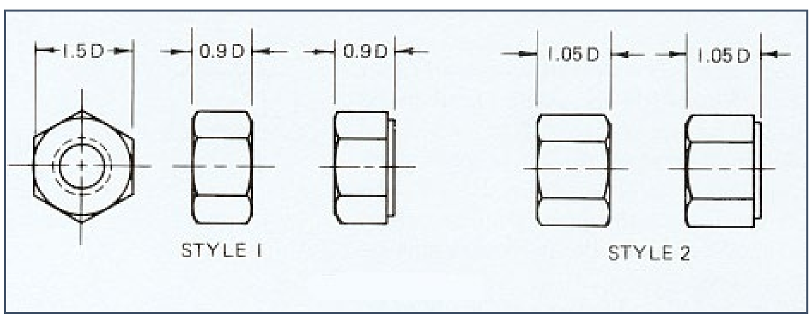


Set screws with different head and point styles applied to hold a collar on a shaft



	CUP Most generally used. Suitable for quick and semi-permanent location of parts on soft shafts, where cutting in of edges of cup shape on shaft is not objectionable.
	FLAT Used where frequent resetting is required, on hard steel shafts, and where minimum damage to shafts is necessary. Flat is usually ground on shaft for better contact.
	CONICAL For setting machine parts permanently on shaft, which should be spotted to receive cone point. Also used as a pivot or hanger.
	SPHERICAL Should be used against shafts spotted, splined, or grooved to receive it. Sometimes substituted for cup point.
	HALF DOG For permanent location of machine parts, although cone point is usually preferred for this purpose. Point should fit closely to diameter of drilled hole in shaft. Sometimes used in place of a dowel pin.

Hex Nut:



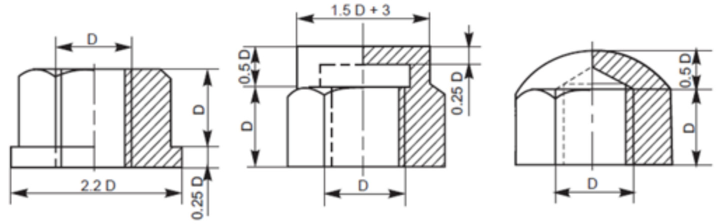
Other types of nuts:

Flanged nut:

This is a hexagonal nut with a collar or flange, provided integral with it. This permits the use of a bolt in a comparatively large size hole

Cap Nut:

It is a hexagonal nut with a cylindrical cap at the top. This design protects the end of the bolt from corrosion and also prevents leakage through the threads. Cap nuts are used in smoke boxes or locomotive and steam pipe connections.



a - Flanged nut b - Cap nut c - Dome nut

Dom Nut:
It is another form of a cap nut, having a spherical dome at the top.

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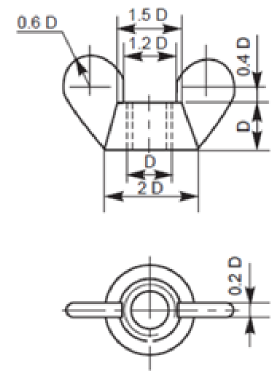
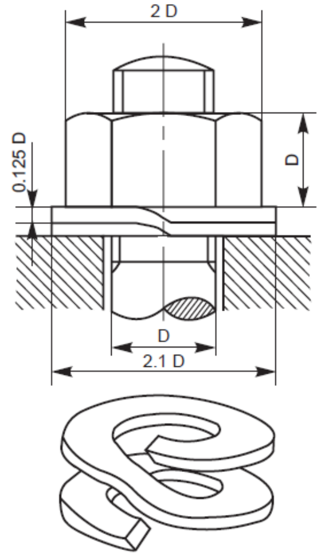
Wing Nut:
This nut is used when frequent removal is required, such as inspection covers, lids, etc. It is operated by the thumb,



Locking arrangements for nuts:

- The bolted joints are required to stay firm without becoming loose, of their own accord.
- However, the joints used in the moving parts of a machinery may be subjected to vibrations. This may slacken the joint, leading to serious breakdown.
- To eliminate the slackening tendency, different arrangements are used to lock the nuts

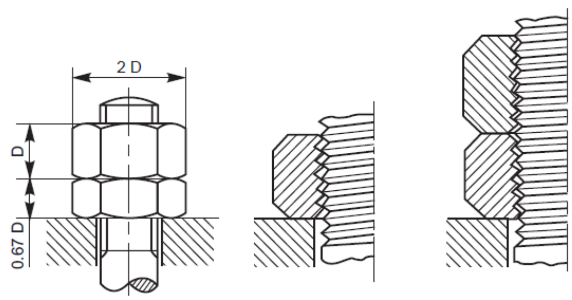
Locking by a spring washer



In this arrangement, a spring washer of either single or double coil is placed under the nut and tightened.
The spring force of the washer will be acting upwards on the nut.
This force makes the threads in the nut jammed on the bolt threads; thus, preventing the nut from loosening

Locking Nut:

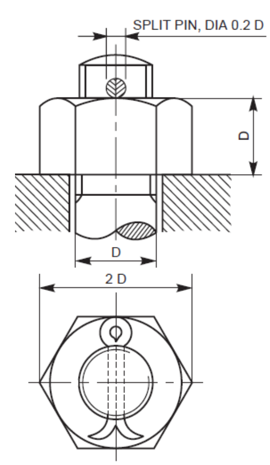
- It is the most commonly used locking device.
- In this arrangement, a second nut, known as lock nut, is used in combination with a standard nut .
- The thickness of a lock nut is usually two-thirds D, where D is the major diameter of the bolt.
- The lock nut is usually placed below the standard nut. To make the joint, the lock nut is first screwed.
- tightly and then the standard nut is tightened till it touches the lock nut. Afterwards, the locknut is then screwed back on the standard nut, which is held by a spanner.
- The threads of the two nuts become wedged between the threads of the bolt.
- When the lock nut is first screwed into its position, the top flanks of it press against the bottom flanks of the bolt (Fig. b).
- Fig *c* shows the condition between the flanks of the nuts and the bolt, when the second nut is locked in position. It may be observed that in this position, the top flanks of the top nut, press against the bottom flanks of the bolt, whereas, the bottom flanks of the lock nut press against the top flanks of the bolt



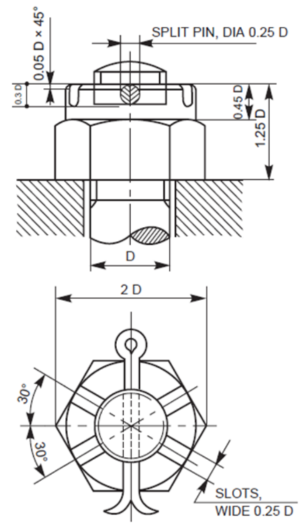
(a) (b) (c)

Locking by a split pin

- A split pin, made of steel wire of semi-circular cross-section is used for locking the nut.
- In this arrangement, the split pin is inserted through a hole in the bolt body and touching just the top surface of the nut. Then, the ends of the pin are split open to prevent it from coming out while in use



Locking by a castell nut:



- A castle nut is a hexagonal nut with a cylindrical collar turned on one end.
- Threads are cut in the nut portion only and six rectangular slots are cut through the collar.
- A split pin is inserted through a hole in the bolt body after adjusting the nut such that the hole in the
- bolt body comes in-line with slots.
- This arrangement is used in automobile works

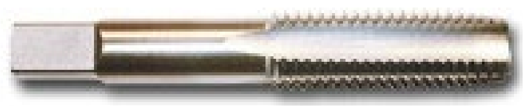
Nylon lock nut:

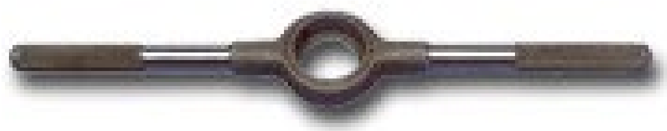
Spin Lock Nuts:

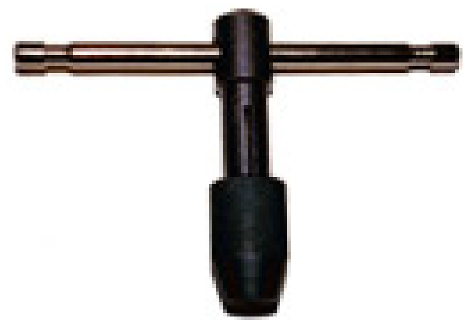
- A flange provides a large bearing surface and also acts as a washer for these nuts. Locking action is provided by serrations on the flange bottom.

Tap and Dies:

- Taps for internal threading
- Die for external threading







SAE: Society of automotive engineering
ASTM: American society for testing of material
Proof strength: around %85-95 of the yields strength
Grades applies on STEEL made bots only, not for other material

Fastener Materials and Methods of Manufacture:

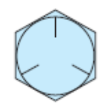
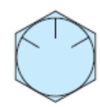
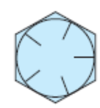
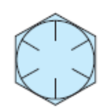
- Materials for screws, nuts, and bolts are normally selected on the basis of strength (at the operating temperatures involved), weight, corrosion resistance, magnetic properties, life expectancy, and cost.

- Most fasteners are made from steel of specifications standardized by the Society of Automotive Engineers and summarized in Tables below:

MENG 3050-Power Transmission Components

Specifications for Steel Used in Inch Series Screws and Bolts

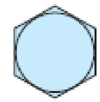
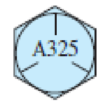
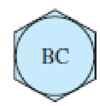
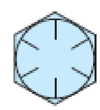
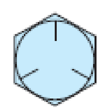
SAE Specifications for Steel Bolts

SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$ $\frac{7}{8}$–$1\frac{1}{2}$	55 33	74 60	57 36	Low or medium carbon	
4	$\frac{1}{4}$–$1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
5.2	$\frac{1}{4}$–1	85	120	92	Low-carbon martensite, Q&T	
7	$\frac{1}{4}$–$1\frac{1}{2}$	105	133	115	Medium-carbon alloy, Q&T	
8	$\frac{1}{4}$–$1\frac{1}{2}$	120	150	130	Medium-carbon alloy, Q&T	
8.2	$\frac{1}{4}$–1	120	150	130	Low-carbon martensite, Q&T	

Note: Q & T : Quenched and Tempered
SAE stand for Society of Automobile Engineers.

MENG 3050-Power Transmission Components

ASTM Specifications for Steel Bolts

ASTM Desig-nation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$–1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 2	$\frac{1}{2}$–1	85	120	92	Low-carbon, martensite, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 3	$\frac{1}{2}$–1	85	120	92	Weathering steel, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A354, grade BC	$\frac{1}{4}$–$2\frac{1}{2}$	105	125	109	Alloy steel, Q&T	
	$2\frac{3}{4}$–4	95	115	99		
A354, grade BD	$\frac{1}{4}$–4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$–1	85	120	92	Medium-carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
	$1\frac{3}{4}$–3	55	90	58		
A490, type 1	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

ASTM: stand for American Society for Testing and Materials

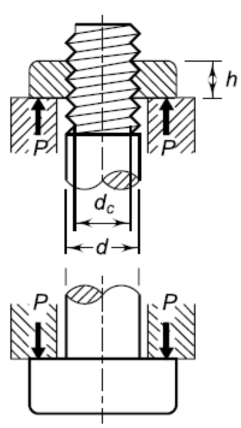
Specifications for Steel Used in Millimeter Series Screws and Bolts

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

Bolted joint analyses:

P, Bolted joint tensile force.
d_c: the weakest cross section
d: bolt nominal diameter
h: nut height
f_s: factor of safety



Maximum tensile stress is given by:

$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} d_c^2\right)}$$

To find the shear in the nut thread: need to make the following assumptions:

- Equal loads are distributing on nut threads,
- There is no stress concentration,
- Failure occurs in bolt thread not in the nut.
- Yield strength in shear is equal to ½ the yield strength in tension.

$(S_{sy} = 0.5S_{yt})$

Hence:

$$\sigma_t = \frac{S_{yt}}{(fs)}$$

Lead to:

$$P = \frac{\pi}{4} d_c^2 \left(\frac{S_{yt}}{fs} \right) \qquad \text{(a)}$$

Shear stress will be subjected to the nut thread area equal to:

$(\pi\, d_c\, h)$,

Applying shear stress formula:

$$\begin{aligned} P &= (\pi d_c h)\left(\frac{S_{sy}}{fs} \right) \\ &= (\pi d_c h)\left(\frac{S_{yt}}{2\, fs} \right) \qquad \text{(b)} \end{aligned}$$

Equating (a) and (b),

$$h = 0.5d_c$$

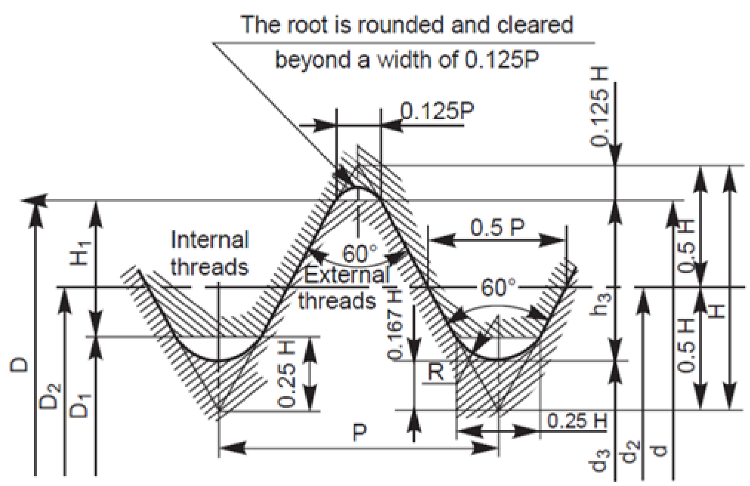
Assuming $(d_c = 0.8d)$,

$$h = 0.4d$$

Therefore:

- For standard coarse threads, when the height of nut is approximately 0.4 times the nominal diameter of the bolt, the threads are equally strong in failure by shear and failure by tension,
- The height of standard hexagonal nut is 0.8d, hence threads will not fail by shear.

Or use the geometry of an ISO thread:



Internal thread diameters
D - Major diameter
D_2 - Pitch diameter
D_1 - Minor diameter

External thread diameters
d - Major diameter
d_2 - Pitch diameter
d_3 - Minor diameter

Where:

P = Pitch
$H = 0.86\ P$
$D = d$ = Major diameter
$D_2 = d_2 = d - 0.75H$
$D_1 = d_2 - 2(H/2 - H/4) = d - 2H_1$

$d_3 = d_2 - 2\ (H/2 - H/6)$
$= d - 1.22P$
$H_1 = (D - D_1)/2 = 5H/8 = 0.54P$
$h_3 = (d - d_3)/2 = 17/24H = 0.61P$
$R = H/6 = 0.14P$

Basic dimension for ISO metric coarse threads:

Designation	*Nominal or major dia d/D (mm)*	*Pitch (p) (mm)*	*Pitch diameter d_p / D_P (mm)*	*Minor diameter (mm)*		*Tensile stress area (mm^2)*
				d_c	D_c	
M 4	4	0.70	3.545	3.141	3.242	8.78
M 5	5	0.80	4.480	4.019	4.134	14.20
M 6	6	1.00	5.350	4.773	4.917	20.10
M 8	8	1.25	7.188	6.466	6.647	36.60
M 10	10	1.50	9.026	8.160	8.376	58.00
M 12	12	1.75	10.863	9.853	10.106	84.30
M 16	16	2.00	14.701	13.546	13.835	157
M 20	20	2.50	18.376	16.933	17.294	245
M 24	24	3.00	22.051	20.319	20.752	353
M 30	30	3.50	27.727	25.706	26.211	561
M 36	36	4.00	33.402	31.093	31.670	817
M 42	42	4.50	39.077	36.479	37.129	1120
M 48	48	5.00	44.752	41.866	42.587	1470
M 56	56	5.50	52.428	49.252	50.046	2030
M 64	64	6.00	60.103	56.639	57.505	2680
M 72	72	6.00	68.103	64.639	65.505	3460
M 80	80	6.00	76.103	72.639	73.505	4340
M 90	90	6.00	86.103	82.639	83.505	5590
M 100	100	6.00	96.103	92.639	93.505	7000

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Basic dimension for ISO metric Fine threads:

Designation	Nominal or major dia d/D (mm)	Pitch (p) (mm)	Pitch diameter d_p/D_p (mm)	Minor diameter (mm)		Tensile stress area (mm^2)
				d_c	D_c	
M 6 × 1	6	1.00	5.350	4.773	4.917	20.1
M 6 × 0.75	6	0.75	5.513	5.080	5.188	22.0
M 8 × 1.25	8	1.25	7.188	6.466	6.647	36.6
M 8 × 1	8	1.00	7.350	6.773	6.917	39.2
M 10 × 1.25	10	1.25	9.188	8.466	8.647	61.2
M 10 × 1	10	1.00	9.350	8.773	8.917	64.5
M 12 × 1.5	12	1.50	11.026	10.160	10.376	88.1
M 12 × 1.25	12	1.25	11.188	10.466	10.647	92.1
M 16 × 1.5	16	1.50	15.026	14.160	14.376	167
M 16 × 1	16	1.00	15.350	14.773	14.917	178
M 20 × 2	20	2.00	18.701	17.546	17.835	258
M 20 × 1.5	20	1.50	19.026	18.160	18.376	272
M 24 × 2	24	2.00	22.701	21.546	21.835	384
M 24 × 1.5	24	1.50	23.026	22.160	22.376	401
M 30 × 3	30	3.00	28.051	26.319	26.752	581
M 30 × 2	30	2.00	28.701	27.546	27.835	621
M 36 × 3	36	3.00	34.051	32.319	32.752	865
M 36 × 2	36	2.00	34.701	33.546	33.835	915
M 42 × 4	42	4.00	39.402	37.093	37.670	1150
M 42 × 3	42	3.00	40.051	38.319	38.752	1210
M 48 × 4	48	4.00	45.402	43.093	43.670	1540
M 48 × 3	48	3.00	46.051	44.319	44.752	1600

Basic Dimensions of Unified Screw Threads (imperial threads)

Size	Major Diameter d (in.)	Coarse Threads—UNC: Threads per Inch	Coarse Threads—UNC: Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC: Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF: Threads per Inch	Fine Threads—UNF: Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF: Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599
$\frac{9}{16}$	0.5625	12	0.4603	0.182	18	0.4943	0.203
$\frac{5}{8}$	0.6250	11	0.5135	0.226	18	0.5568	0.256
$\frac{3}{4}$	0.7500	10	0.6273	0.334	16	0.6733	0.373
$\frac{7}{8}$	0.8750	9	0.7387	0.462	14	0.7874	0.509
1	1.0000	8	0.8466	0.606	12	0.8978	0.663
$1\frac{1}{8}$	1.1250	7	0.9497	0.763	12	1.0228	0.856
$1\frac{1}{4}$	1.2500	7	1.0747	0.969	12	1.1478	1.073
$1\frac{3}{8}$	1.3750	6	1.1705	1.155	12	1.2728	1.315
$1\frac{1}{2}$	1.5000	6	1.2955	1.405	12	1.3978	1.581
$1\frac{3}{4}$	1.7500	5	1.5046	1.90			
2	2.0000	$4\frac{1}{2}$	1.7274	2.50			
$2\frac{1}{4}$	2.2500	$4\frac{1}{2}$	1.9774	3.25			
$2\frac{1}{2}$	2.5000	4	2.1933	4.00			
$2\frac{3}{4}$	2.7500	4	2.4433	4.93			
3	3.0000	4	2.6933	5.97			
$3\frac{1}{4}$	3.2500	4	2.9433	7.10			
$3\frac{1}{2}$	3.5000	4	3.1933	8.33			
$3\frac{3}{4}$	3.7500	4	3.4433	9.66			
4	4.0000	4	3.6933	11.08			

Example:

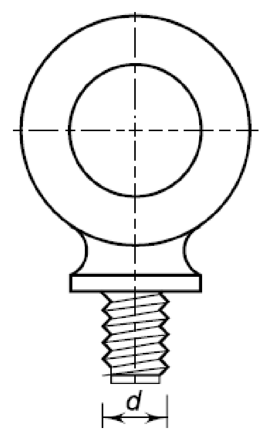
A metal block weighing 10KN, lifted by an eye bolt to be screws to the center of gravity line of action of the block. If the eye bolt has coarse thread, made of carbon steel 30C8 (S_{yt}=400N/mm^2) with factor of safter equal to 6. What would be the bolt size?

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{6} = 66.67 \text{ N/mm}^2$$

$$\sigma_t = \frac{P}{\frac{\pi}{4} d_c^2} \qquad \therefore 66.67 = \frac{(10 \times 10^3)}{\frac{\pi}{4} d_c^2}$$

$$d_c = 13.82 \text{ mm}$$

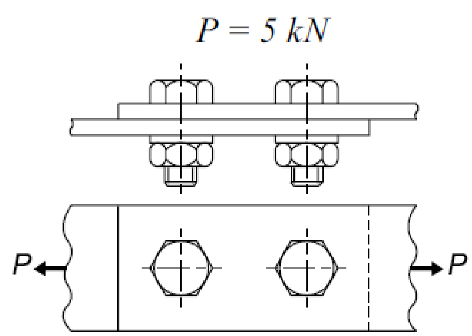
$$d = \frac{d_c}{0.8} = \frac{13.82}{0.8} = 17.27 \text{ or } 18 \text{ mm}$$



From the table we go with M20…

Example:

Two plates are fastened as shown, the bolt made of carbon steel 30C8 (S_{yt}=400N/mm^2), and safety factor of 5. Determine the bolt size.



$$S_{sy} = 0.5\, S_{yt} = 0.5\,(400) = 200 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{200}{5} = 40 \text{ N/mm}^2$$

$$\text{Shear area of 2 bolts} = 2\left(\frac{\pi}{4} d^2\right) \text{ mm}^2$$

$$\text{Therefore,} \quad P = 2\left(\frac{\pi}{4} d^2\right)\tau$$

$$5 \times 10^3 = 2\left(\frac{\pi}{4} d^2\right)(40)$$

or $\quad d = 8.92$ or 9 mm

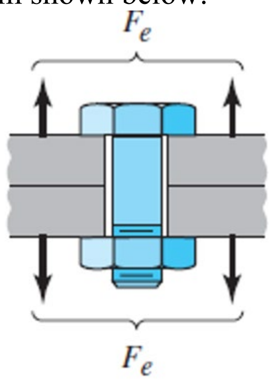
Hence, we will use M10,

HUMBER

MENG 3050-Power Transmission Components

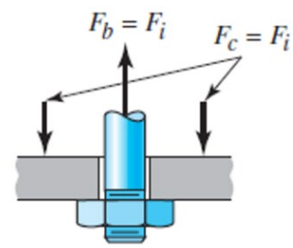
BOLT TENSION WITH EXTERNAL JOINT-SEPARATING FORCE:

Bolts are typically used to hold parts together in opposition to forces tending to pull, or slide, them apart. The general case of two parts connected with a bolt and subjected to an external force *Fe* tending to separate them shown below:



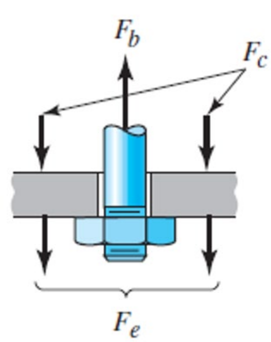
(*a*)
Complete joint

Apportion of the above assembly as a free body diagram shown, the nut has been tightened, but the external force has not yet been applied. The bolt axial load *Fb* and the clamping force between the two plates *Fc* are both equal to the initial tightening force *Fi*.



(*b*)
Free body without external load

The figure below shows the same members as a free body after external force F_e has been applied.

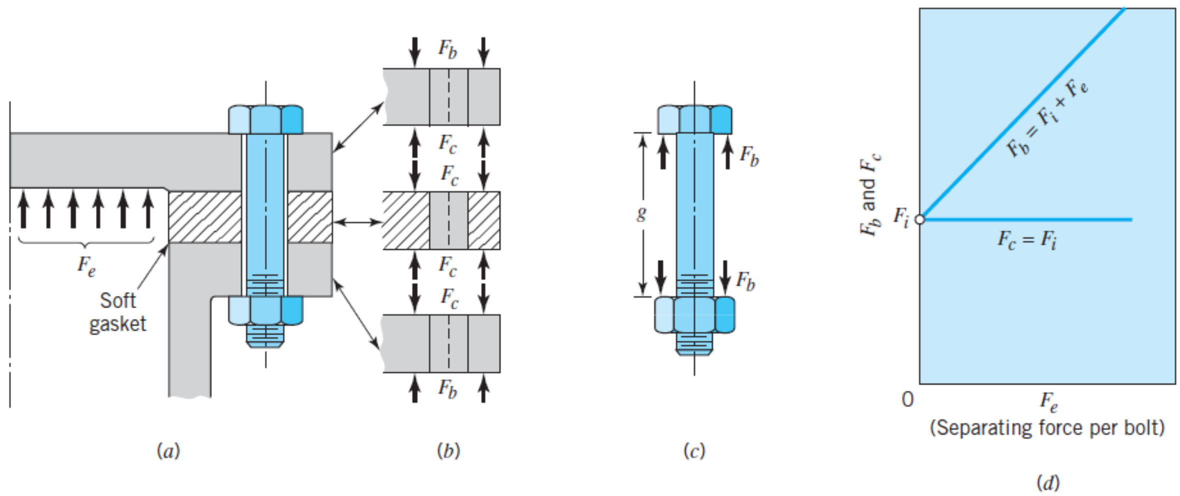


(*c*)
Free body with external load

MENG 3050-Power Transmission Components

Example:

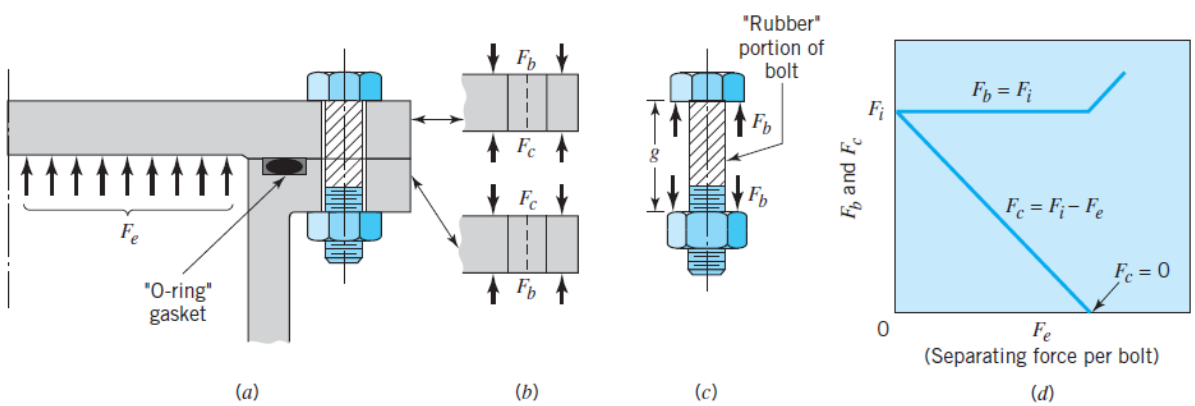
F_b and F_c versus F_e per bolt for soft clamped members - rigid bolt



The change in *Fb* and *Fc* as separating load *Fe* is applied. The elastic stretch of the bolt caused by *Fe* is so small that the thick rubber gasket cannot expand significantly. Thus, the clamping force *Fc* does not diminish, and the *entire* load *Fe* goes to increasing bolt tension.

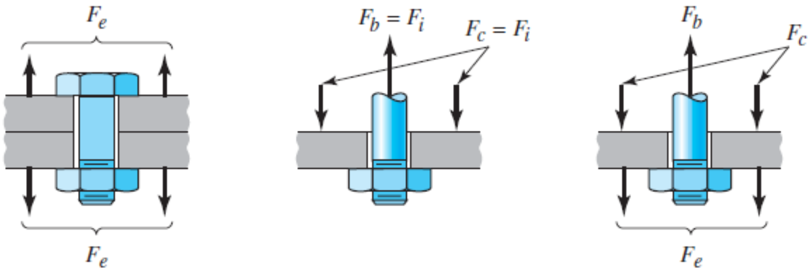
Example:

Fb and Fc versus Fe per bolt for rigid clamped members—soft bolt.



The clamped members are "rigid" metal parts with precision-ground mating surfaces and no gasket between them. The bolt has a center portion made of rubber. In this case the initial tightening stretches the bolt; it does not significantly compress the clamped members. (Sealing of the fluid is accomplished by a "confined gasket" in the form of a rubber O-ring. Before being compressed by the cover plate, the cross section of the O-ring was circular).

The separating force must be equal to the sum of the increased bolt force plus the decreased clamping force, or



(*a*) Complete joint | (*b*) Free body without external load | (*c*) Free body with external load

$$F_e = \Delta F_b + \Delta F_c$$

By definition,

$$\Delta F_b = k_b \delta \quad \text{and} \quad \Delta F_c = k_c \delta$$

$$F_e = (k_b + k_c)\delta \quad \text{or} \quad \delta = \frac{F_e}{k_b + k_c}$$

Combining the equations:

$$\Delta F_b = \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad \Delta F_c = \frac{k_c}{k_b + k_c} F_e$$

Substitute:

$$F_b = F_i + \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad F_c = F_i - \frac{k_c}{k_b + k_c} F_e$$

The basic equations for axial deflection ($\delta = PL/AE$) and for spring rate ($k = P/\delta$),

$$k_b = \frac{A_b E_b}{g} \quad \text{and} \quad k_c = \frac{A_c E_c}{g}$$

where the grip g represents the approximate effective length for both

THE TORQUE METHOD:

To impart a preloaded part**:**

T = C. D. F_i

where T = torque
C = torque coefficient
D = nominal diameter of thread
F_i = desired initial preload

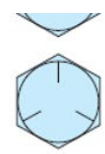
Torque coefficient equal to:

- 0.15 for intensely lubricated fasteners
- 0.2 for partially lubricated, and if cutting oil not washed out yet,
- 0.34 for dry assembly.

Example:

A ¾-UNC-grade 5 bolt is to be preloaded to 85 percent of its proof strength.
- The length of the engagement is 5 inches.
- The bolt is new and non-lubricated but likely has traces of cutting oil present.
- Determine the required torque.

5	$\frac{1}{4}$–1	85	120	92	Medium carbon, Q&T
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81	



A_S = .334 in^2
S_p = 85 ksi
$F_i = .85\ S_p\ A_s$
F_i = .85 (85,000 lb/in^2) (.334 in^2)
F_i = 24,130 lb

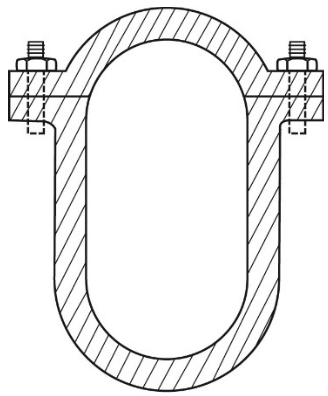
Using C = .2 non-lubricated with traces of oil:

$T = C\ D\ F_i$
T = 0.2 (¾ in) 24,130 lb
T = 3620 in-lb or 302 ft-lb

MENG 3050-Power Transmission Components

Example Problem:

- As shown below, a pressure vessel has a sealed head diameter of 20 inches. Using a gasket in between.
- It uses ten 1¼ UNF studs that are fully threaded and have an effective clamping length of 6 inches.
- The clamped area of the flange is 50 in².
- The studs are made of class 8 material.
- $E = 30 \times 10^6$ psi (both bolts and flange).



Section view

A) Find the torque, assuming new lubricated threads necessary to tighten to 90 percent of the proof strength:
B) If the pressure vessel is now pressurized to 500 psi, determine the total load on the bolts. Find the stiffness of both studs and flanges, would the bolts bear such load?

Solution: Find the torque

A) $T = C.D.\,F_i$

$F_i = S\,A$

$F_i = 0.90\ (120{,}000\ lb/in^2)\ (1.073\ in^2)$

$F_i = 115{,}884\ lb$

$T = C\,D\,F_i$

$C = 0.15$ lubricating threads

$T = 0.15\ (1.25\ in)\ (115{,}884\ lb)$

$T = 21{,}728$ in-lb or 1810 ft-lb

b) total load on the bolts

$$k_b = \frac{AE}{L} = \frac{1.073\ in^2 \quad 30 \text{ x } 10^6\ lb/in^2}{6\ in}$$

$k_b = 5.365 \text{ x } 10^6$ *lb/in per stud*

$$k_c = \frac{AE}{L} = \frac{50\ in^2 \quad 30 \text{ x } 10^6\ lb/in^2}{6\ in}$$

$k_c = 2.5 \text{ x } 10^8\ lb/in$

$F_e = P\ A$ *(pressure x projected area)*

$$A = \frac{\pi D^2}{4} = \pi\,(20\ in)^2/4 = 314\ in^2$$

$F_e = 500\ lb/in^2\ (314\ in^2)$

$F_e = 157{,}000\ lb$

$$F_B = F_i + \left(\frac{k_b}{k_b + k_c}\right) F_e$$

$$F_B = 10\,(115{,}884\ lb) + \left(\frac{10 \text{ x } 5.365 \text{ x } 10^6\ lb/in}{10 \text{ x } 5.365 \text{ x } 10^6\ lb/in + 2.5 \text{ x } 10^8\ lb/in}\right) 157{,}000\ lb$$

$F_B = 1.19 \text{ x } 10^6\ lb$ *or* *119,000 lb/stud*