

Fasteners and Non-Permanent Joints – Applications to Motorcycles

CVMG - Essex-Kent Chapter Meeting

Peter Frise
March 2, 2017

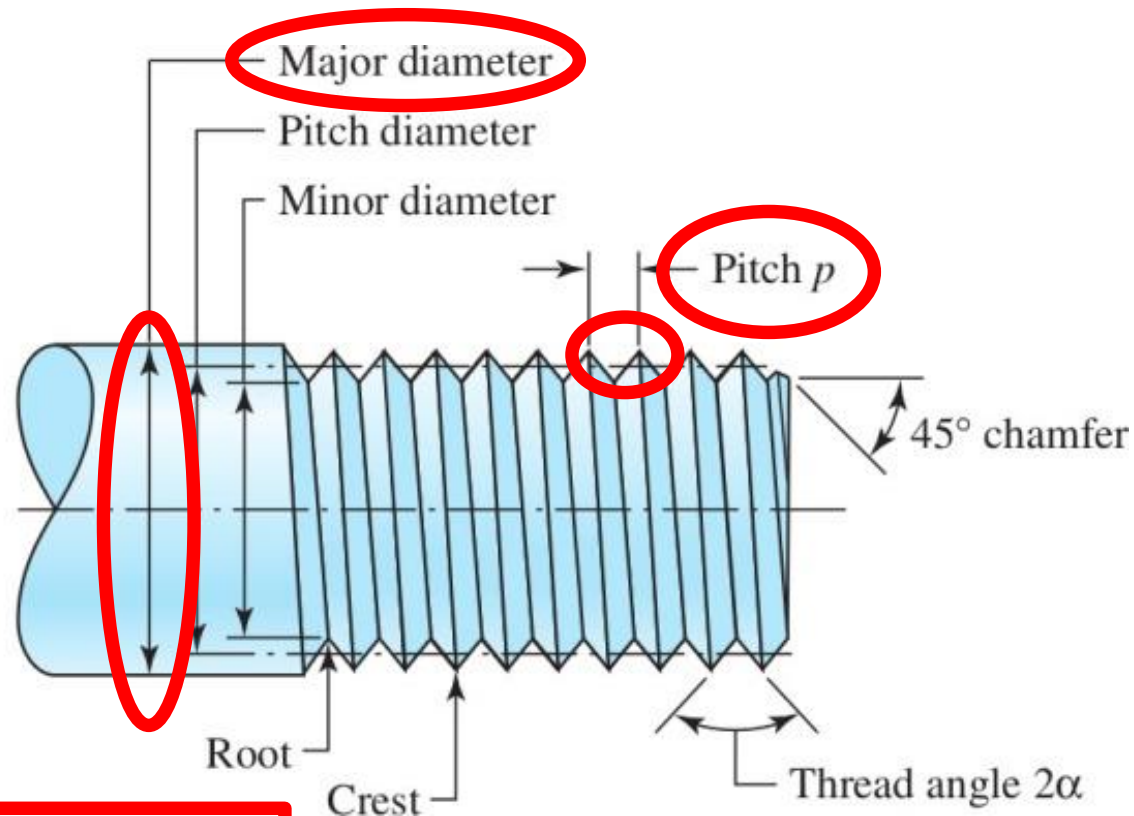
PRESENTATION OUTLINE

- **Threaded fastener terminology & geometry**
- **How threaded fasteners work**
 - *Strength of engineering materials*
 - *Tensile loading vs. shear loading and failure modes*
 - *Torque variation*
 - *Bolt stretch – torque-to-yield fasteners*
 - *Fastener strength properties*
 - *Fastener grades*
- **A simple sample calculation**
- **Applications to motorcycles**

THREAD GEOMETRY & NOMENCLATURE

Figure 8-1

Terminology of screw threads. Sharp vee threads shown for clarity; the crests and roots are actually flattened or rounded during the forming operation.



Examples:

$\frac{1}{4}$ -20 UNC = $\frac{1}{4}$ major dia., 20 TPI Unified
National Coarse Series

M6-1 = Metric 6mm major dia., 1mm pitch

DETAILS OF FASTENER THREAD GEOMETRY

Figure 8-2

Basic profile for metric M and MJ threads.

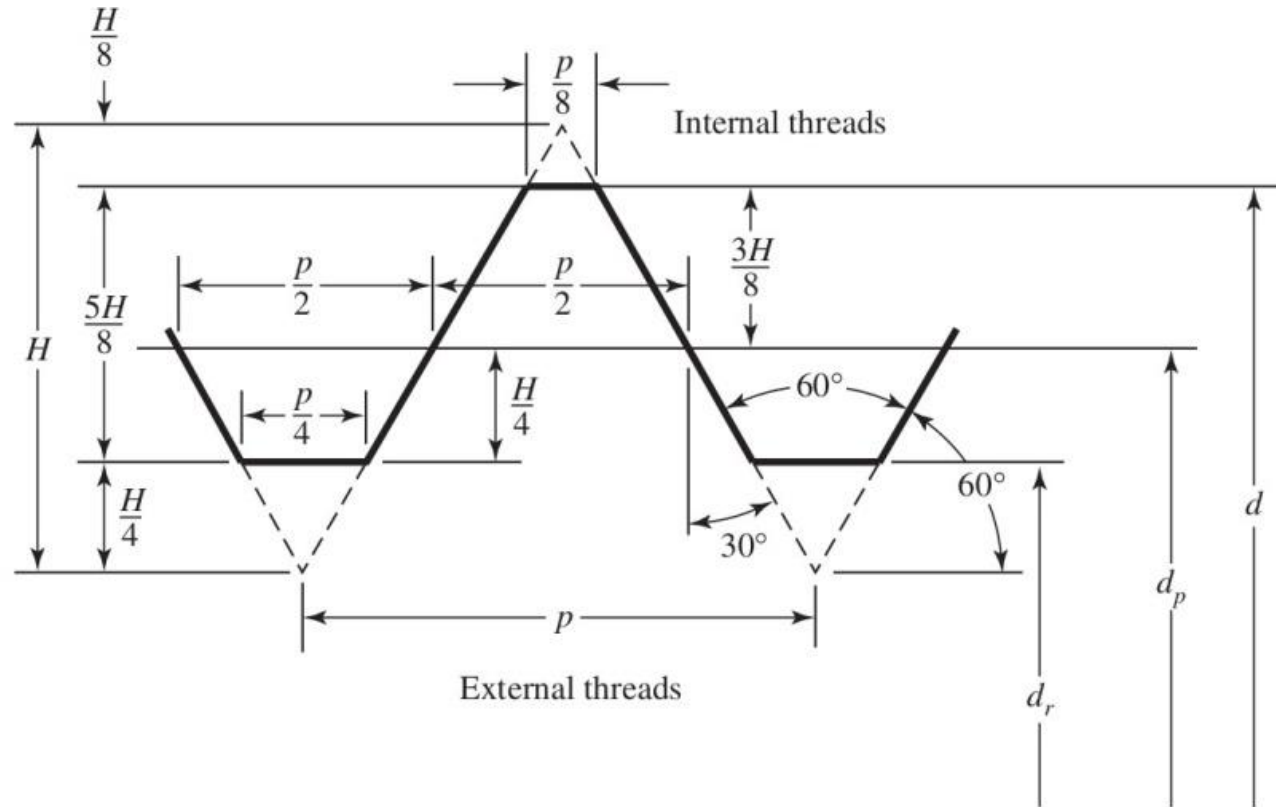
d = major diameter

d_r = minor diameter

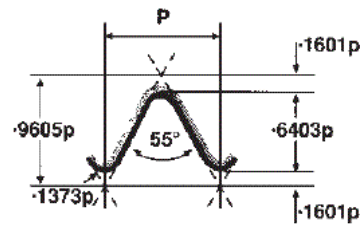
d_p = pitch diameter

p = pitch

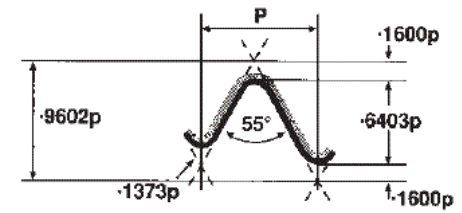
$H = \frac{\sqrt{3}}{2} p$



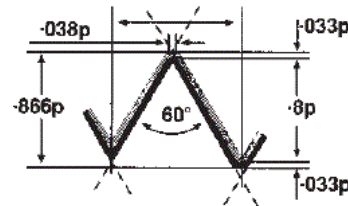
THREAD GEOMETRY & NOMENCLATURE



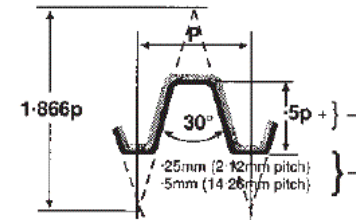
B.S.W.



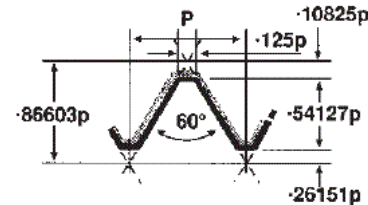
B.S.P.T. (I.S.O. TAPER PIPE)



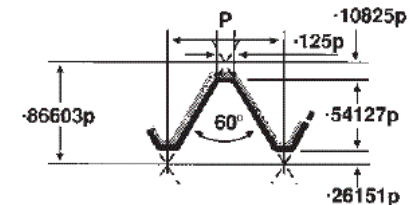
U.S. PIPE TAPER



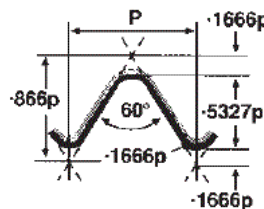
TRAPEZIODAL



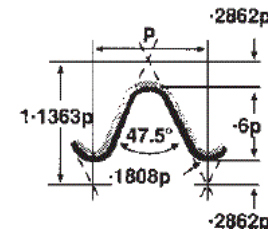
I.S.O. UNIFIED



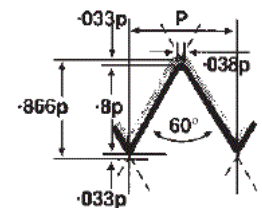
I.S.O. METRIC



B.S. CYCLE



B.A.



U.S. PIPE STRAIGHT

INCH-METRIC TAP DRILL CHART

(NO HOME SHOULD BE
WITHOUT ONE!)

Threaded Fasteners - Peter Frise

INCH/METRIC TAP DRILL SIZES & DECIMAL EQUIVALENTS

Drill Size	Decimal eQuIvalent	Tap Size	Drill Size	Decimal eQuIvalent	Tap Size	Drill Size	Decimal eQuIvalent	Tap Size
1 64	.0156		10	.1935		59	.9219	1 - 12
	.0145			.1960		64	.9375	1 - 14
	.0160			.1990			.9531	
	.0180		13 64	.2010	1/4 - 20	64	.9688	1 1/8 - 7
	.0200			.2031		31	.9844	
	.0210			.2040		64	1.0000	
	.0225			.2055			1.0469	1 1/8 - 12
	.0240			.2090		13/64	1.1094	1 1/4 - 7
	.0250			.2130	1/4 - 28	17/64	1.1250	
	.0260			.2188			1.1719	1 1/4 - 12
	.0280		7 32	.2210		11/64	1.2188	1 3/8 - 6
	.0292			.2280		17/32	1.2500	
1 32	.0310		15 64	.2340			1.2969	1 3/8 - 12
	.0312			.2344		119/64	1.3438	1 1/2 - 6
	.0320			.2380		11/32	1.3750	
	.0330			.2420			1.4219	1 1/2 - 12
	.0350			.2460		127/64	1.5000	
	.0360			.2500	5/16 - 18			
	.0370			.2570				
	.0380			.2610				
	.0390		17 64	.2656				
	.0400			.2660	5/16 - 24			
	.0410			.2720				
	.0420			.2770				
	.0430			.2810				
	.0465		9 32	.2812				
3 64	.0469	0 - 80		.2900				
	.0520			.2950				
	.0550		19 64	.2969				
	.0595	1 - 64, 72		.3020				
1 16	.0625			.3125	3/8 - 16			
	.0635			.3160				
	.0670			.3230				
	.0700	2 - 56, 64		.3281				
	.0730			.3320				
	.0760		11 32	.3390	3/8 - 24			
5 64	.0781			.3438				
	.0785	3 - 48		.3480				
	.0810		23 64	.3580				
	.0820	3 - 56		.3594				
	.0860			.3680	7/16 - 14			
	.0890	4 - 40		.3750				
	.0935	4 - 48		.3770				
3 32	.0938			.3860				
	.0960		25 64	.3906	7/16 - 20			
	.0980			.3970				
	.0995			.4040				
	.1015	5 - 40	13 32	.4062				
	.1040	5 - 44		.4130				
	.1065	6 - 32	27 64	.4219	1/2 - 13			
7 64	.1094			.4375				
	.1100		29 64	.4531	1/2 - 20			
	.1110		31 32	.4688				
	.1130	6 - 40	64	.4844	9/16 - 12			
	.1160			.5000				
1 8	.1200		33 64	.5156	9/16 - 18			
	.1250		35 32	.5312	5/8 - 11			
	.1285		64	.5469				
	.1360	8 - 32, 36	37 64	.5625	5/8 - 18			
	.1405		39 32	.5781				
9 64	.1406		64	.5938				
	.1440		64	.6094				
	.1470			.6250				
	.1495	10 - 24	41 64	.6406				
	.1520		43 32	.6562	3/4 - 10			
5 32	.1540		64	.6719	3/4 - 16			
	.1562			.6875				
	.1570		45 64	.7031				
	.1590	10 - 32	47 32	.7188				
	.1610		64	.7344				
	.1660			.7500				
	.1695		49 64	.7656	7/8 - 9			
11 64	.1719		51 32	.7812				
	.1730		64	.7969	7/8 - 14			
	.1770	12 - 24		.8125				
	.1800		53 64	.8281				
	.1820	12 - 28	55 32	.8438				
	.1850		64	.8594				
3 16	.1875			.8750				
	.1890		57 64	.8906	1 - 8			
	.1910		59 32	.9062				

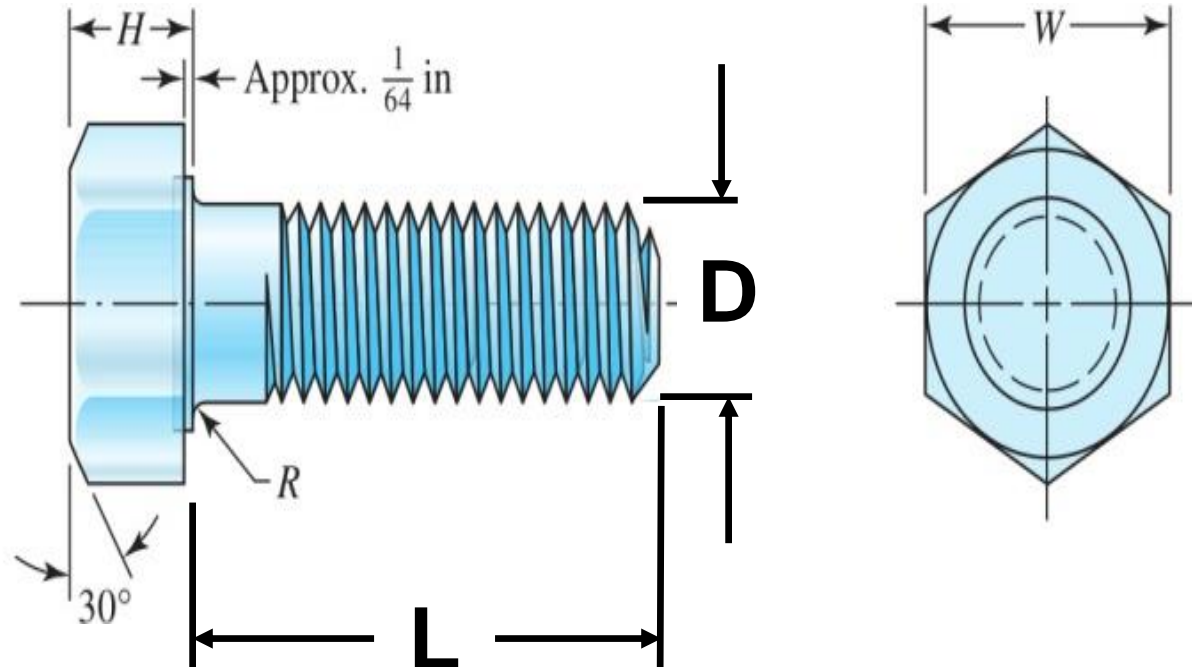
metric tap Drill Sizes		
metric tap	tap Drill (mm)	Decimal (inch)
m1.6 x 0.35	1.25	.0492
m1.8 x 0.35	1.45	.0571
m2 x 0.4	1.60	.0630
m2.2 x 0.45	1.75	.0689
m2.5 x 0.45	2.05	.0807
m3 x 0.5	2.50	.0984
m3.5 x 0.6	2.90	.1142
m4 x 0.7	3.30	.1299
m4.5 x 0.75	3.70	.1457
m5 x 0.8	4.20	.1654
m6 x 1	5.00	.1968
m7 x 1	6.00	.2362
m8 x 1.25	6.70	.2638
m8 x 1	7.00	.2756
m10 x 1.5	8.50	.3346
m10 x 1.25	8.70	.3425
m12 x 1.75	10.20	.4016
m12 x 1.25	10.80	.4252
m14 x 2	12.00	.4724
m14 x 1.5	12.50	.4921
m16 x 2	14.00	.5512
m16 x 1.5	14.50	.5709
m18 x 2.5	15.50	.6102
m18 x 1.5	16.50	.6496
m20 x 2.5	17.50	.6890
m20 x 1.5	18.50	.7283
m22 x 2.5	19.50	.7677
m22 x 1.5	20.50	.8071
m24 x 3	21.00	.8268
m24 x 2	22.00	.8661
m27 x 3	24.00	.9449
m27 x 2	25.00	.9843
m30 x 3.5	26.50	1.0433
m30 x 2	28.00	1.1024
m33 x 3.5	29.50	1.1614
m33 x 2	31.00	1.2205
m36 x 4	32.00	1.2598
m36 x 3	33.00	1.2992
m39 x 4	35.00	1.3780
m39 x 3	36.00	1.4173

pipe Thread Sizes (npsC)			
Thread	Drill	Thread	Drill
1/8 - 27	11/32	1 1/2 - 11 1/2	1 3/4
1/4 - 18	7/16	2 - 11 1/2	2 7/32
3/8 - 18	37/64	2 1/2 - 8	2 21/32
1/2 - 14	23/32	3 - 8	3 1/4
3/4 - 14	59/64	3 1/2 - 8	3 3/4
1 - 11 1/2	15/32	4 - 8 6	4 1/4
1 1/4 - 11 1/2	1 1/2		

HEX HEAD BOLT GEOMETRY & NOMENCLATURE

Figure 8-9

Hexagon-head bolt; note the washer face, the fillet under the head, the start of threads, and the chamfer on both ends. Bolt lengths are always measured from below the head.



Examples:

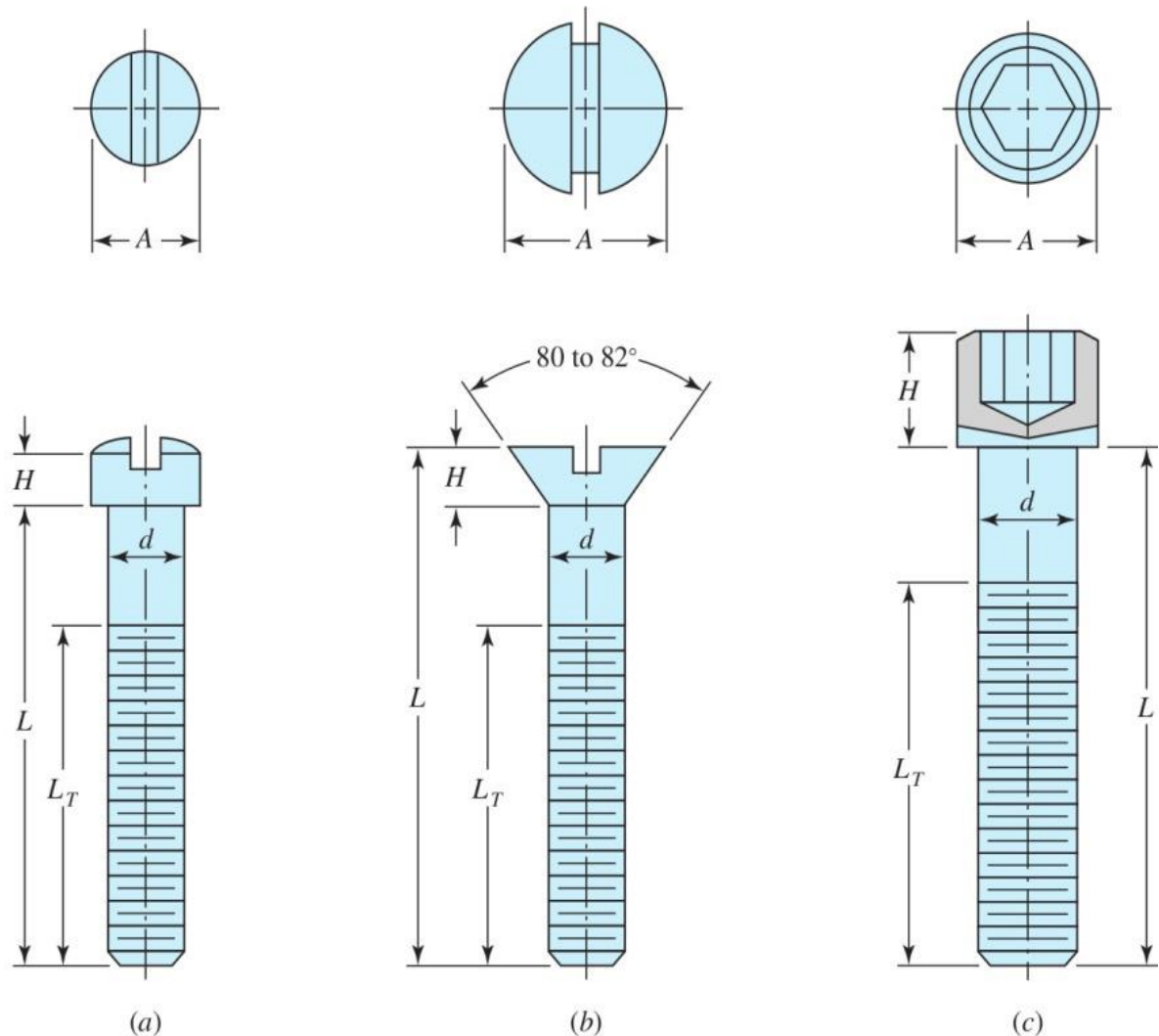
$\frac{1}{4}$ -20 x 1 = $\frac{1}{4}$ -20 UNC bolt – 1" long

M6-1 x 25 = Metric 6mm-1mm pitch bolt 25 mm long

TYPICAL CAPSCREWS

Figure 8-10

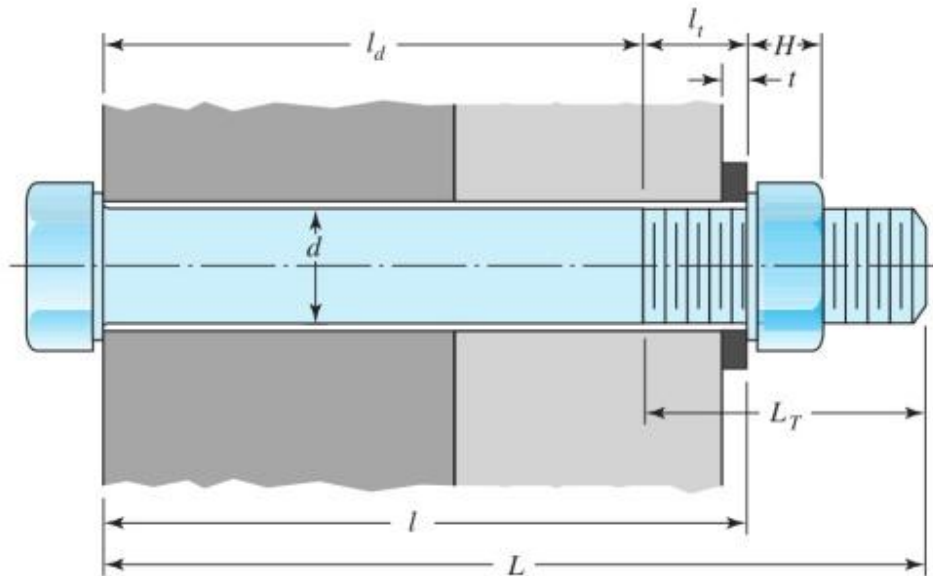
Typical cap-screw heads:
(a) fillister head; (b) flat head;
(c) hexagonal socket head. Cap screws are also manufactured with hexagonal heads similar to the one shown in Fig. 8-9, as well as a variety of other head styles. This illustration uses one of the conventional methods of representing threads.



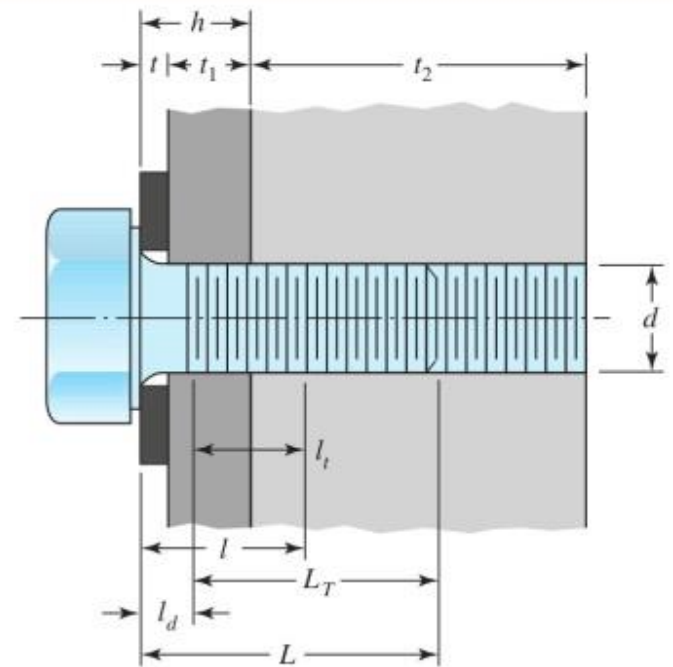
FASTENER STIFFNESS

Table 8-7

Suggested Procedure for Finding Fastener Stiffness



(a)



(b)

FASTENER STIFFNESS CALCULATION

Given fastener diameter d and pitch p in mm or number of threads per inch

Washer thickness: t from Table A-32 or A-33

Nut thickness [Fig. (a) only]: H from Table A-31

Grip length:

For Fig. (a): l = thickness of all material squeezed between face of bolt and face of nut

$$\text{For Fig. (b): } l = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 \geq d \end{cases}$$

Fastener length (round up using Table A-17*):

For Fig. (a): $L > l + H$

For Fig. (b): $L > h + 1.5d$

Threaded length L_T : Inch series:

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

Metric series:

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

Length of unthreaded portion in grip: $l_d = L - L_T$

Length of threaded portion in grip: $l_t = l - l_d$

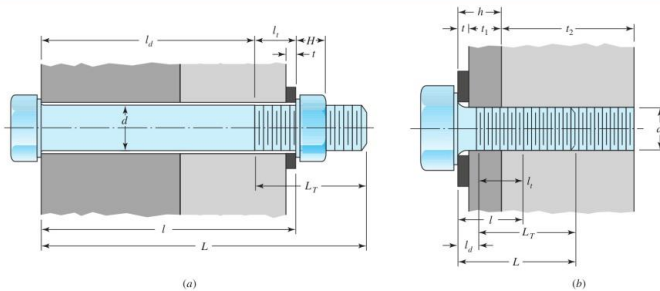
Area of unthreaded portion: $A_d = \pi d^2/4$

Area of threaded portion: A_t from Table 8-1 or 8-2

$$\text{Fastener stiffness: } k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

Table 8-7

Suggested Procedure for Finding Fastener Stiffness



*Bolts and cap screws may not be available in all the preferred lengths listed in Table A-17. Large fasteners may not be available in fractional inches or in millimeter lengths ending in a nonzero digit. Check with your bolt supplier for availability.

STRENGTH PROPERTIES OF ENGINEERING MATERIALS

Material Used	Poisson Ratio	Elastic GPa	Modulus Mpsi	A	B
Steel	0.291	207	30.0	0.787 15	0.628 73
Aluminum	0.334	71	10.3	0.796 70	0.638 16
Copper	0.326	119	17.3	0.795 68	0.635 53
Gray cast iron	0.211	100	14.5	0.778 71	0.616 16
General expression				0.789 52	0.629 14

Table 8-8

Stiffness Parameters of
Various Member
Materials[†]

[†]Source: J. Wileman,
M. Choudury, and I. Green,
“Computation of Member
Stiffness in Bolted
Connections,” *Trans. ASME,
J. Mech. Design*, vol. 113,
December 1991, pp. 432–437.

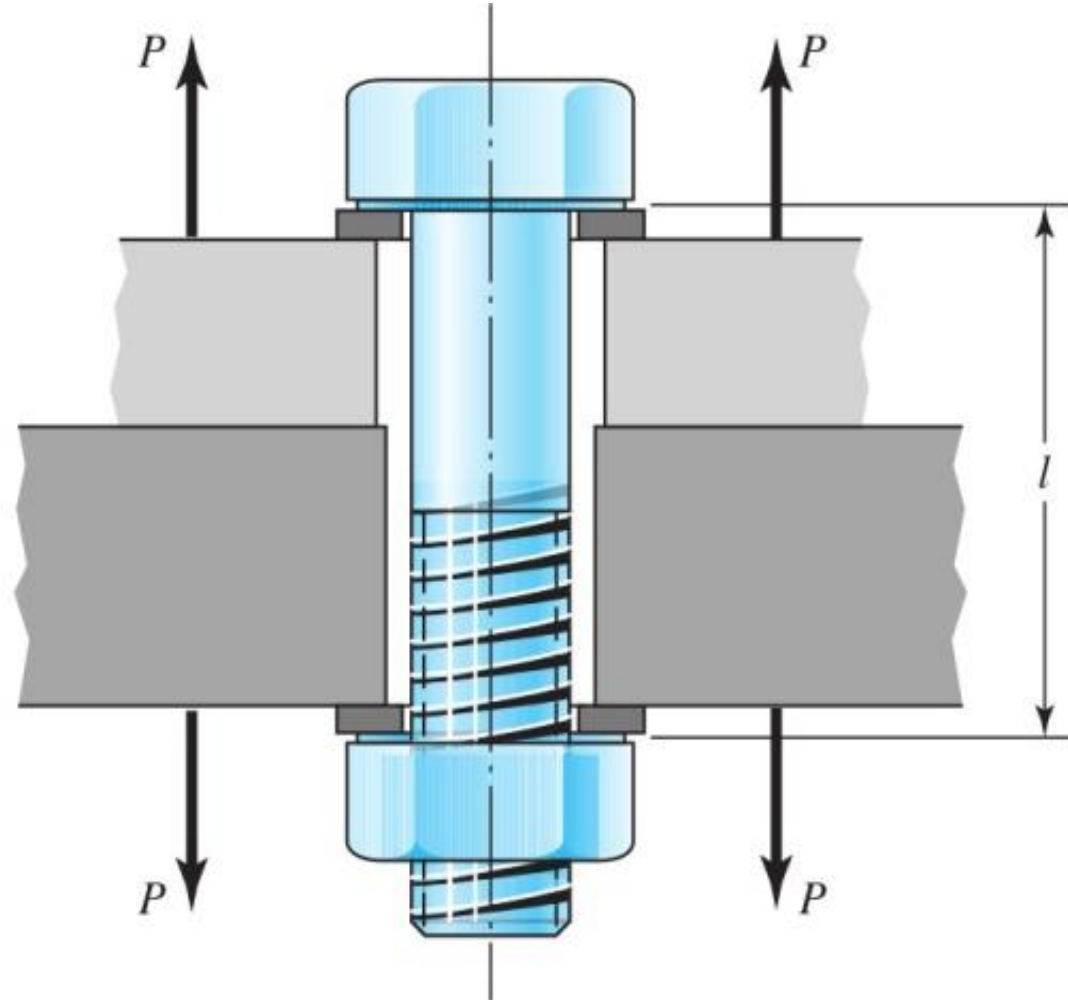
The elastic modulus of a material is basically a measure of its stiffness.

Note the much higher value for steel than aluminium – this is one of several reasons why fasteners are not made of aluminium.

TENSION LOADED LONG FASTENER WITH WASHERS & NUT

Figure 8-13

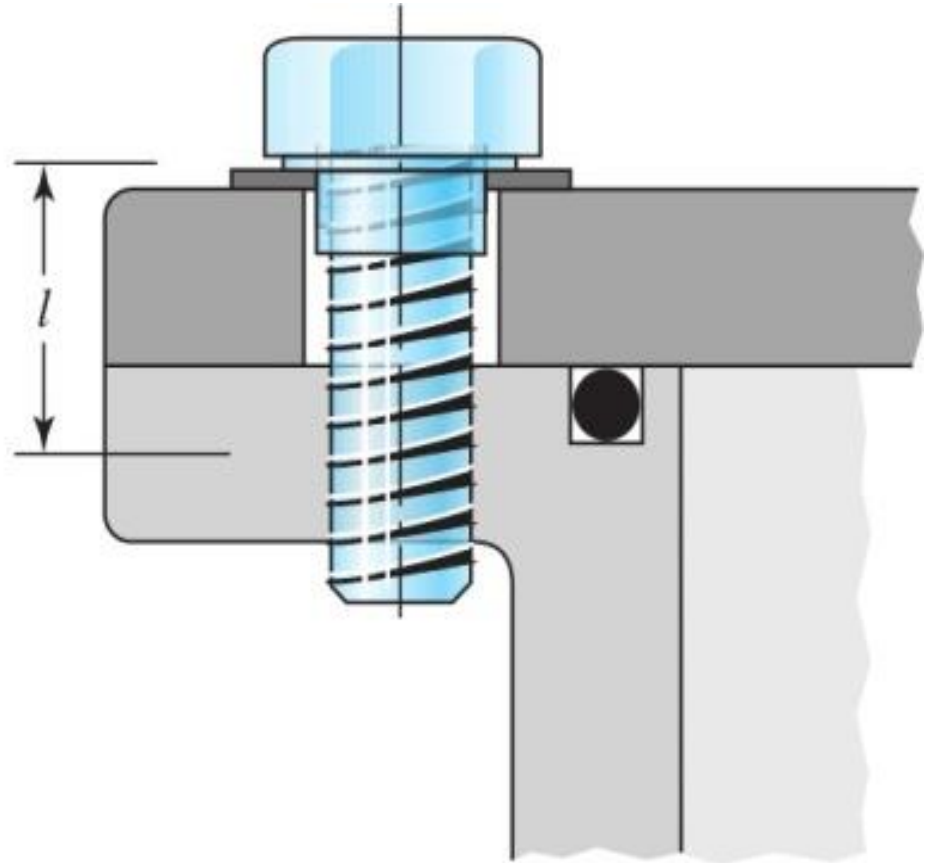
A bolted connection loaded in tension by the forces P . Note the use of two washers. Note how the threads extend into the body of the connection. This is usual and is desired. l is the grip of the connection.



TENSION LOADED SHORT FASTENER WITH O-RING OR GASKET

Figure 8-14

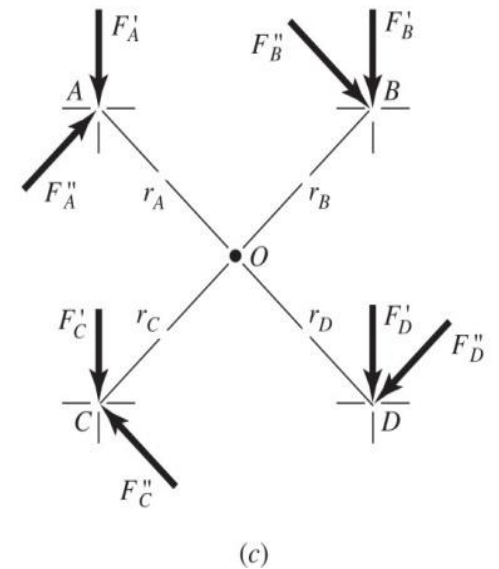
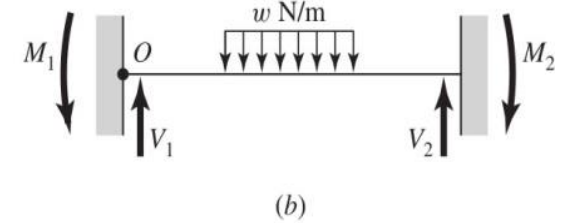
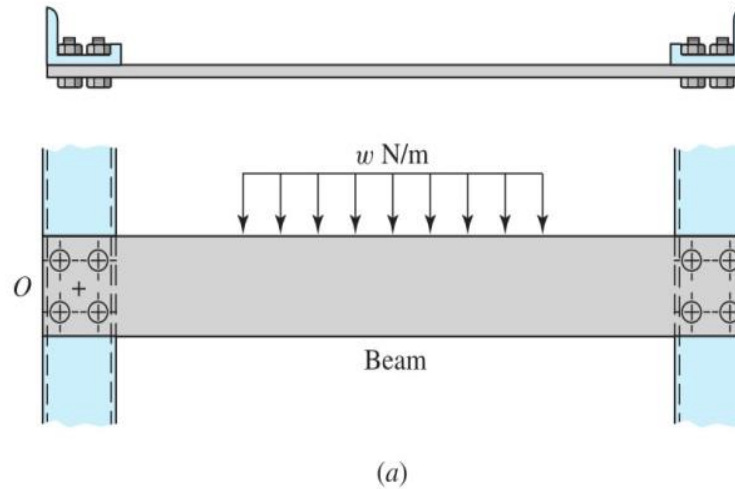
Section of cylindrical pressure vessel. Hexagon-head cap screws are used to fasten the cylinder head to the body. Note the use of an O-ring seal. l is the effective grip of the connection (see Table 8-7).



SHEAR LOADED FASTENERS

Figure 8-26

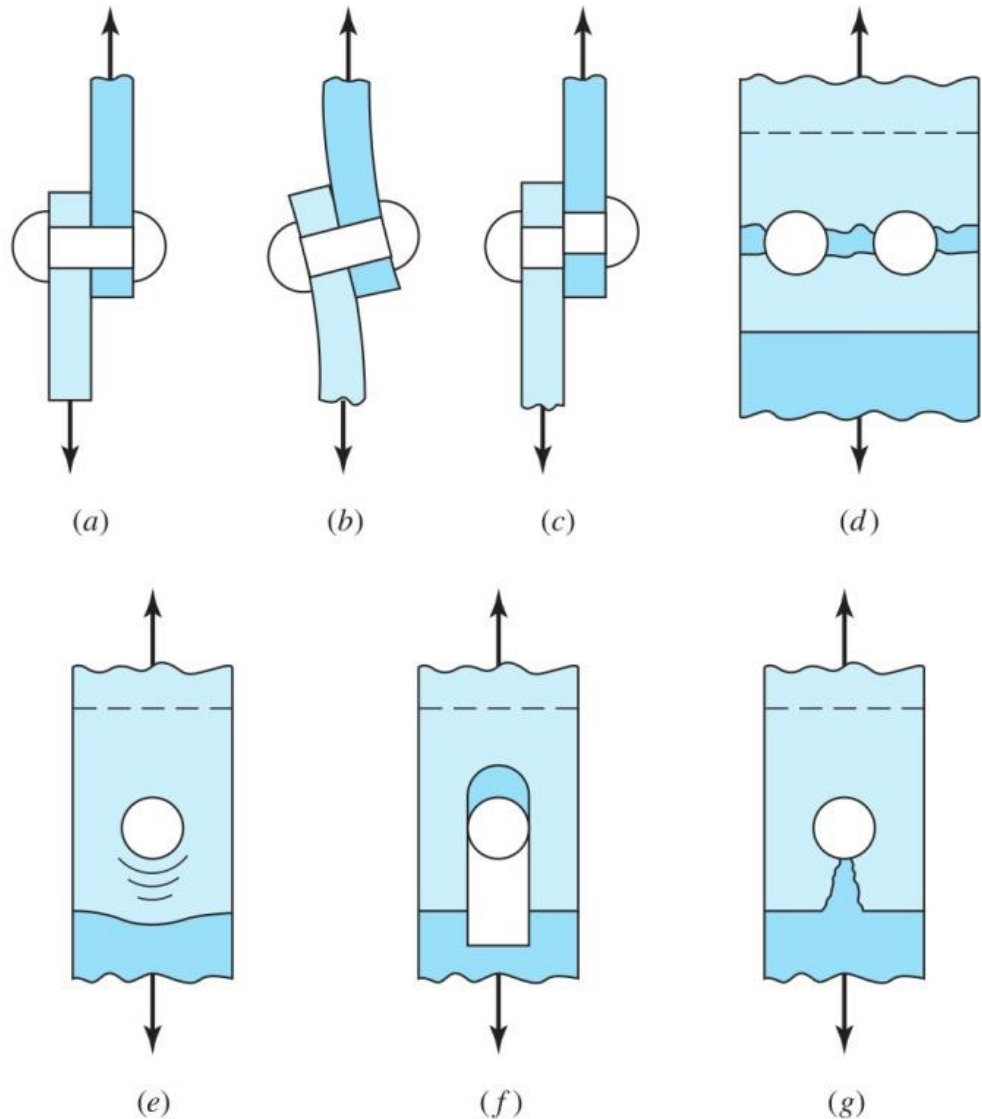
- (a) Beam bolted at both ends with distributed load;
 (b) free-body diagram of beam;
 (c) enlarged view of bolt group centered at O showing primary and secondary resultant shear forces.



FASTENER FAILURE MODES

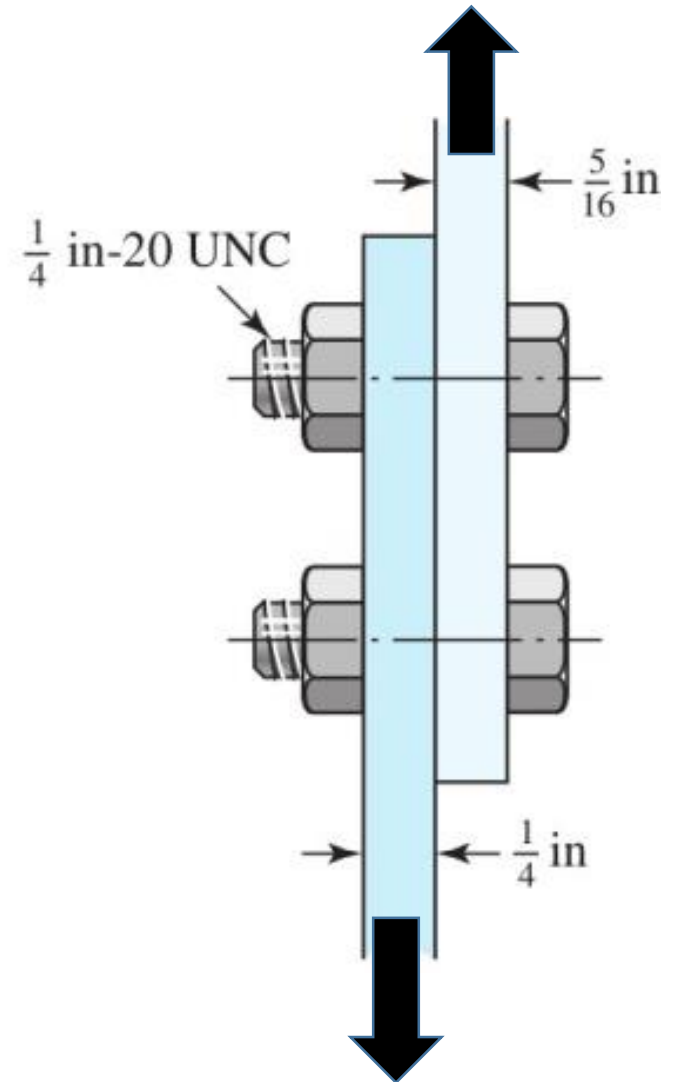
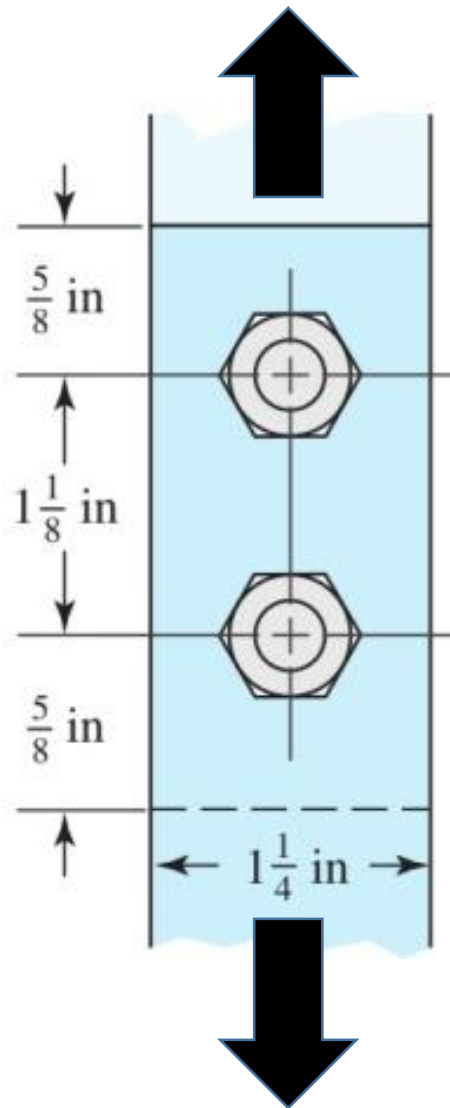
Figure 8-23

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.



SHEAR LOADED FASTENERS

Problem 8-66



TORQUE VARIATION – UN-LUBRICATED BOLTS

Table 8-13

Distribution of Preload
 F_i for 20 Tests of
Unlubricated Bolts
Torqued to 90 N · m

23.6,	27.6,	28.0,	29.4,	30.3,	30.7,	32.9,	33.8,	33.8,	33.8,
34.7,	35.6,	35.6,	37.4,	37.8,	37.8,	39.2,	40.0,	40.5,	42.7
Mean value $\bar{F}_i = 34.3$ kN. Standard deviation, $\hat{\sigma} = 4.91$ kN.									



TORQUE VARIATION - LUBRICATED BOLTS

Table 8-14

Distribution of Preload
 F_i for 10 Tests of
Lubricated Bolts
Torqued to 90 N · m

30.3,	32.5,	32.5,	32.9,	32.9,	33.8,	34.3,	34.7,	37.4,	40.5
Mean value, $\bar{F}_i = 34.18$ kN. Standard deviation, $\hat{\sigma} = 2.88$ kN.									



CONCLUSION: *Lubricated fasteners provided more consistent torque values and that ensures better joint strength and gasket sealing.*

Torque-to-Yield Fasteners

- All fastener materials have some degree of elasticity or “*stretch*” before they permanently deforms and eventually breaks;
- It can be shown that the highest level of force a bolt can achieve occurs when it is stretched to its yield point where it is permanently deformed;
- So, to achieve the maximum bolted joint strength, engineers design the joint to just make the fasteners yield;
- This means that the fastener cannot be used twice – it must be discarded after it has been torqued;
- The graph on the next slide shows this effect and it also shows how important it is to lubricate fasteners before attempting to torque them.



Torque-to-Yield Fasteners

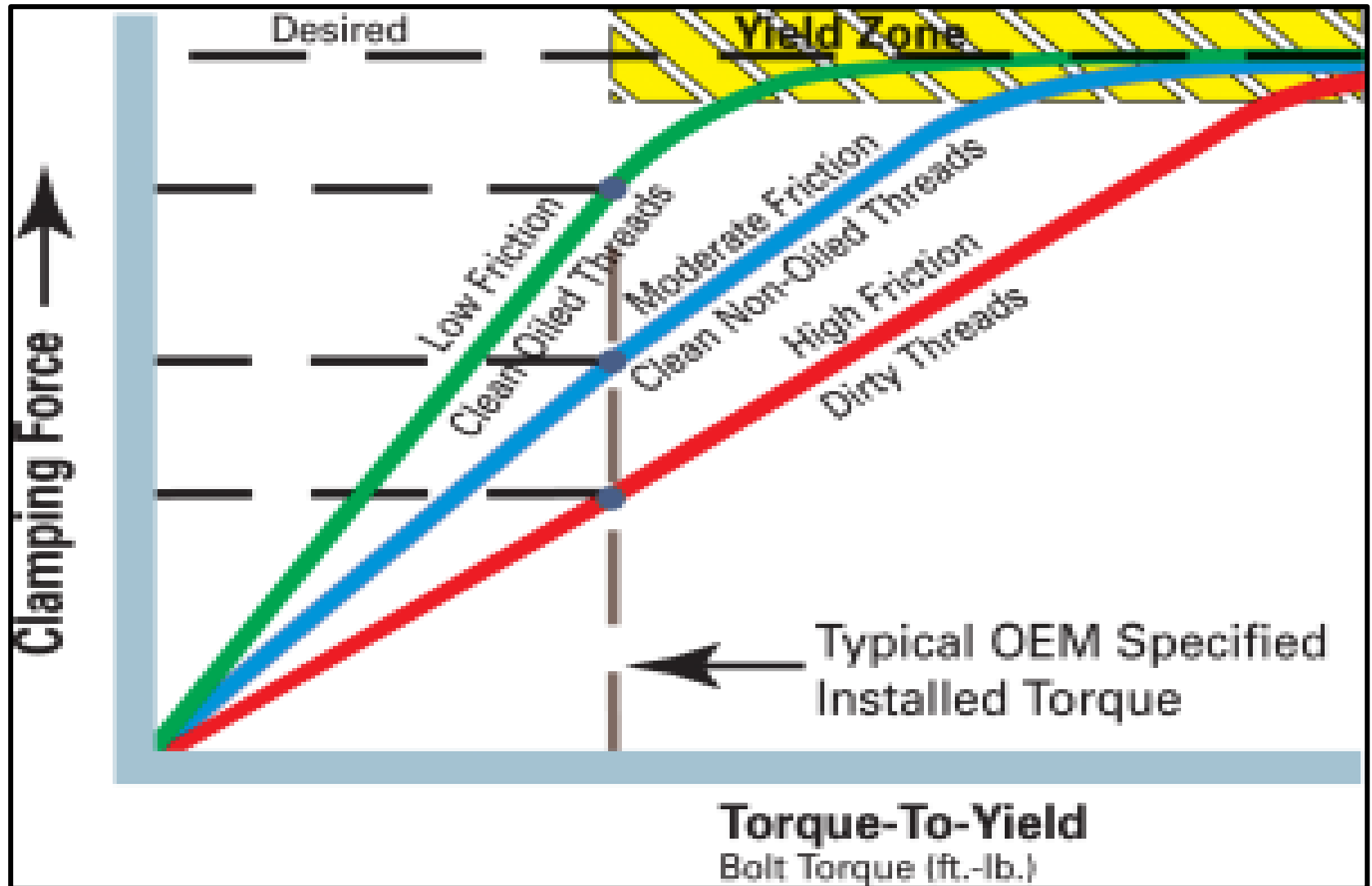


Table 8-2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC			Fine Series—UNF		
		Threads per Inch N	Tensile-Stress Area A_t in ²	Minor-Diameter Area A_r in ²	Threads per Inch N	Tensile-Stress Area A_t in ²	Minor-Diameter Area A_r in ²
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

*This table was compiled from ANSI B1.1-1974. The minor diameter was found from the equation $d_r = d - 1.299\,038p$, and the pitch diameter from $d_p = d - 0.649\,519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Table 8-1







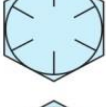

Diameters and Areas of
Coarse-Pitch and Fine-
Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series			Fine-Pitch Series		
	Pitch p mm	Tensile- Stress Area A_t mm ²	Minor- Diameter Area A_r mm ²	Pitch p mm	Tensile- Stress Area A_t mm ²	Minor- Diameter Area A_r mm ²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
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
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

*The equations and data used to develop this table have been obtained from ANSI B1.1-1974 and B18.3.1-1978. The minor diameter was found from the equation $d_r = d - 1.226\ 869p$, and the pitch diameter from $d_p = d - 0.649\ 519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Table 8-9








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	$\frac{1}{4}$ – $\frac{3}{4}$	55	74	57	Low or medium carbon	
	$\frac{7}{8}$ – $1\frac{1}{2}$	33	60	36		
4	$\frac{1}{4}$ – $1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$ –1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
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	

*Minimum strengths are strengths exceeded by 99 percent of fasteners.

Table 8-11

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

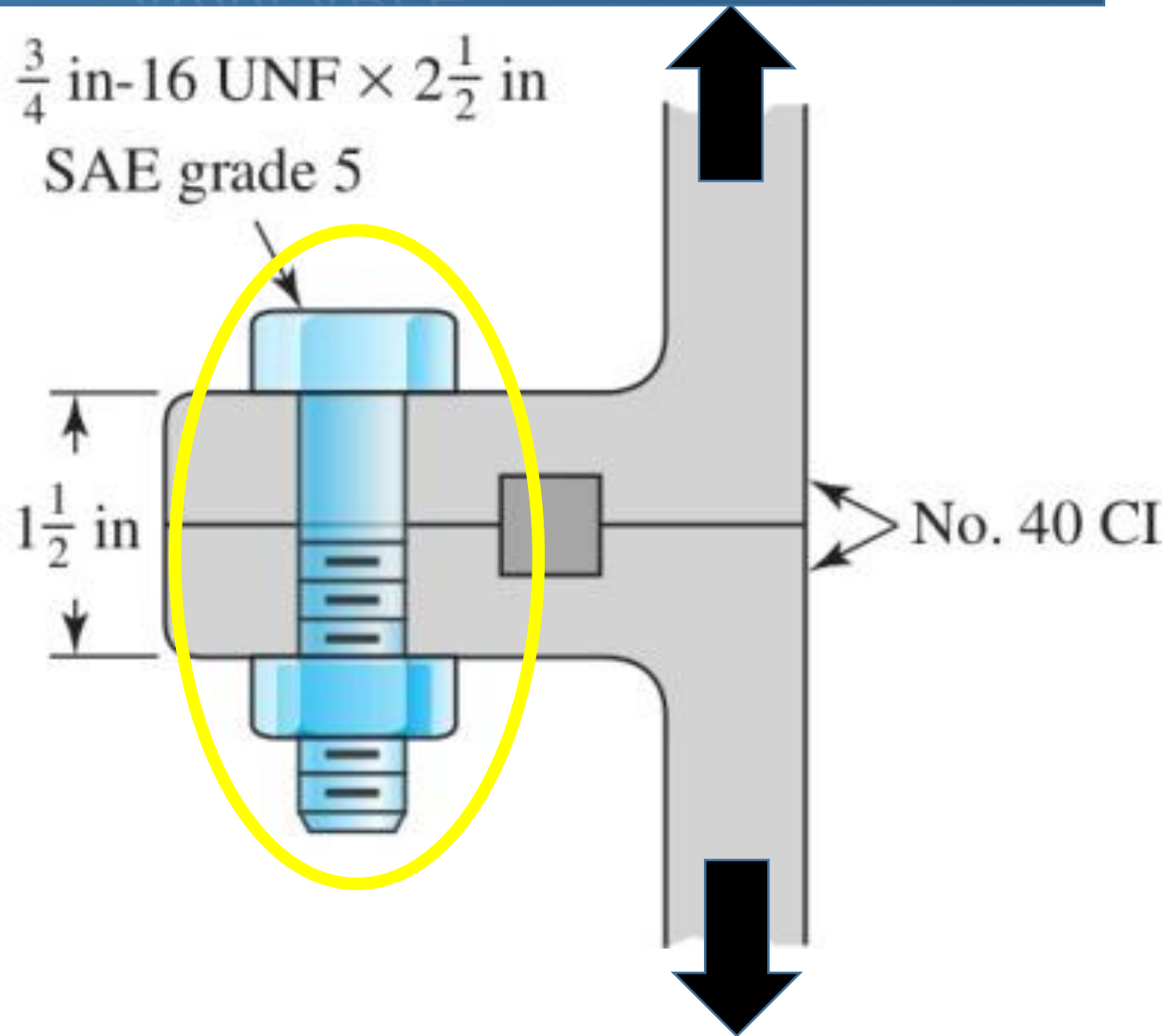
Property Class	Size Range, Inclusive	Minimum Proof Strength,* MPa	Minimum Tensile Strength,* MPa	Minimum Yield Strength,* MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600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



*Minimum strengths are strengths exceeded by 99 percent of fasteners.

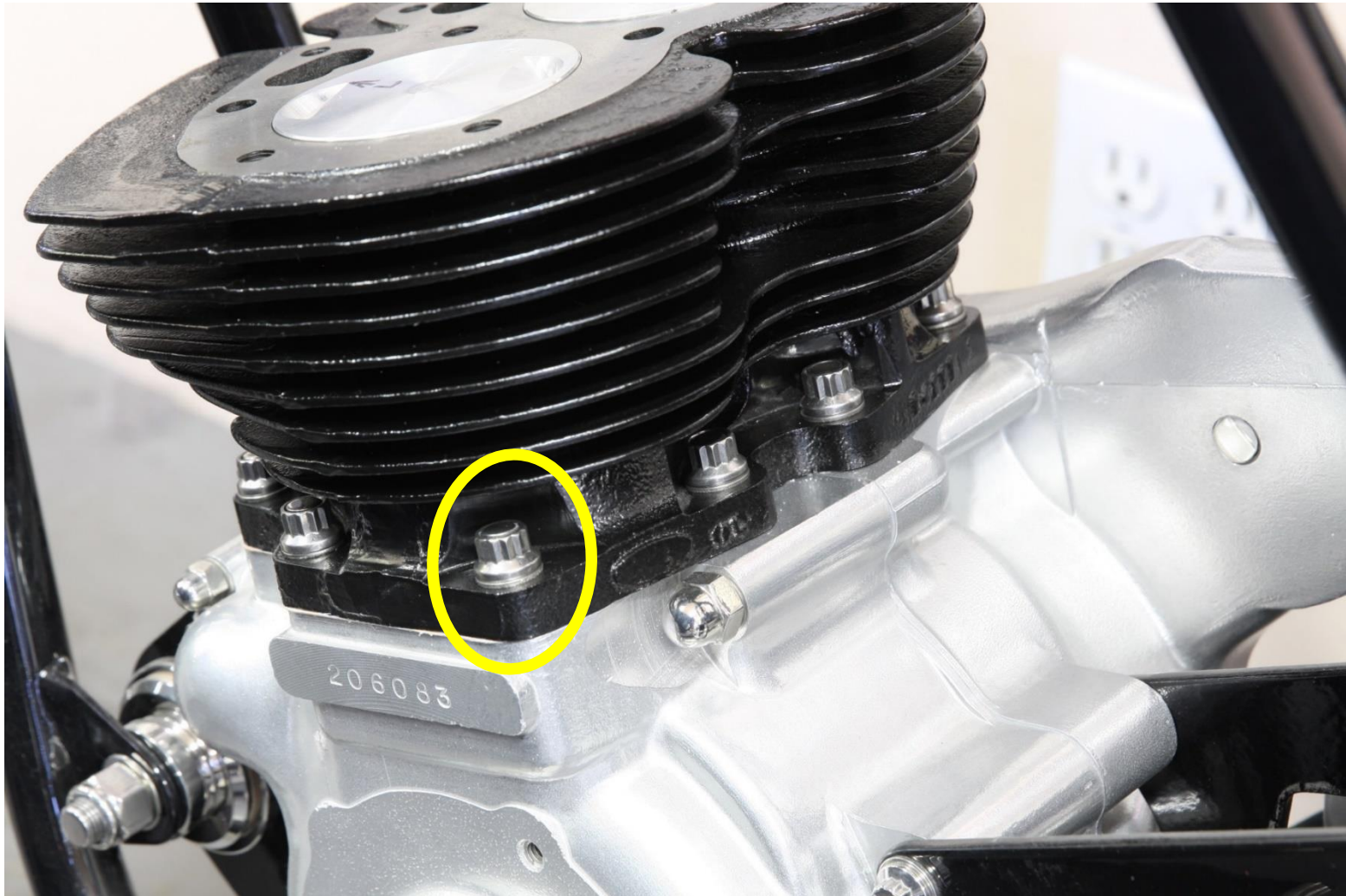
SHORT FASTENERS – NOT MUCH STRETCH AVAILABLE

Problem 8-60

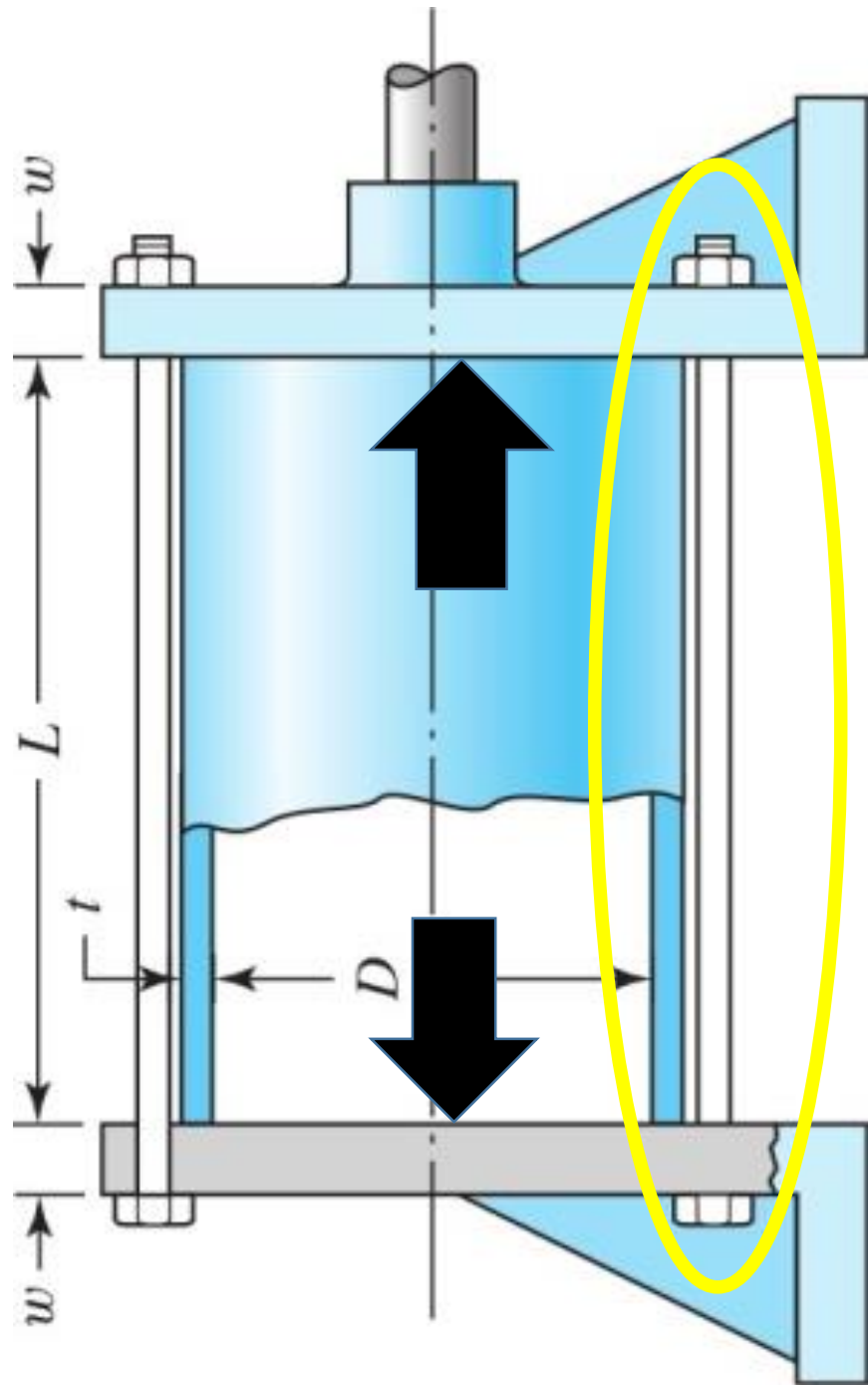


Air Cooled Engine Cylinders

Short Fastener / Bolt Variant (*small amount of fastener stretch available*)

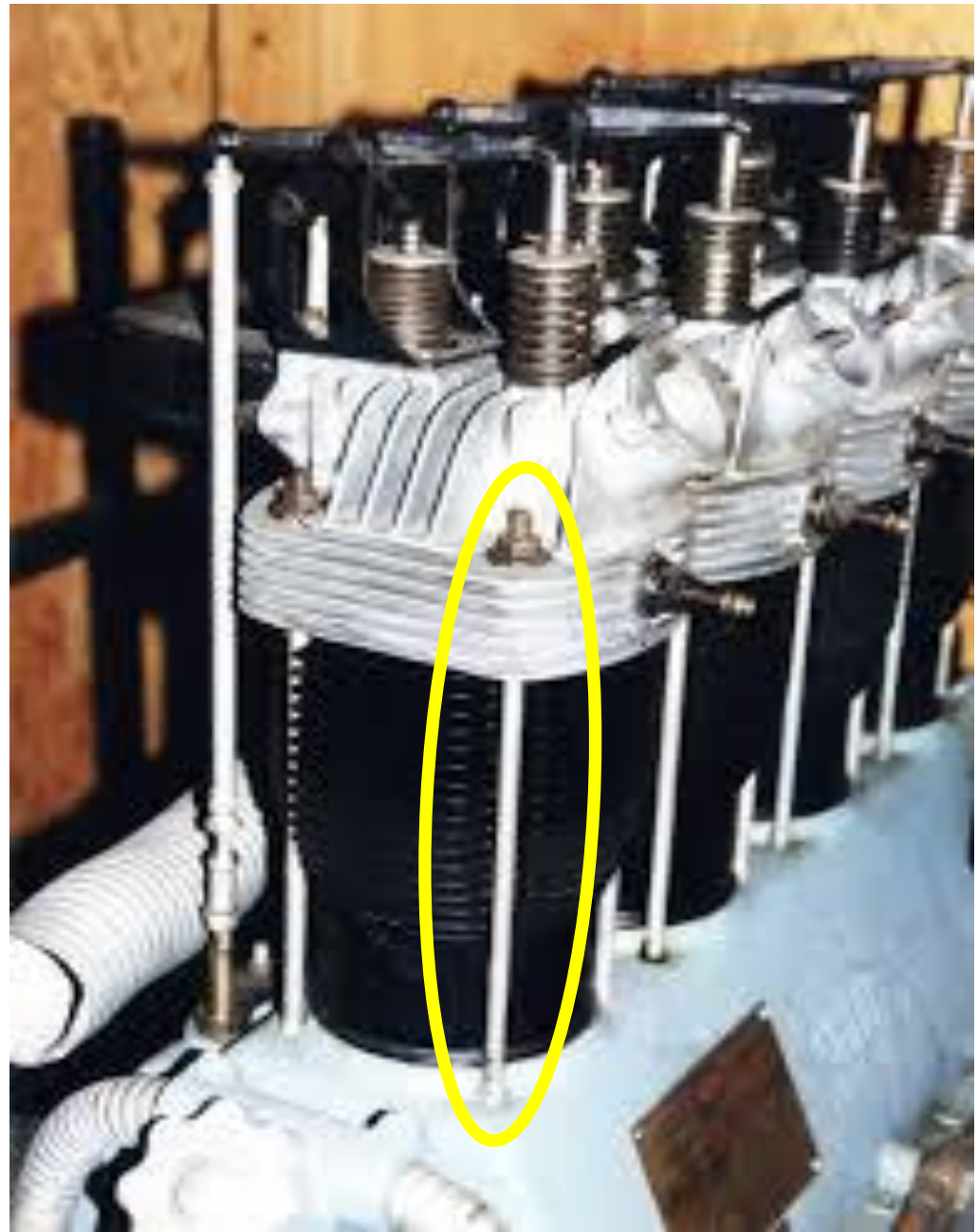


CYLINDER
STUDS
- MORE
STRETCH
AVAILABLE



Air Cooled Engine Cylinders

Long Fastener / Stud Variant
(lots of stretch)



Sample Calculation: *the importance of bolt stretch*

- General approximation (GA) for the clamp load to reliably seal a gasket is 3 times the lift-off force which is trying to separate the head from the block.
- The lift-off force for a 4.250" bore race motor with 1,400 psi combustion firing pressure is 19,861 lbs.
- Clamping Force is $19,861 \times 3 = \underline{59,583 \text{ lbs.}}$ *per cylinder (that is about 30 tons).*
- 5-bolt pattern requires 11,917 lbs. of clamping force per bolt;
- 6-bolt pattern requires 9,930 lbs. of clamping force per bolt

....to reliably seal the engine and prevent head gasket leaks



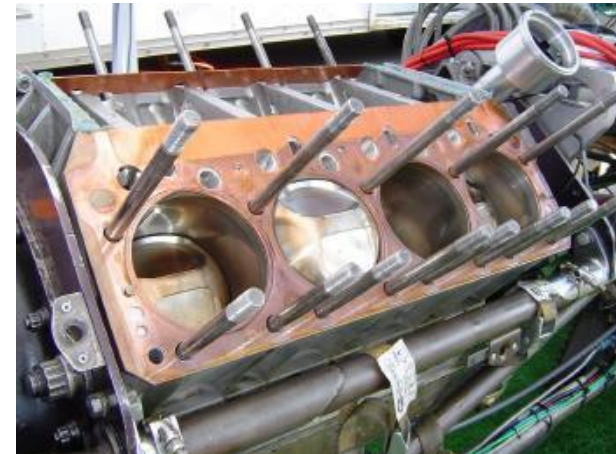
Sample Calculation: *the importance of bolt stretch*

So, let's select a 6-bolt pattern and look at different diameter fasteners and how much they stretch when properly torqued:

- 7/16"Ø bolt stretches .070" = 11,900 lbs. clamp load;
- 9/16"Ø bolt stretches .030" = 11,900 lbs. clamp load;

BUT a typical gasket installed is .045" thick but it relaxes about 25% when clamped, for a loss in thickness of about .011" which effectively reduces bolt stretch and "loosens" the bolts;

So, the 7/16"Ø bolt loses 1/7 of the load (.070/.011), leaving $(11,600 \times 6/7) = 10,200$ lbs. for clamping, but a 9/16"Ø bolt loses 1/3 of the load (.030"/0.11"), leaving only 7,933 lbs for clamping which is NOT ENOUGH to reliably seal the engine (recall that we needed about 9930 lb for a good seal).



Sample Calculation: *the importance of bolt stretch*

- General approximation (GA) for clamp load to reliably seal a gasket is 3X the lift-off force.
- Lift-off force for a 4.250" bore race motor with 1,400 psi combustion firing pressure is 19,861 lbs.
- GA is $19,861 \times 3 = \underline{59,583 \text{ lbs.}}$ per cylinder (about 30 tons).
- With a 5-bolt pattern, 11,917 lbs. of force is needed per bolt and with a 6-bolt pattern, 9,930 lbs. of clamping force is needed per bolt to reliably seal the engine.

So, let's select a 6-bolt pattern and look at different sized fasteners:

- 7/16"Ø bolt stretched .070" = 11,900 lbs. clamping load;
 - 9/16"Ø bolt stretched .030" = 11,900 lbs. clamping load;
- BUT a typical gasket installed is .045" thick and it relaxes about 25% when clamped, for a net thickness loss of about .011" which effectively reduces bolt stretch and "loosens" the bolts;

So, the 7/16"Ø bolt loses 1/7 of the load, leaving 10,200 lbs. for clamping but a 9/16"Ø bolt loses fully 1/3 of the load, leaving only 7,933 lbs. for clamping which is NOT ENOUGH to reliably seal the engine.



Brake Disc Mounting

*(shear loaded
short
fasteners)*



Thanks very much for your
attention...

Any questions?