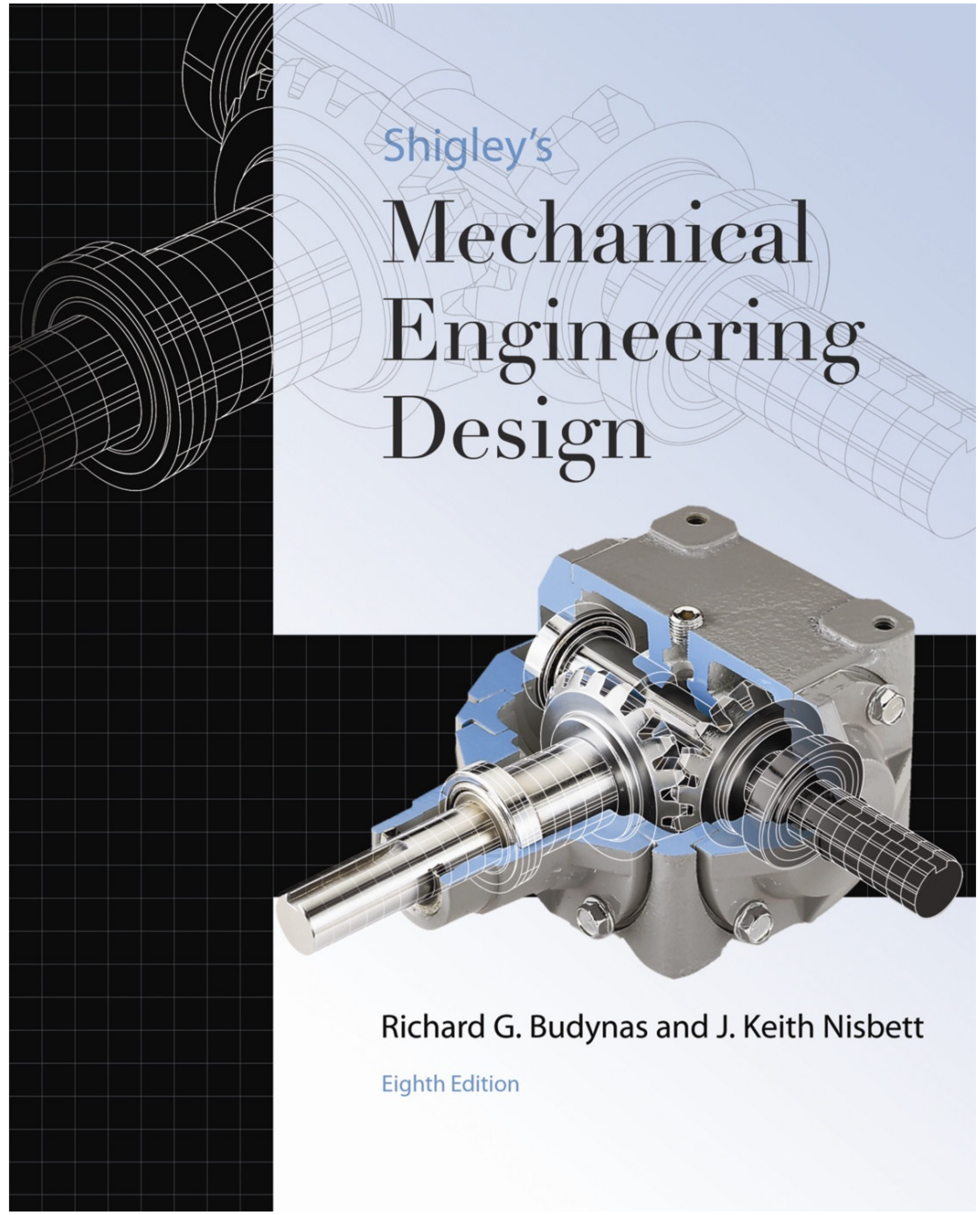
Threaded Fasteners

Lecture Notes
Prepared by H. Orhan YILDIRAN



LECTURE NOTES- MECE 304 Mechanical Machine Elements

Chapter 6- Threaded fasteners

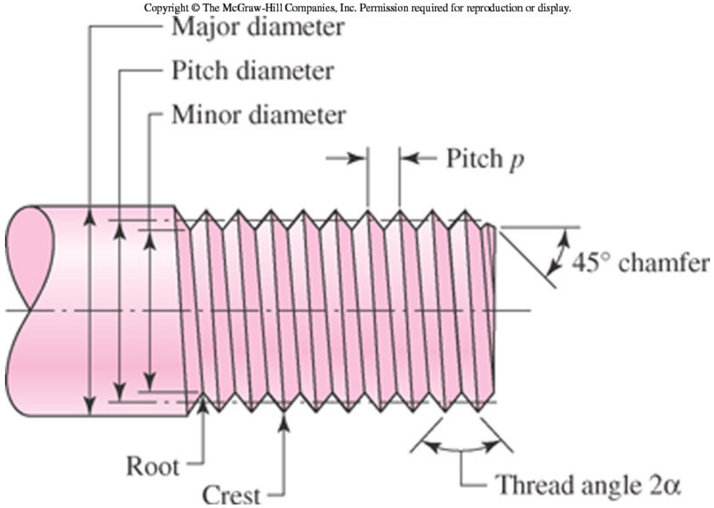
(Notes from: *Chapter 8*, *Budynas R.G., Nisbett J.K., Shigley's* Mechanical Engineering Design, *Mc Graw Hill, 8th Edition)*

Spring Semester 2009/2010

Halil Orhan YILDIRAN, MS

8-1 Thread Standards And Definitions

Fig 8-1 Terminology of Screw threads is



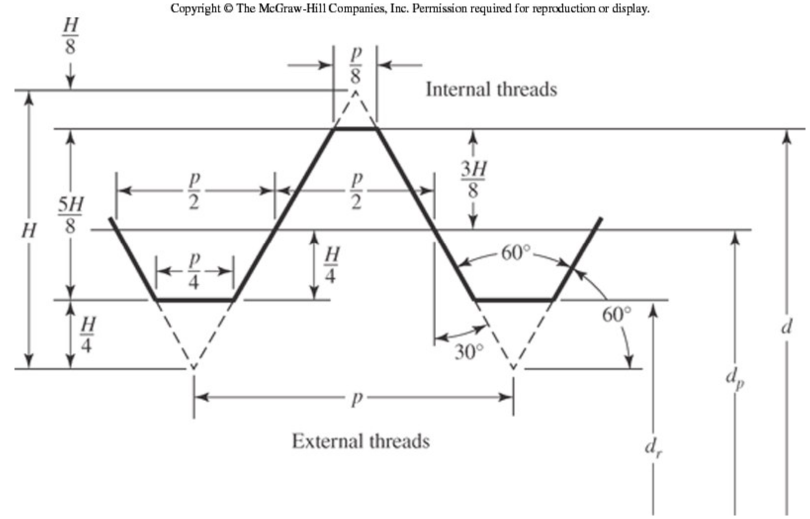


Fig 8-2 Basic profile for metric M and MJ threads

Thread Standards And Definitions

The *pitch p is the distance between adjacent thread forms.*
The pitch line is located at one half of the height of the theoretical sharp v-thread profile.
The *major diameter d is the largest dia. of a screw thread.*
The *lead l is the distance the nut moves parallel to the screw axis when the* nut is given one turn.
For a *single-thread, the lead is the same as the pitch.*

Table 8–1

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

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm^2	Coarse-Pitch Series: Minor-Diameter Area A_r mm^2	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm^2	Fine-Pitch Series: 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

8-2 The Mechanics of Power screws

The power screw is a device used in machinery to change angular motion into linear motion. Square and Acme threads are used on screws when power is transmitted.

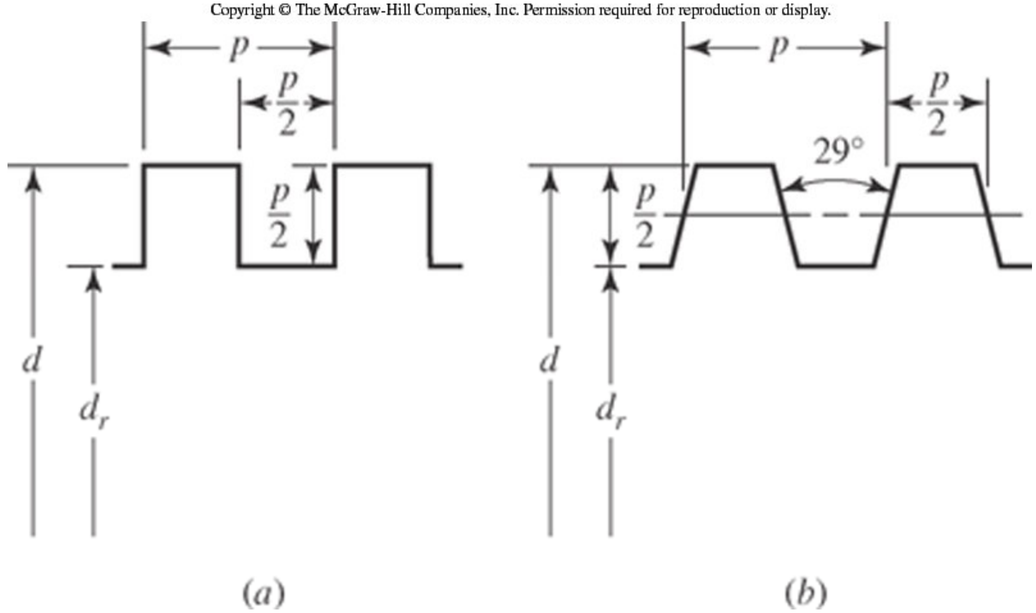


Fig 8-3 Power Screws (a) Square thread,(b) Acme thread

The Mechanics of Power screws

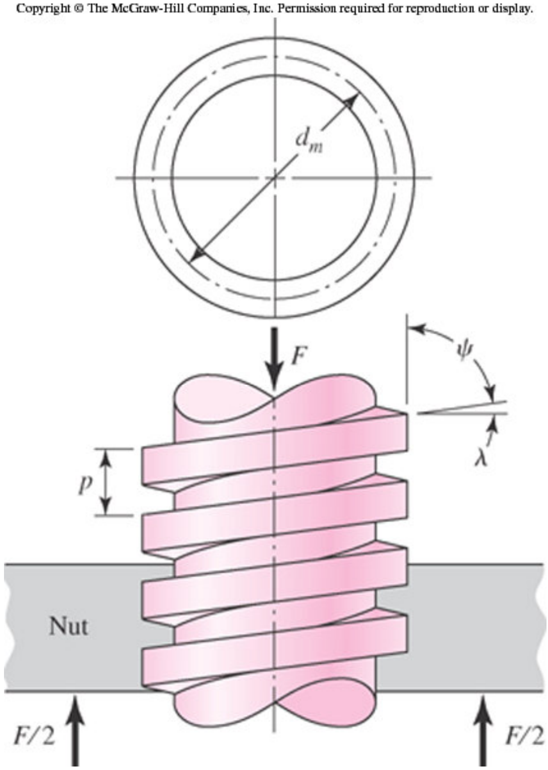


Fig 8-5 Portion of a Power Screw

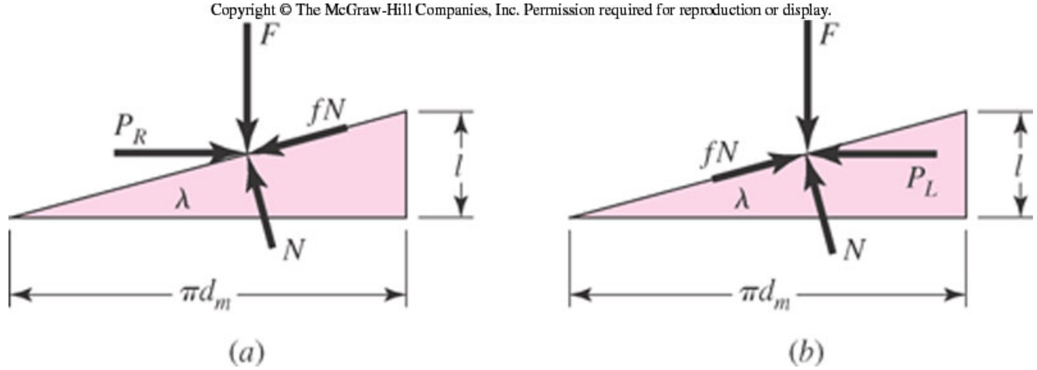


Fig 8-6 Power Screw force diagrams (a) Lifting the load (b) Lowering the load

The Mechanics of Power screws

For raising the load the torque is

$$T_R = \frac{Fd_m}{2}\left(\frac{l+\pi.fd_m}{\pi.d_m - f.l}\right) \qquad (8\text{-}1)$$

For lowering the load the torque is

$$T_L = \frac{Fd_m}{2}\left(\frac{\pi.fd_m - l}{\pi.d_m + f.l}\right) \qquad (8\text{-}2)$$

Where

dm= mean diameter

P=pitch

f=friction coeefficient

l=lead of thread

The condition of self locking in eq 8-1

$$\pi.fd_m > l \qquad f > \tan\lambda \qquad (8\text{-}3)$$

The efficiency is;

$$e = \frac{T_0}{T_R} = \frac{F.l}{2\pi.T_R} \qquad (8\text{-}4)$$

For raising the load for threads other than square threads with thread angle 2 α

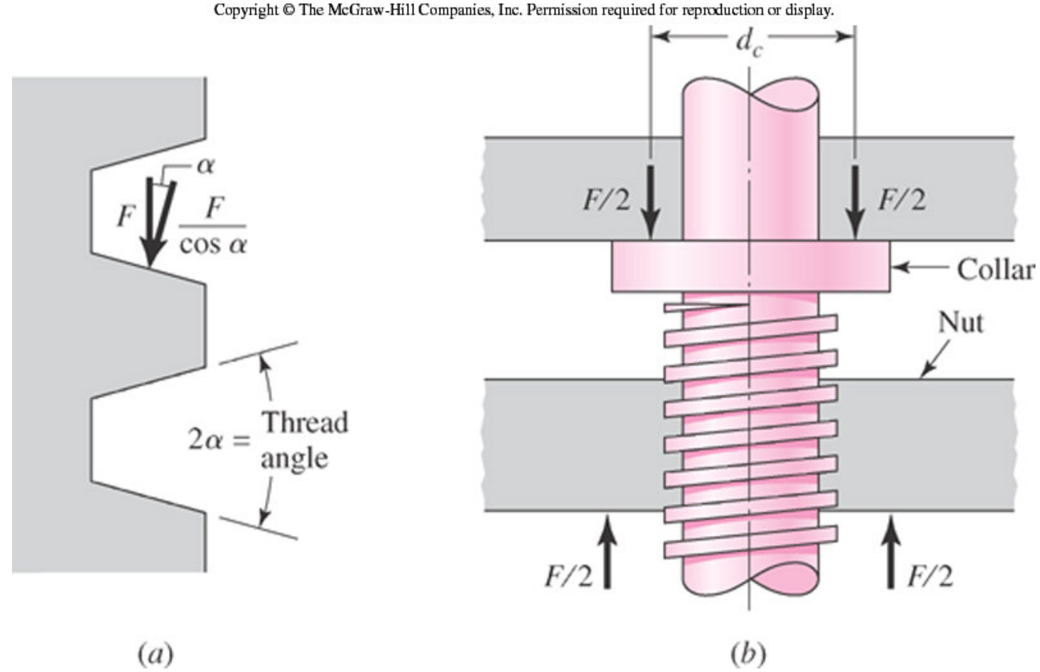
$$T_R = \frac{Fd_m}{2}\left(\frac{l + \pi.fd_m \sec\alpha}{\pi.d_m - f.l\sec\alpha}\right) \qquad (8\text{-}5)$$

The torque at the collar of the power screw

$$Tc = \frac{Ff_c.d_c}{2} \qquad (8\text{-}6)$$

Where fc is the friction coeefficient at the collar. Eqs 8-5 and 8-6 are added together to find total torque.

Fig-8-7 (a) Normal thread force is increased because of angle α
(b)Thrust collar has frictional diameter fc



The maximum nominal shear stress;

$$\tau = \frac{16T}{\pi . d_r^3} \qquad (8\text{-}7)$$

The axial stress in the body of the screw

$$\sigma = \frac{F}{A} = \frac{4F}{\pi . d_r^2} \qquad (8\text{-}8)$$

The bearing stress

$$\sigma = -\frac{F}{\pi . d_m . n_t . p/2} = -\frac{2F}{\pi . d_m . n_t . p} \qquad (8\text{-}10)$$

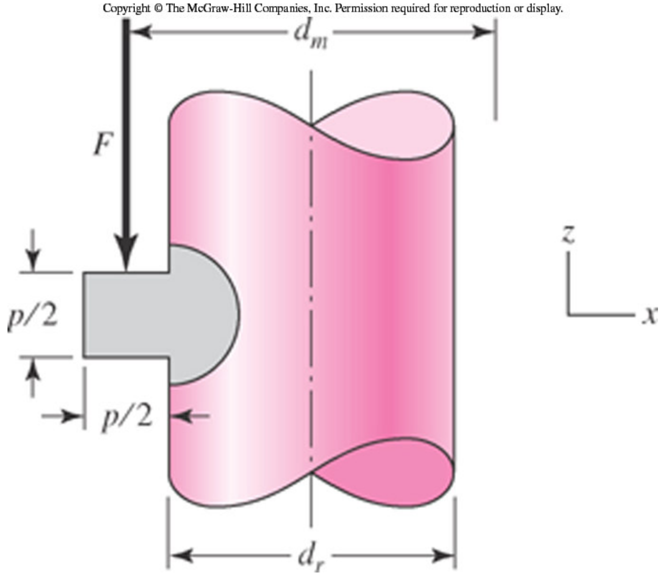


Fig 8-8 Geometry of square thread in finding bending and transverse shear stress at the thread root

The bending stress at the root of the thread

$$\frac{I}{c} = \frac{\pi . d_r . n_t (p/2)^2}{6} = \frac{\pi}{24} d_r . n_t . p^2 \qquad M = \frac{Fp}{4}$$

$$\sigma_b = \frac{M}{I/c} = \frac{Fp}{4} \frac{24}{\pi . d_r . n_t . p^2} = \frac{6F}{\pi . d_r . n_t . p} \qquad (8\text{-}11)$$

The transverse shear stress at the center of the tooth of the thread due to load F

$$\tau = \frac{3V}{4A} = \frac{3}{2} \frac{F}{\pi . d_r . n_t . p/2} = \frac{3F}{\pi . d_r . n_t . p} \qquad (8\text{-}12)$$

The Von Mises stresses at the top of the root plane with the coordinate system in Fig 8-8

$$\sigma_x = \frac{6F}{\pi . d_r . n_t . p} \qquad \tau_{xy} = 0$$

$$\sigma_y = 0 \qquad \tau_{yz} = \frac{16T}{\pi . d_r^3}$$

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

Note:
Threads do not carry equal loads!

8-4 Joints-Fastener stiffness

Bolts clamp parts together. Twisting the nut stretches the bolt and produces clamping force. This clamping force is called pretension or bolt preload.

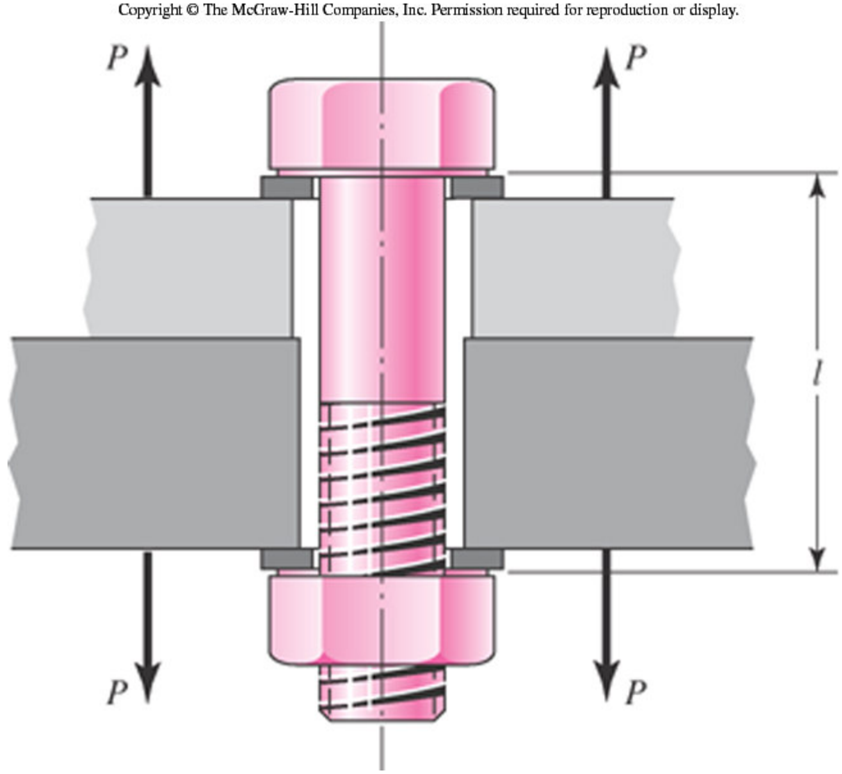


Fig 8-13 .A bolted connection loaded in tension by forces P

Stiffness constant of the bolt is equivalent to the stiffness of two springs in series.

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} \qquad \text{8-17}$$

A_t=tensile stress area (table 8-1)
l_t=length of threaded portion of grip
A_d= major diameter area of fastener
l_d=length of unthreaded portion of grip
kb=estimated stiffness of bolt
E=Modulus of elesticity

8-5 Joints-Member stiffness

Bolts clamp parts together. Twisting the nut stretches the bolt and produces clamping force and compresses the member. Member stiffness must also be known in order to study the assembled parts.

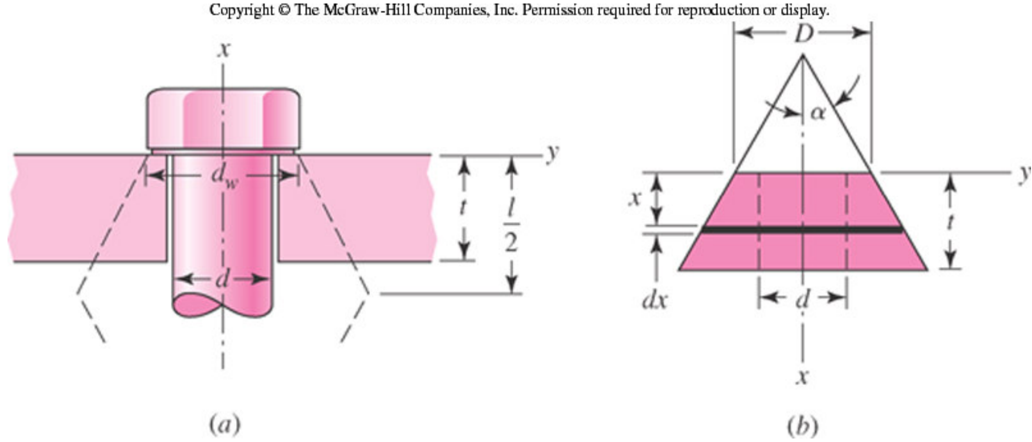


Fig 8-15 .Compression of member with the equivalent elastic properties represented by a frustrum of a cone. L is the grip length

Using the grip lenth l=2t we have the stiffness of members as:

$$k_m = \frac{\pi E d \tan\alpha}{2\ln\frac{(l\tan\alpha + d_w - d)(d_w + d)}{(l\tan\alpha + d_w + d)(d_w - d)}} \qquad 8\text{-}21$$

d_w=diameter of washer (=D)
d=diameter of bolt
k_m=stiffness of members
We take as α=30 ° for steel, cast iron and aluminum.

8-6 Bolt Strength

Proof load (*Fp): The maximum load that a bolt* can withstand without acquiring a permanent set.

Proof Strength (Sp): The strength value corresponding to proof load (*Sp= Fp / At)*

Preload (Fi): Initial tensile force for tightening screws and nut-bolt assemblies:

8-7 Tension joints –External load

F_i=preload

P=external tensile load

P_b=portion of P taken by bolt

P_m=portion of P taken by members

$F_b=P_b+F_i$ resultant bolt load

$F_m=P_m-F_i$=resultant load on members

C= fraction of external load P carried by bolt (stiffness constant of joint)

1-C=fraction of external load P carried by members

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

Pb=CP
Pm=(1-C)P

Fb=Pb+Fi=CP+Fi   Fm<0   8-24

The resultant load on the connected member is

Fm=Pm-Fi=(1-C)P-Fi   Fm<0   8-25

8-8 Relating bolt Torque to bolt tension

$$K=\left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda+f.\sec\alpha}{1-f.\tan\lambda\sec\alpha}\right)+0.625f_c \qquad (8\text{-}26)$$

$$T=KF_i d \qquad (8\text{-}27)$$

Where K is torque coefficient
On the average when f and fc are 0.15 , K=0.20 regardless of the size

$$T=0.2F_i d \qquad (8\text{-}27a)$$

8-9 Statically loaded tension joint

$$n = \frac{S_p A_t - Fi}{CP} \qquad \text{8-28}$$

$$n_o = \frac{Fi}{(1-C)P} \qquad \text{8-29}$$

$$F_i = \begin{cases} 0.75F_p & for.reused.fasteners \\ 0.90F_p & for.permanent.connections \end{cases} \qquad \text{8-30}$$

$F_p = A_t S_p$

S_p is obtain from Table 8-11 or by $S_p = 0.85 S_y$

n=load factor

n_o=factor of safety against separation of joint

S_p=proof strength, F_p=Proof load

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	4.6
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

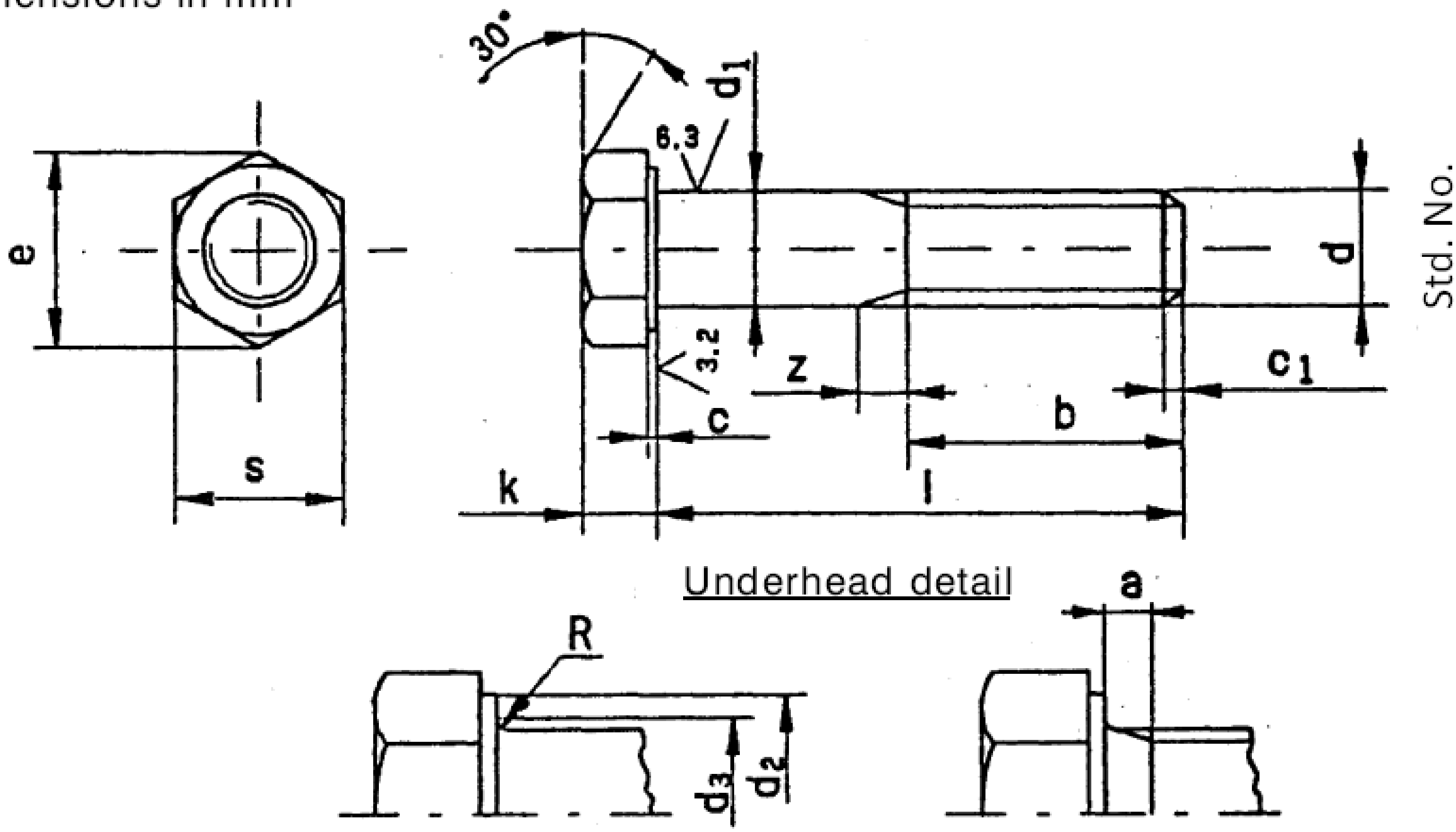
*The thread length for bolts and cap screws is

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

where L is the bolt length. The thread length for structural bolts is slightly shorter than given above.

†Minimum strengths are strength exceeded by 99 percent of fasteners.

Dimensions in mm



Example of designation

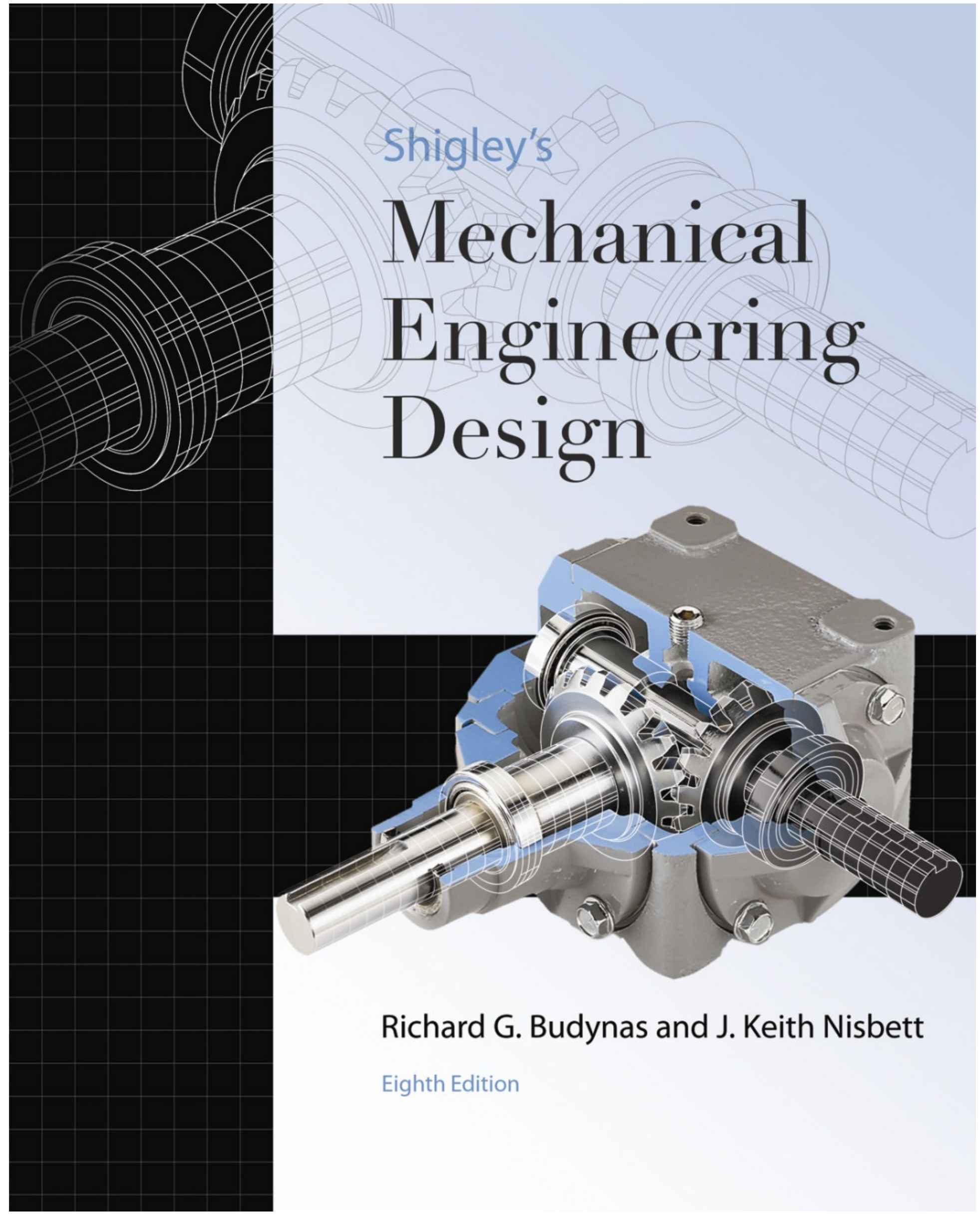
Part No – Std. No. – Hexagon Bolt M10 x 1.25 x 50 – Mat. 8.8

Fig. Dimensions of a Bolt. Designation of M10x1.25x l=50 mm long bolt

Springs

Lecture Notes

Prepared by H. Orhan YILDIRAN



LECTURE NOTES- MECE 304 Mechanical Machine Elements

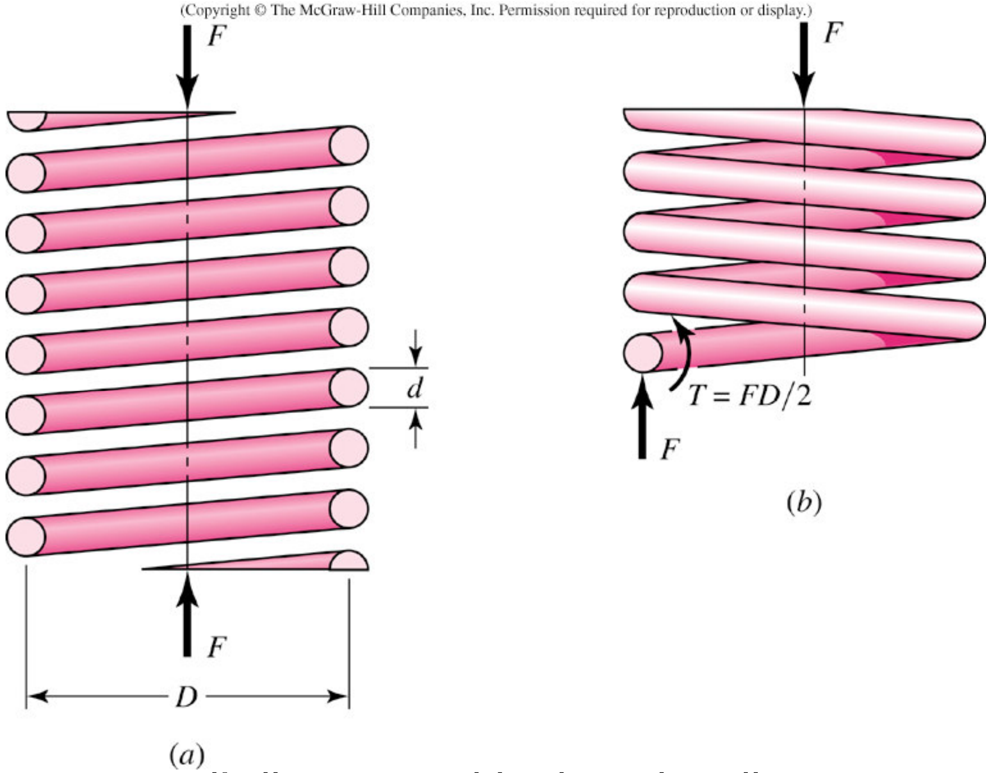
Chapter 6- Springs

(Notes from: *Chapter 10*, *Budynas R.G., Nisbett J.K., Shigley's* Mechanical Engineering Design*, Mc Graw Hill, 8th Edition)*

Spring Semester 2009/2010

Halil Orhan YILDIRAN, MS

10-1 Stresses in Helical Springs



Where D= mean coil diameter, d is the wire diameter

A compression spring is an open-coil helical spring that offers resistance to a compressive force applied axially. They provide wide range of load deflection curves.

Stresses in helical springs can be found by

$$\tau = \frac{8FD}{\pi.d^3} + \frac{4F}{\pi.d^2} \qquad (10\text{-}1)$$

The spring index C is defined by

$$C = \frac{D}{d} \qquad (10\text{-}2)$$

Rearranging Eq 10-1

$$\tau = Ks\frac{8FD}{\pi.d^3} \qquad (10\text{-}3)$$

Where Ks is the shear stress correction factor and defined by (See Eq. 10-6)

$$Ks = \frac{2C+1}{2C} \qquad (10\text{-}4)$$

10-2 Curvature Effect

There are various correction factors for shear. We will use below value for correction

$$K_B = \frac{4C+2}{4C-3} \qquad (10\text{-}6)$$

10-3 Deflection of helical springs

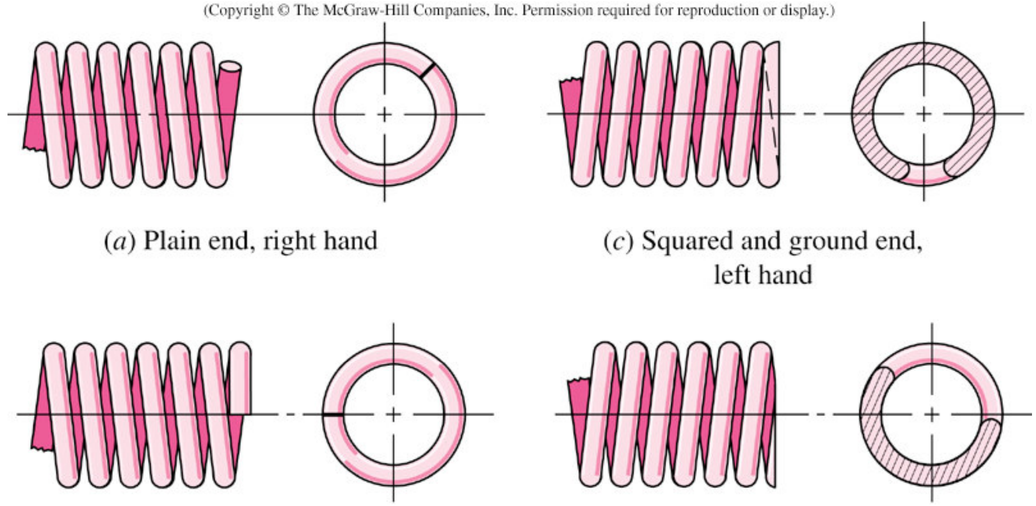
The slope of force vs deflection curve is called spring rate k. That is k=F/y and is defined by

$$k \doteq \frac{d^4 . G}{8D^3 N} \qquad (10\text{-}9)$$

Where
G= shear modulus
N= number of active coils

10-4 Compression springs

Generally 4 end types are used



(*a*) Plain end, right hand

(*c*) Squared and ground end, left hand

(*b*) Squared or closed end, right hand

(*d*) Plain end, ground, left hand

Fig 10-2 Types of ends for compression springs

Presetting of is a process to induce residual stress in the manufacture of compression springs

Table 10–1

Formulas for the Dimensional Characteristics of Compression-Springs. (N_a = Number of Active Coils)

Source: From *Design Handbook*, 1987, p. 32. Courtesy of Associated Spring.

	Type of Spring Ends			
Term	**Plain**	**Plain and Ground**	**Squared or Closed**	**Squared and Ground**
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

10-6 Spring Materials There are various spring materials
Tensile strength vs wire diameter is almost a straigth line for some materials when plotted on a log-log paper

$$S_{ut}=A/d^m \qquad (10\text{-}14)$$

We can find constants A and m using Table 10-4

For high tensile spring steels the allowable stress is

$$S_{sy}=S_{all}= 0.56\ S_{ut} \qquad (10\text{-}16)$$

Spring materials

Table 10–4

Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Source: From *Design Handbook*, 1987, p. 19. Courtesy of Associated Spring.

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire†	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire‡	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire§	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‖	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire#	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire**	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

*Surface is smooth, free of defects, and has a bright, lustrous finish.

†Has a slight heat-treating scale which must be removed before plating.

‡Surface is smooth and bright with no visible marks.

§Aircraft-quality tempered wire, can also be obtained annealed.

‖Tempered to Rockwell C49, but may be obtained untempered.

#Type 302 stainless steel.

**Temper CA510.

Table 10–5

Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Torsion	Diameter d, in	E Mpsi	GPa	G Mpsi	GPa
Music wire A228	65–75	45–60	<0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			>0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	<0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			>0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Chrome-vanadium A231	88–93	65–75		29.5	203.4	11.2	77.2
A232	88–93			29.5	203.4	11.2	77.2
Chrome-silicon A401	85–93	65–75		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3
Phosphor-bronze B159	75–80	45–50		15	103.4	6	41.4
Beryllium-copper B197	70	50		17	117.2	6.5	44.8
	75	50–55		19	131	7.3	50.3
Inconel alloy X-750	65–70	40–45		31	213.7	11.2	77.2

*Also includes 302, 304, and 316.

Note: See Table 10–6 for allowable torsional stress design values.

Table 10–6

Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications

Source: Robert E. Joerres, "Springs," Chap. 6 in Joseph E. Shigley, Charles R. Mischke, and Thomas H. Brown, Jr. (eds.), *Standard Handbook of Machine Design*, 3rd ed., McGraw-Hill, New York, 2004.

Material	Maximum Percent of Tensile Strength: Before Set Removed (includes K_W or K_B)	Maximum Percent of Tensile Strength: After Set Removed (includes K_s)
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

10-7 **Helical Compression Spring Design for Static Service**

$$F_s = (1+\xi)F_{\max} \qquad (10\text{-}17)$$

Where Fs is the force bringing the spring to solid heigth, and max force be limited to 7/8 of Fs.

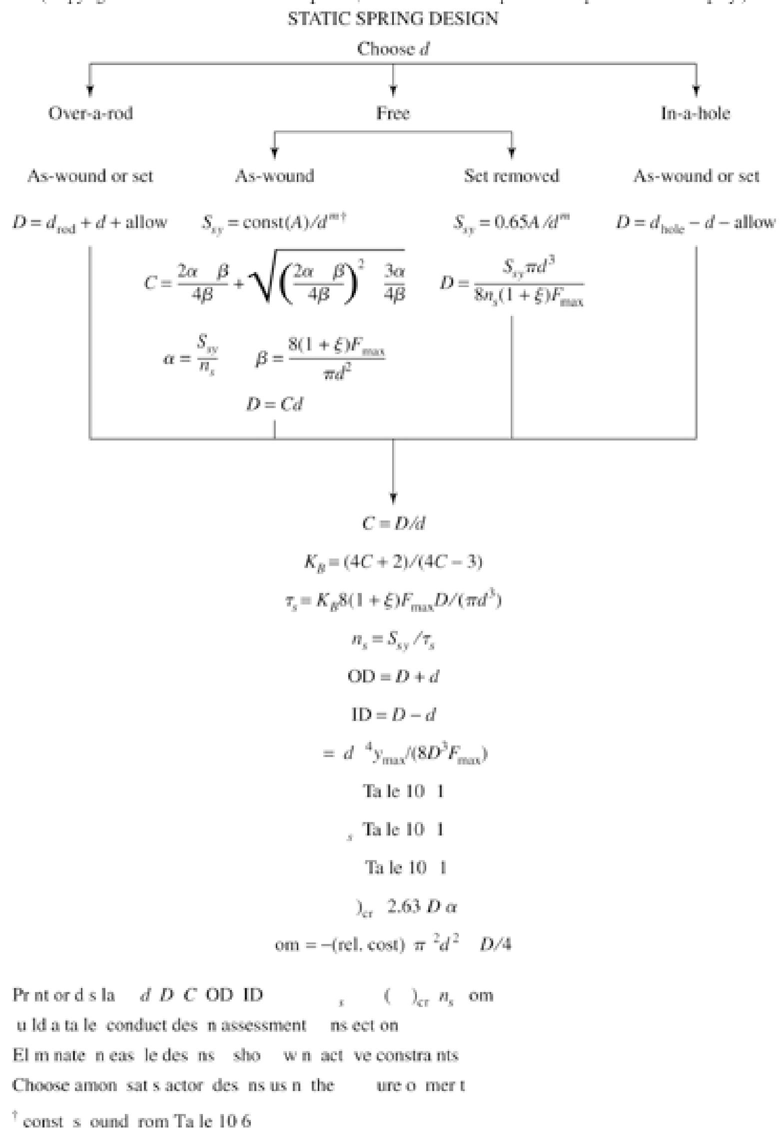
For F_{max} to be less than 7/8 of F_s we have;
the fractional overrun to closure $\xi \geq 0.15$

Recommended design conditions

$$4 \leq C \leq 12 \qquad (10\text{-}18)$$
$$3 \leq Na \leq 15 \qquad (10\text{-}19)$$
$$\xi \geq 0.15 \qquad (10\text{-}20)$$
$$ns \geq 1.2 \qquad (10\text{-}21)$$

Where ns is factor of safety at solid height

Fig 10-3 Helical compression springs design flowchart



10-11 Extension Springs

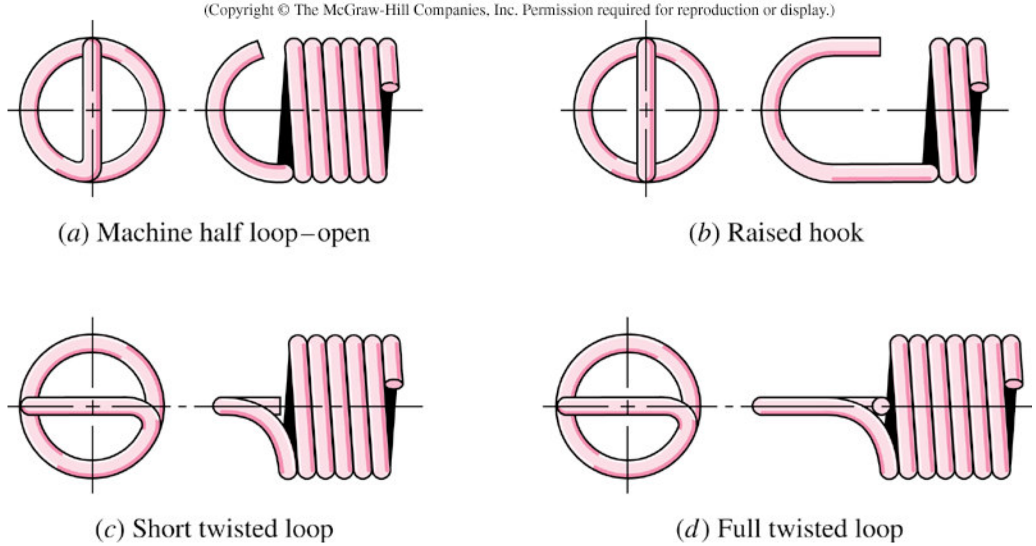
$$\sigma_A = F\left[(K)_A \frac{16D}{\pi.d^3} + \frac{4}{\pi.d^2}\right] \qquad (10\text{-}34)$$

$$(K)_A = \frac{4C_1^2 - C_1 - 1}{4C_1(C_1 - 1)} \qquad C_1 = \frac{2r_1}{d} \qquad (10\text{-}35)$$

$$\tau_B = (K)_B \frac{8Fd}{\pi.d^3} \qquad (10\text{-}36)$$

$$(K)_B = \frac{4C_2 - 1}{4C_2 - 4} \qquad C_2 = \frac{2r_2}{d} \qquad (10\text{-}37)$$

Fig 10-6 Types of ends used on extension springs

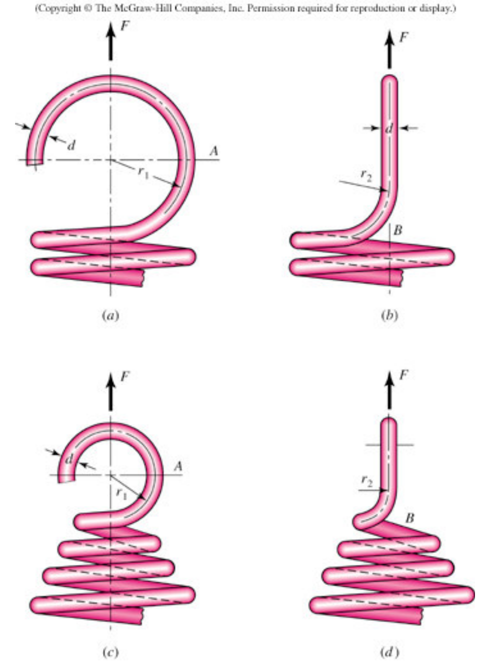


(a) Machine half loop–open (b) Raised hook

(c) Short twisted loop (d) Full twisted loop

An extension spring is an open-coil helical spring that is designed to offer resistance to a tensile force applied axially. Unlike compression springs, they have specially designed hooks at both ends. They provide wide range of load-deflection curves. Standard springs has constant diameter and pitch, thus providing a constant spring rate.

Fig 10-7 Ends for extension springs.
(a) Usual design; stress at A is due to combined axial force and bending moment. (b) Side view of part *a; stress is mostly* torsion *at B. (c) Improved* design; stress at *A is due to* combined *axial force and* bending moment. *(d) Side* view of part c; stress *at B is* mostly torsion



Note: Radius r_1 is in the plane of the end coil for curved beam bending stress. Radius r_2 is at a right angle to the end coil for torsional shear stress.

Fig 10-9 Torsion springs
They are used in any application where torque is required, such as door hinges, automobile starters, etc

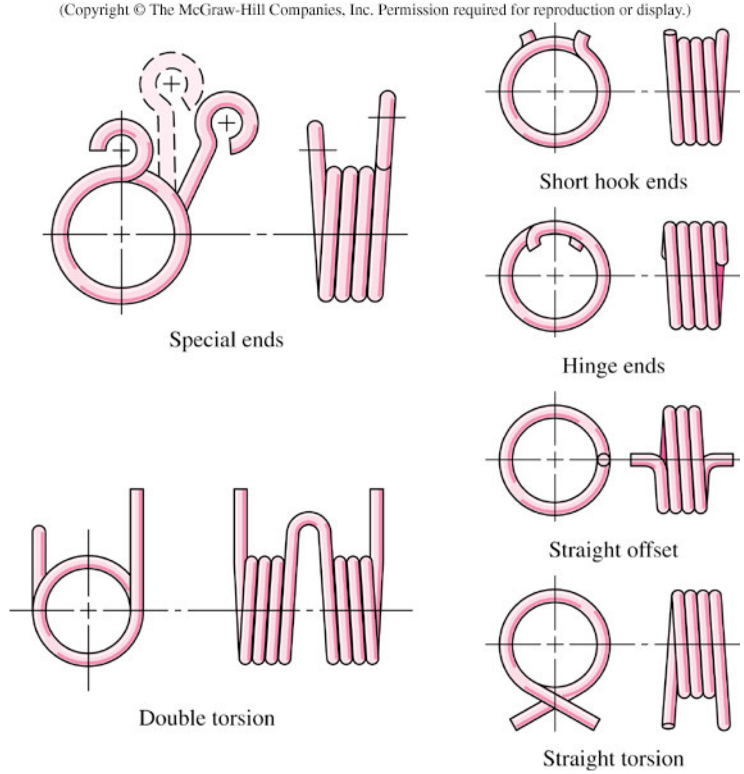
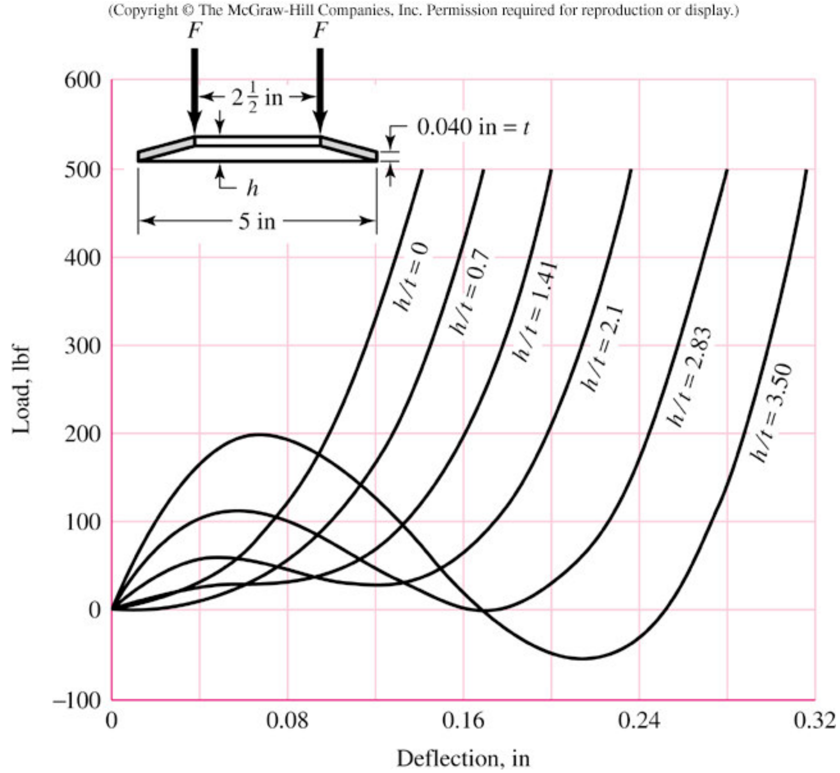
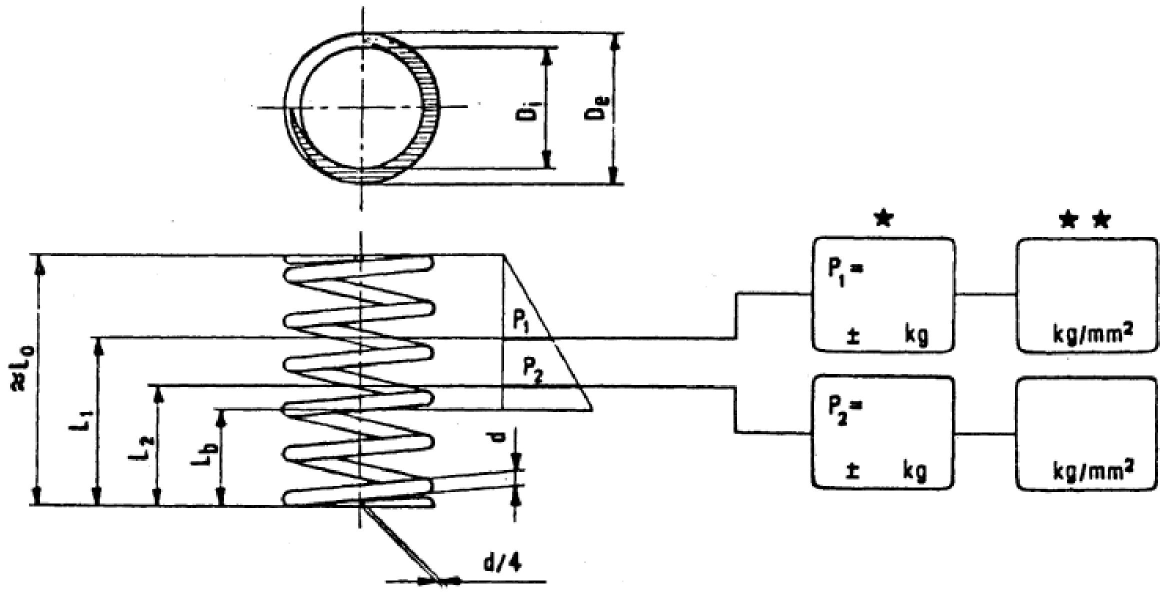


Fig 10-9 Belleville springs

*Belleville spring is a cone-disk spring. It occupies a small axial space.
*Their geometry can be engineered to produce highly non-linear characteristics which may display *negative stiffness. Belleville springs are* often used in stacks.
* Small ones are used as washers in threaded fastener subjected to dynamic loading.



GRAPHIC REPRESENTATION of a right helix spring with closed and ground ends.



Spring data			Tolerances
Wire diameter	**d**	mm	± mm
Outside diameter	$\mathbf{D_e}$	1) mm	± mm
Inside diameter	$\mathbf{D_i}$	1) mm	± mm
Number of active coils	**i**		–
Total number of coils	$\mathbf{i_t}$		±
Flexibility, give – in	**C**	mm/kg	–
Direction of helix		2)	–
Testing load	$\mathbf{P_c}$	kg	–
Inspection class: see P.S. 9.01344			3)
Percentage of springs to be tested			4) %