Keys, Couplings and Seals

Objectives:

- How attach power transmission components to shaft to prevent rotation and axial motion?

  Torque resistance: keys, splines, pins, weld, press fit, etc..

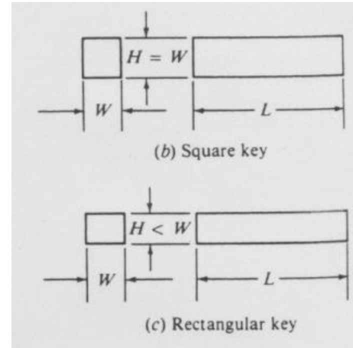
  Axial positioning: retaining rings, locking collars, shoulders machined into shaft, etc….

- What is the purpose of rigid and flexible couplings in a power transmission system?

- Specify seals for shafts and other types of machine elements.

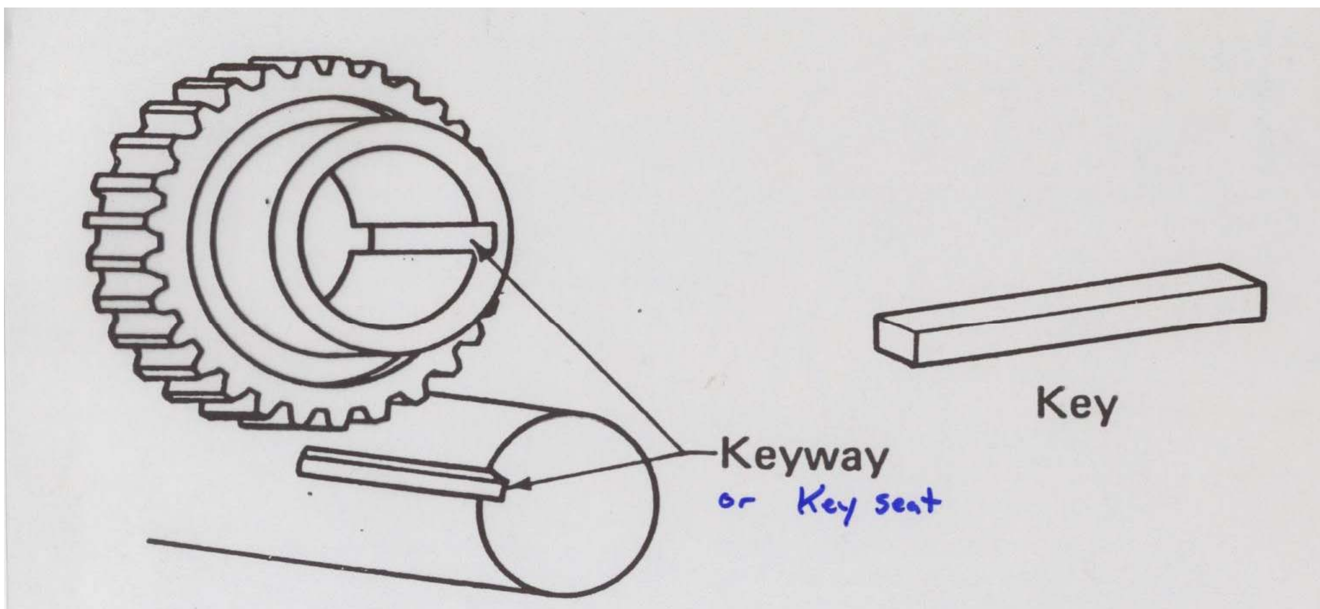
11.2 Keys

Most common for shafts up to 6.5” is the square and rectangular keys:



Advantages:

1. Cost effective means of locking the
2. Can replace damaged component
3. Ease of installation
4. Can use key as “fuse” – fails in shear at some predetermined torque to avoid damaging drive train.



A key and the keyways in a gear and shaft

Figure 11.1

Square and rectangular keys:

TABLE 11–1 Key size vs. shaft diameter

Nominal shaft diameter		Nominal key size		
			Height, *H*	
Over	To (incl.)	Width, *W*	Square	Rectangular
5/16	7/16	3/32	3/32	
7/16	9/16	1/8	1/8	3/32
9/16	7/8	3/16	3/16	1/8
7/8	1 1/4	1/4	1/4	3/16
1 1/4	1 3/8	5/16	5/16	1/4
1 3/8	1 3/4	3/8	3/8	1/4
1 3/4	2 1/4	1/2	1/2	3/8
2 1/4	2 3/4	5/8	5/8	7/16
2 3/4	3 1/4	3/4	3/4	1/2
3 1/4	3 3/4	7/8	7/8	5/8
3 3/4	4 1/2	1	1	3/4
4 1/2	5 1/2	1 1/4	1 1/4	7/8
5 1/2	6 1/2	1 1/2	1 1/2	1
6 1/2	7 1/2	1 3/4	1 3/4	1 1/2
7 1/2	9	2	2	1 1/2
9	11	2 1/2	2 1/2	1 3/4
11	13	3	3	2
13	15	3 1/2	3 1/2	2 1/2
15	18	4		3
18	22	5		3 1/2
22	26	6		4
26	30	7		5

Source: Reprinted from ANSI Standard B17.1-1967(R98), by permission of the American Society of Mechanical Engineers.

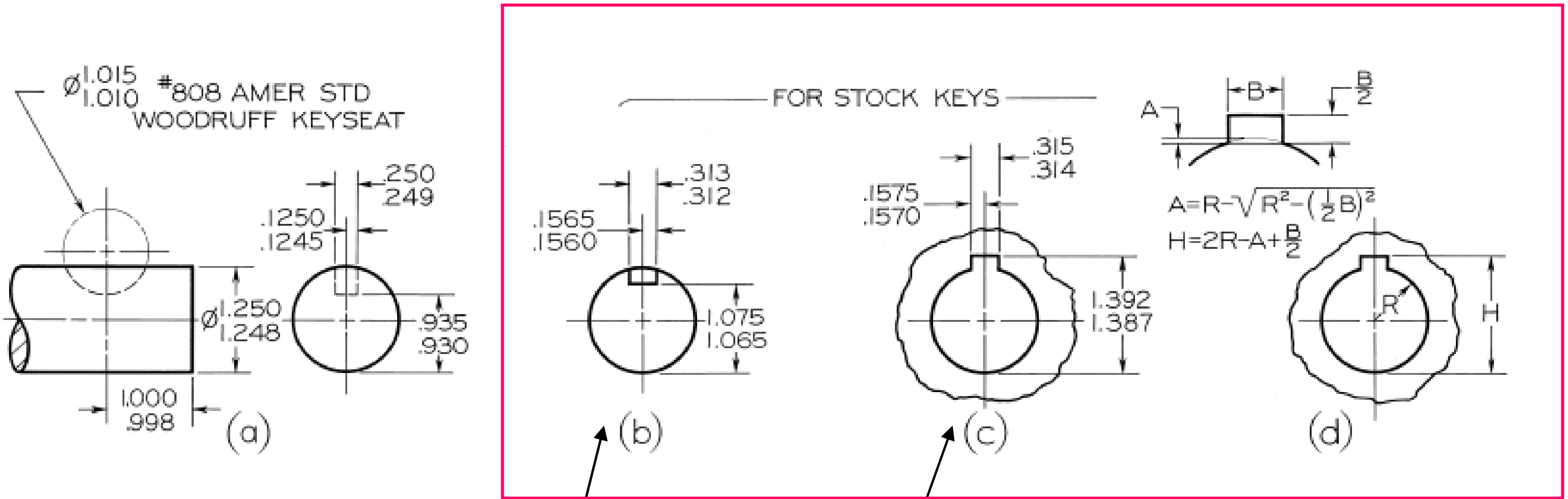
Note: Values in nonshaded areas are preferred. Dimensions are in inches.

Step 1 – Determine key size based on shaft diameter

Step 2 – Calculate required length, L, based on torque (11.4)

Step 3 – Specify appropriate shaft and bore dimensions for keyseat:

See Figure 11.2

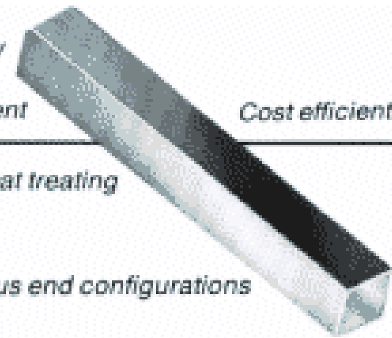


For 5/16" key

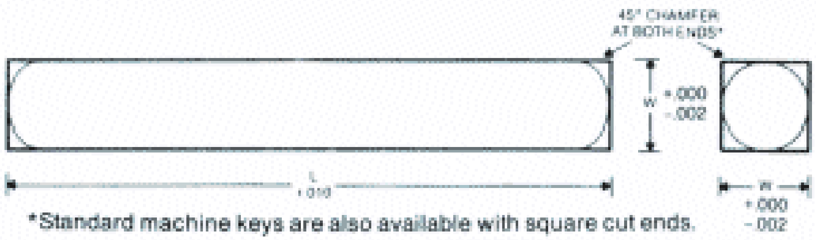
SHAFT

BORE

Note, should also specify fillet radii and key chamfers – see Table 11-2



STANDARD SQUARE MACHINE KEYS



*Standard machine keys are also available with square cut ends.

SQUARE MACHINE KEY STANDARD SIZES

Lengths	Width						
	1/8	3/16	1/4	5/16	3/8	7/16	1/2
1/2	■	■	■	■			
3/4	■	■	■	■	■	■	■
1	■	■	■	■	■	■	■
1 1/4	■	■	■	■	■	■	■
1 1/2	■	■	■	■	■	■	■
1 3/4	■	■	■	■	■	■	■
2	■	■	■	■	■	■	■
2 1/4		■	■	■	■	■	■
2 1/2		■	■	■	■	■	■
2 3/4			■	■	■	■	■
3			■	■	■	■	■
3 1/2					■	■	■
4					■	■	■
4 1/2						■	■

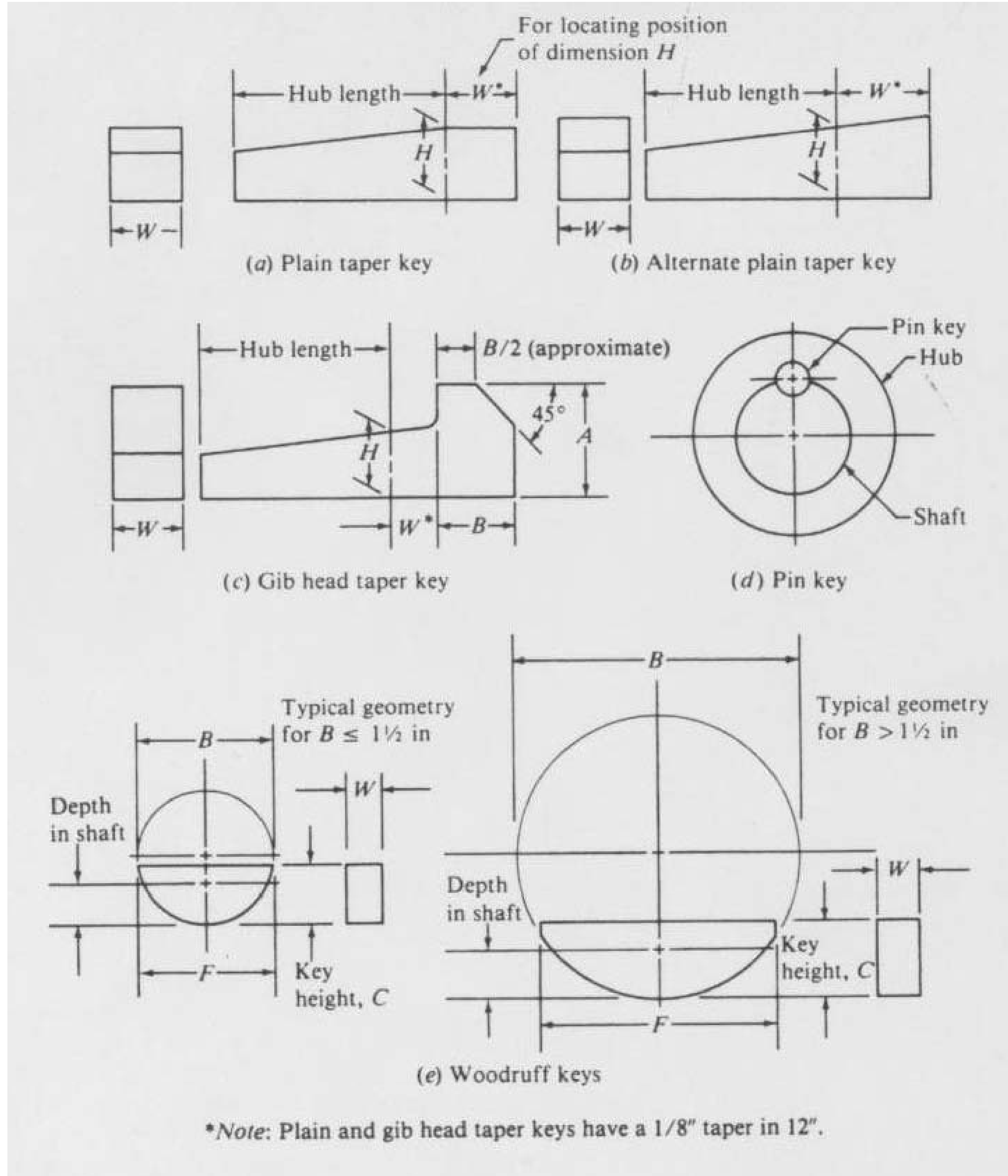
Other types of keys:

a. Tapered key – can install after hub (gear) is installed over shaft.

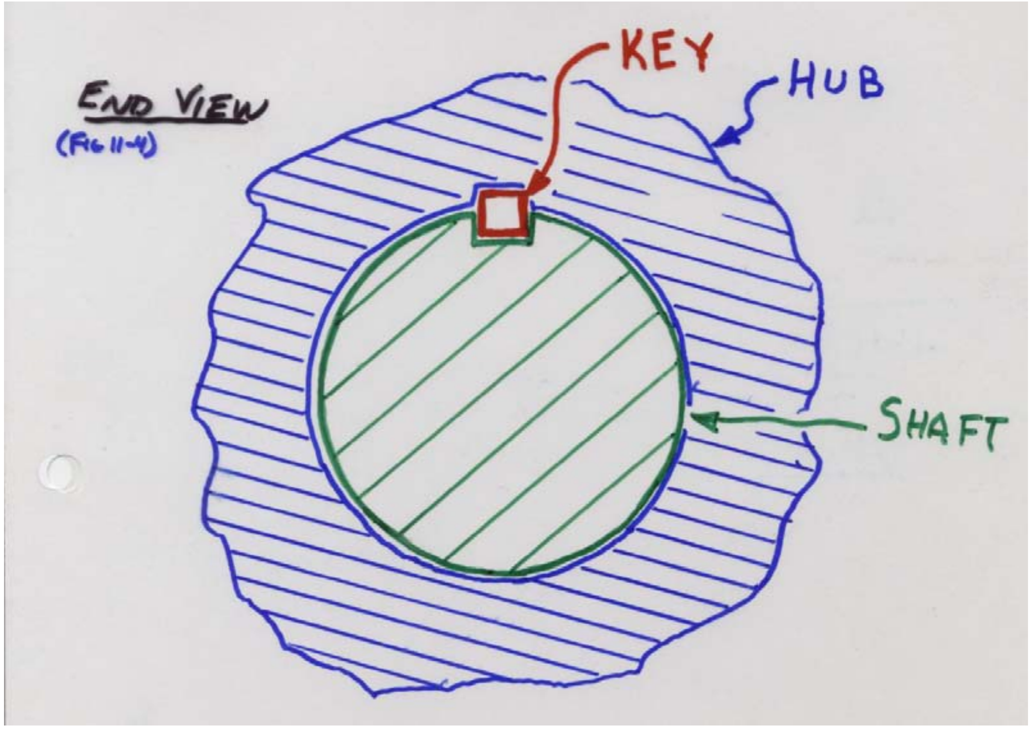
b. Gib head key – ease of extraction

c. Pin keys – low stress concentration

d. Woodruff key – light loading offers ease of assembly



11.4 Design of Keys – stress analysis to determine required length:



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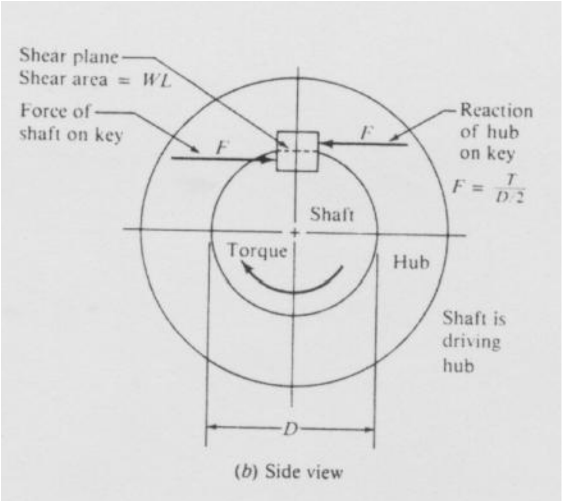


No load

Torque being transmitted

T = F/(D/2) or F = T/(D/2)   this is the force the key must react!!!

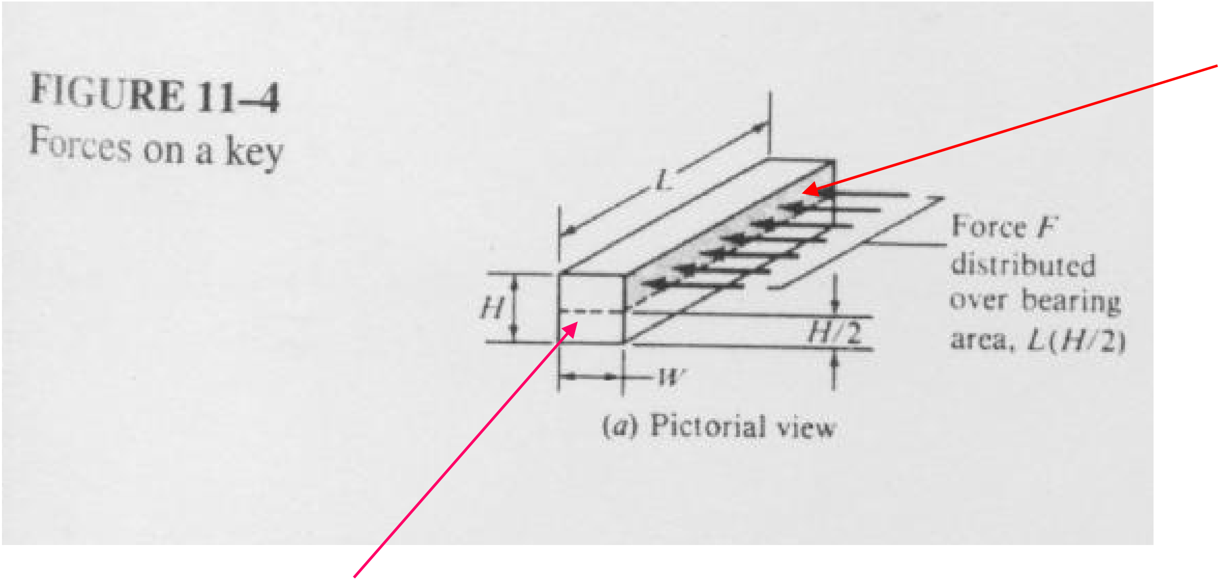


FIGURE 11–4
Forces on a key

Required Length based on Shear Stress:

$$L = \frac{2T}{\tau_d DW} \qquad where\ \tau_d = 0.5Sy / N$$

Required Length based on Bearing Stress:

$$L = \frac{4T}{\sigma_d DH} \qquad where\ \sigma_d = Sy / N$$

Typical parameters for keys:

N = 3, material 1020 CD (Sy = 21,000 psi)

11.4 Splines

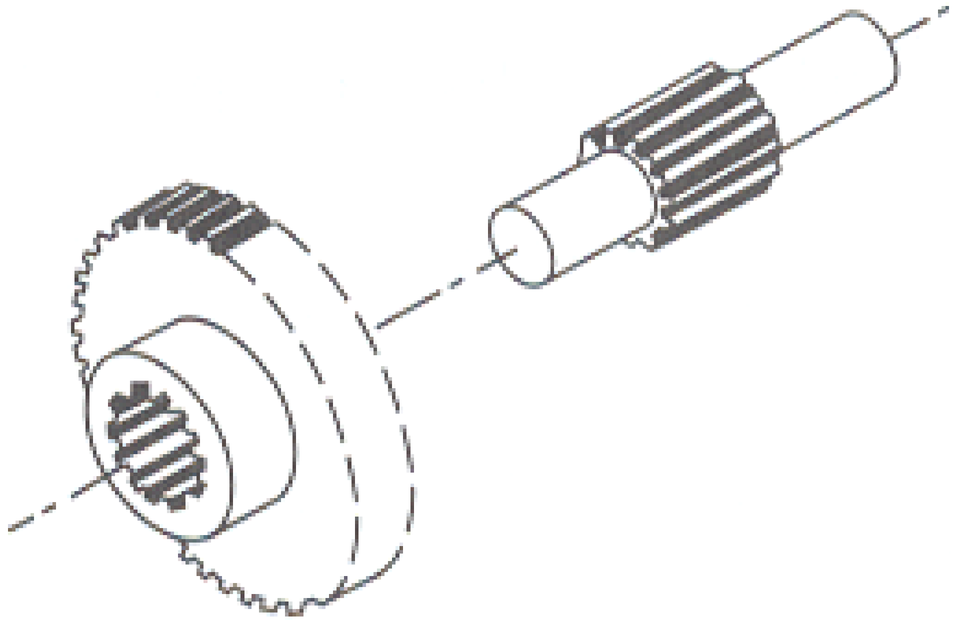
Advantages:

- Can carry higher torque for given diameter (vs keys) or
- Lower stress on attachment (gear)
- Better fit, less vibration (spline integral to shaft so no vibrating key)
- May allow axial motion while reacting torque

Disadvantage:

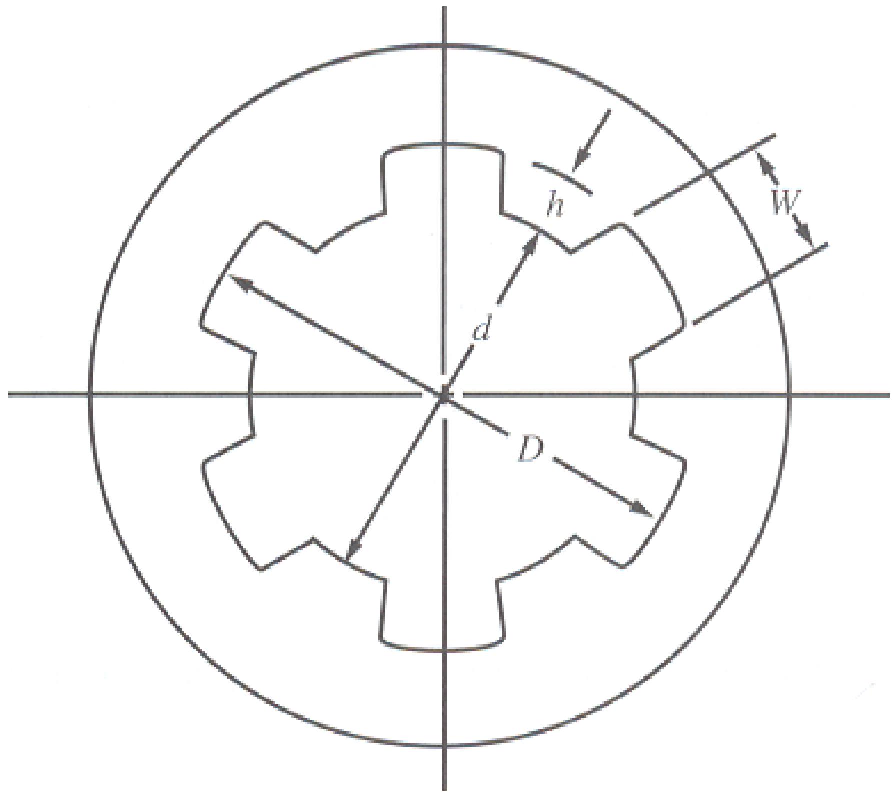
- Cost
- Impractical to use as fuse

Splines



“Axial keys” machined into a shaft

Transmit torque from shaft to another machine element



Advantages

- Uniform transfer of torque
- Lower loading on elements
- No relative motion between “key” and shaft
- Axial motion can be accommodated (can cause fretting and corrosion)
- Mating element can be indexed with a spline
- Generally hardened to resist wear

Spline Types

- Straight
  - SAE
  - 4, 6, 10 or 16 splines
- Involute
  - Pressure angles of 30, 37.5, or 45 deg.
  - Tend to center shafts for better concentricity

SAE Spline Sizes

A: Permanent Fit

B: Slide without Load

C: Slide under Load

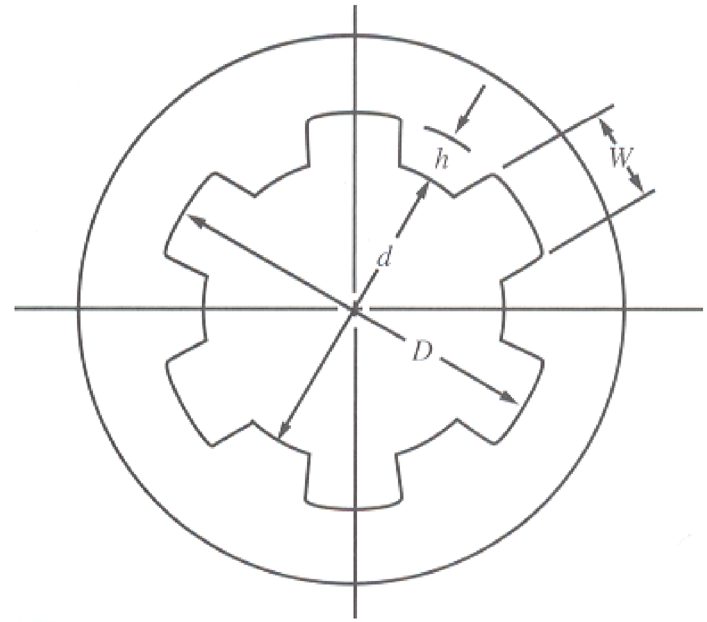


TABLE 11–4 Formulas for SAE straight splines

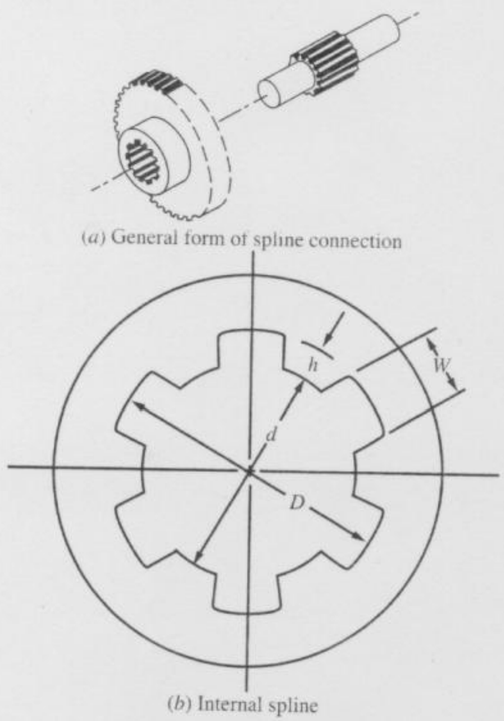
No. of splines	*W*, for all fits	*A:* Permanent fit		*B:* To slide without load		*C:* To slide under load	
		h	*d*	*h*	*d*	*h*	*d*
Four	0.241D	0.075D	0.850D	0.125D	0.750D		
Six	0.250D	0.050D	0.900D	0.075D	0.850D	0.100D	0.800D
Ten	0.156D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D
Sixteen	0.098D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D

Note: These formulas give the maximum dimensions for *W*, *h*, and *d*.

Two types of splines:

Straight Sided

FIGURE 11–6
Straight-sided spline



Involute:

FIGURE 11–8
30° involute spline

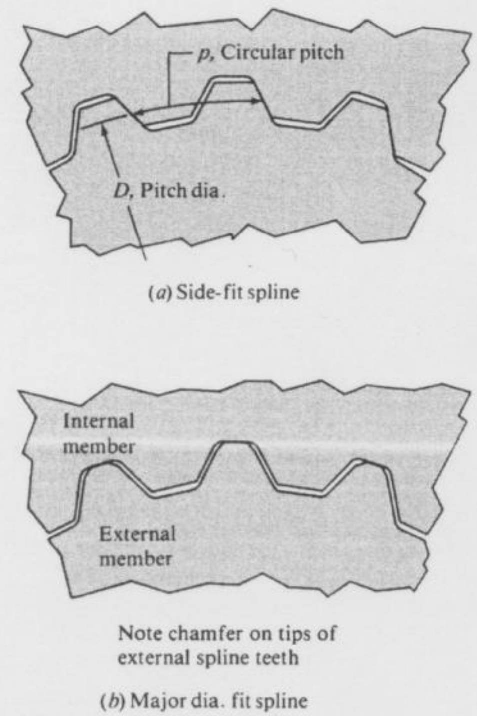


TABLE 11–5 Torque capacity for straight splines per inch of spline length

Number of splines	Fit	Torque capacity	Required diameter
4	*A*	$139D^2$	$\sqrt{T/139}$
4	*B*	$219D^2$	$\sqrt{T/219}$
6	*A*	$143D^2$	$\sqrt{T/143}$
6	*B*	$208D^2$	$\sqrt{T/208}$
6	*C*	$270D^2$	$\sqrt{T/270}$
10	*A*	$215D^2$	$\sqrt{T/215}$
10	*B*	$326D^2$	$\sqrt{T/326}$
10	*C*	$430D^2$	$\sqrt{T/430}$
16	*A*	$344D^2$	$\sqrt{T/344}$
16	*B*	$521D^2$	$\sqrt{T/521}$
16	*C*	$688D^2$	$\sqrt{T/688}$

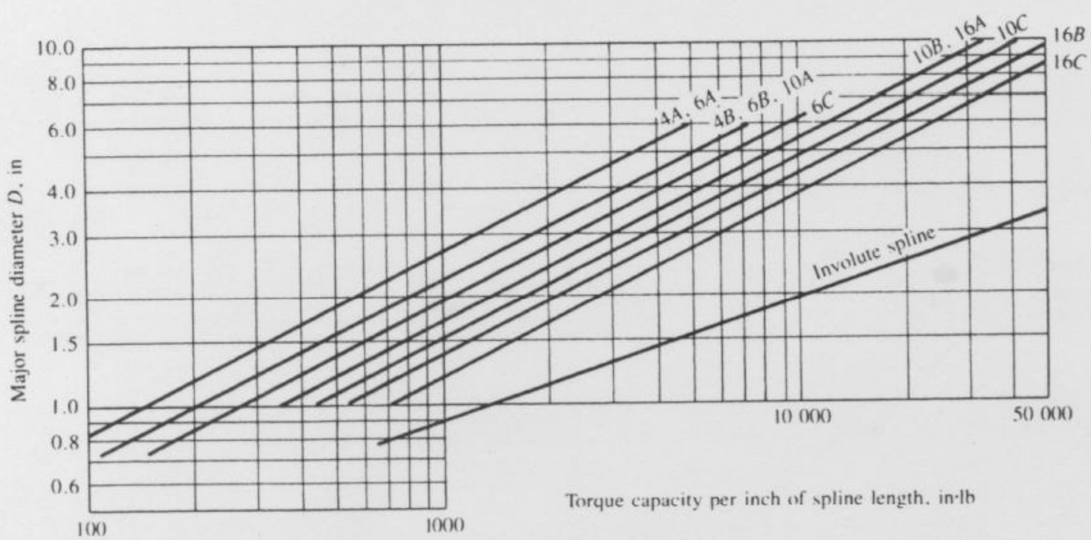


FIGURE 11–7 Torque capacity per inch of spline length, lb · in

Use this for spline design – SAE formulas based on 1,000 psi bearing stress allowable!!

Use this to get diameter. Then table 11.4 to get W, h, d

Torque Capacity

- Torque capacity is based on 1000 psi bearing stress on the sides of the splines

$$T = 1000*N*R*h$$

N = number of splines

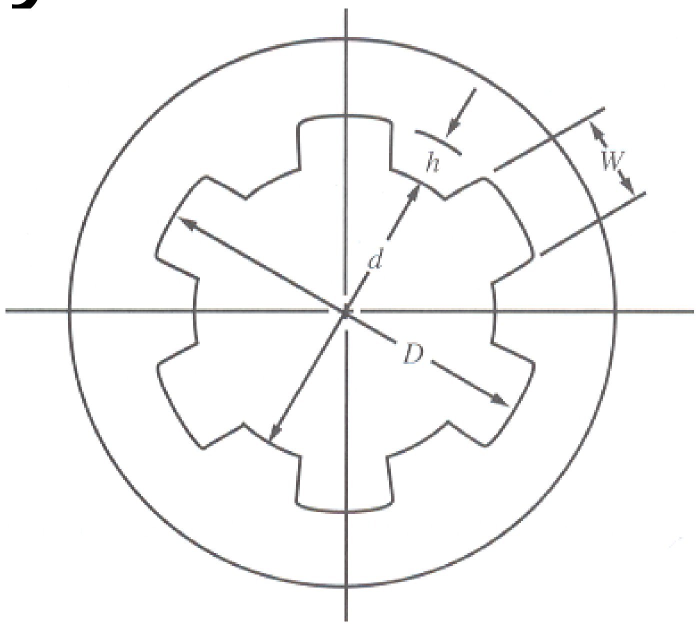
R = mean radius of the splines

h = depth of the splines

Torque Capacity Cont'd

$$R = \frac{1}{2}\left(\frac{D+d}{2}\right) = \frac{D+d}{4}$$

$$h = \frac{1}{2}(D-d)$$



Substituting R and h into torque equation:

$$T = 1000N\left(\frac{D+d}{4}\right)\left(\frac{D-d}{2}\right) = 1000N\left(\frac{D^2-d^2}{8}\right)$$

Torque Capacity Cont'd

- Further refinement can be done by substituting appropriate values for N and d.
- For 16 spline version, with C fit,

$$N = 16 \text{ and } d = .810D$$

$$T = 1000(16)\left(\frac{D^2 - (.810D)^2}{8}\right)$$

$$T = 688D^2$$ Torque in IN-LBS/INCH of spline

$$D = \sqrt{T/688}$$ Required D for given Torque

Torque Capacity for Straight Splines

TABLE 11–5 Torque capacity for straight splines per inch of spline length

Number of splines	Fit	Torque capacity	Required diameter
4	*A*	$139D^2$	$\sqrt{T/139}$
4	*B*	$219D^2$	$\sqrt{T/219}$
6	*A*	$143D^2$	$\sqrt{T/143}$
6	*B*	$208D^2$	$\sqrt{T/208}$
6	*C*	$270D^2$	$\sqrt{T/270}$
10	*A*	$215D^2$	$\sqrt{T/215}$
10	*B*	$326D^2$	$\sqrt{T/326}$
10	*C*	$430D^2$	$\sqrt{T/430}$
16	*A*	$344D^2$	$\sqrt{T/344}$
16	*B*	$521D^2$	$\sqrt{T/521}$
16	*C*	$688D^2$	$\sqrt{T/688}$

Torque Capacity for Straight Splines

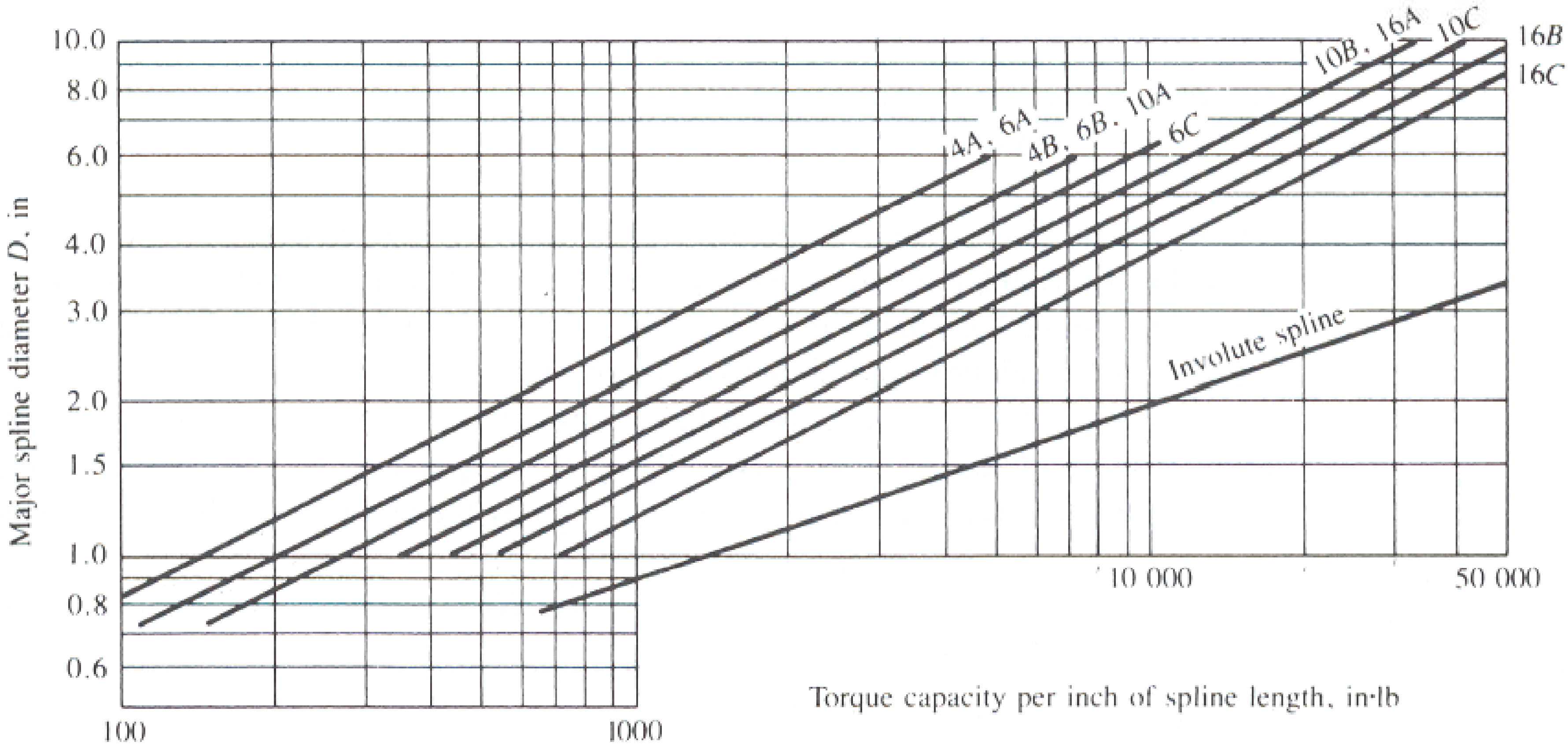


FIGURE 11–7 Torque capacity per inch of spline length, lb · in

Example: A chain sprocket delivers 4076 in-lbs of torque to a shaft having a 2.50 inch diameter. The sprocket has a 3.25 inch hub length. Specify a suitable spline having a B fit.

$$T = kD^2L$$

T	= torque capacity in in-lbs
kD^2	= torque capacity per inch (from Table 11-5)
L	= length of spline in inches

Example Continued

$$k = \frac{T}{D^2 L} = \frac{4076 \text{ in} \cdot \text{lbs}}{(2.50")^2 (3.25")} = 200.7 = 201$$

- From Table 11-5, use 6 splines

TABLE 11–5 Torque capacity for straight splines per inch of spline length

Number of splines	Fit	Torque capacity	Required diameter
4	*A*	$139D^2$	$\sqrt{T/139}$
4	*B*	$219D^2$	$\sqrt{T/219}$
6	*A*	$143D^2$	$\sqrt{T/143}$
6	*B*	$208D^2$	$\sqrt{T/208}$
6	*C*	$270D^2$	$\sqrt{T/270}$
10	*A*	$215D^2$	$\sqrt{T/215}$
10	*B*	$326D^2$	$\sqrt{T/326}$
10	*C*	$430D^2$	$\sqrt{T/430}$
16	*A*	$344D^2$	$\sqrt{T/344}$
16	*B*	$521D^2$	$\sqrt{T/521}$
16	*C*	$688D^2$	$\sqrt{T/688}$

Torque Capacity for Straight Splines

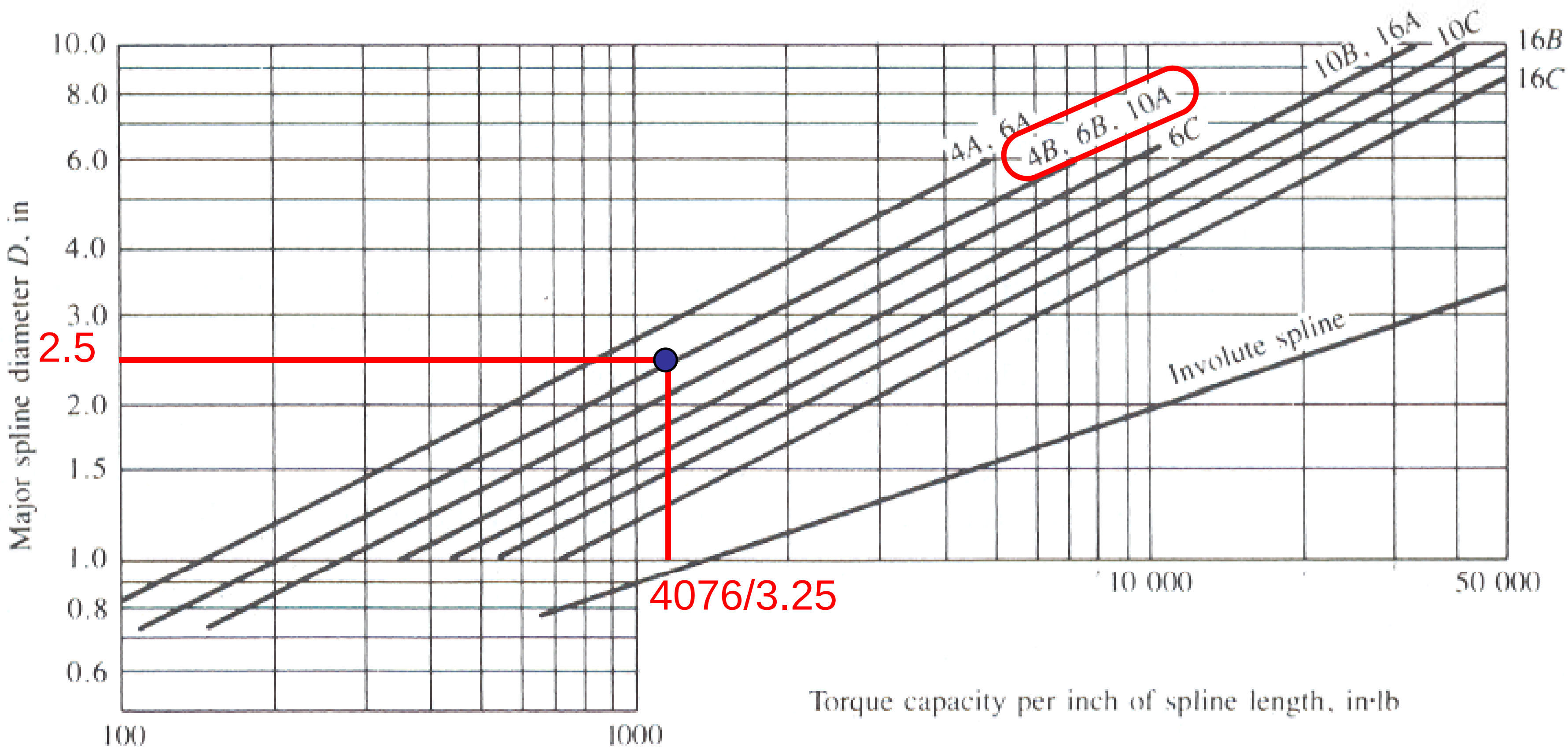
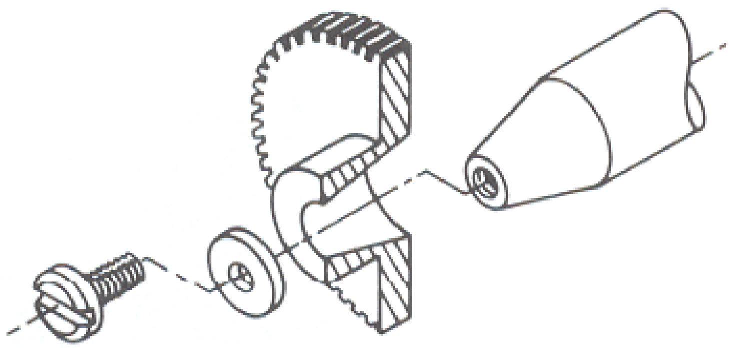


FIGURE 11–7 Torque capacity per inch of spline length, lb · in

Taper & Screw



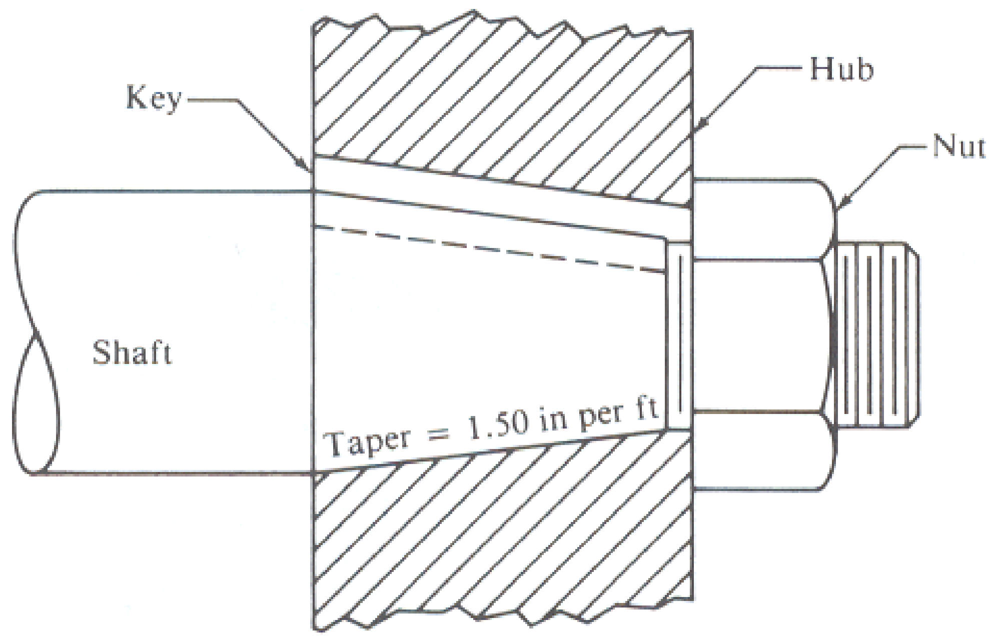
(*a*) Taper and screw

Expensive – machining

Good concentricity

Moderate torque capacity

Can use a key too



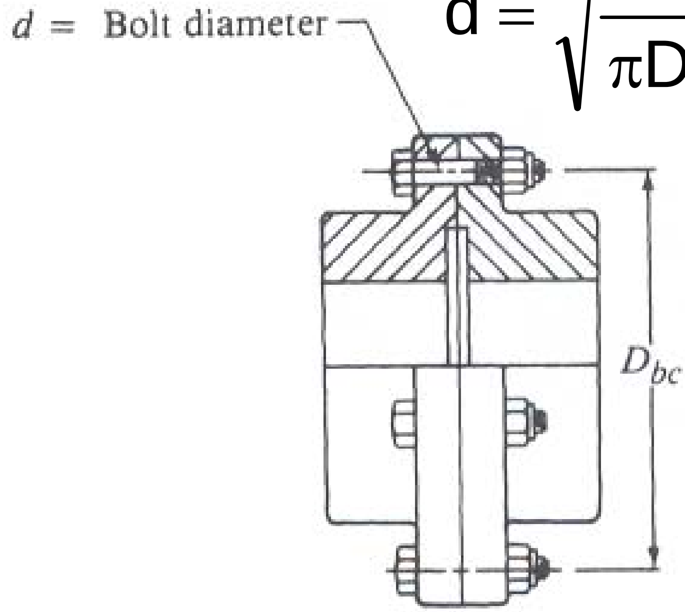
(*b*) Taper and nut

Couplings

- Used to connect two shafts together at their ends to transmit torque from one to the other.
- Two kinds of couplings:
  - RIGID
  - FLEXIBLE

Rigid Couplings

$$d = \sqrt{\frac{8T}{\pi D_{bc} N(\tau_d)}}$$



Cutaway view

NO relative motion between the shafts.

Precise alignment of the shafts

Bolts in carry torque in shear. N = # of bolts.

Flexible Couplings

- Transmit torque smoothly
- Permit some axial, radial and angular misalignment

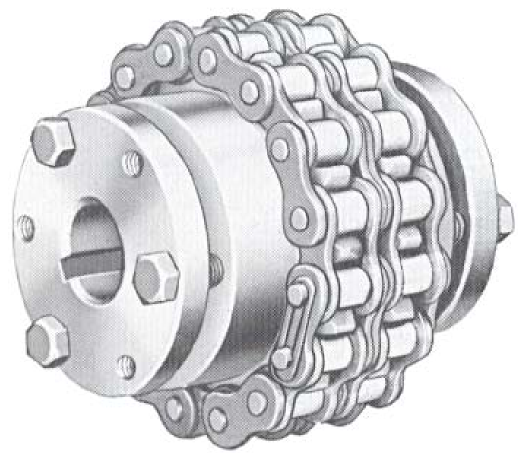


FIGURE 11–16 Chain coupling. Torque is transmitted through a double roller chain. Clearances between the chain and the sprocket teeth on the two coupling halves accommodate misalignment. (Emerson Power Transmission Corporation, Ithaca, NY)

FIGURE 11–17 Ever-Flex coupling. The features of this coupling are that it (1) generally minimizes torsional vibration; (2) cushions shock loads; (3) compensates for parallel misalignment up to 1/32 in; (4) accommodates angular misalignment of ±3°; and (5) provides adequate end float, ±1/32 in. (Emerson Power Transmission Corporation, Ithaca, NY)

Flexible Couplings

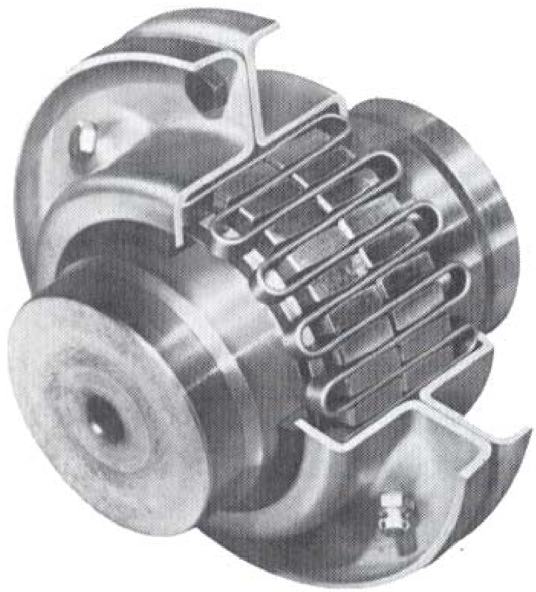


FIGURE 11–18 Grid-Flex coupling. Torque is transmitted through a flexible spring steel grid. Flexing of the grid permits misalignment and makes it torsionally resilient to resist shock loads. (Emerson Power Transmission Corporation, Ithaca, NY)

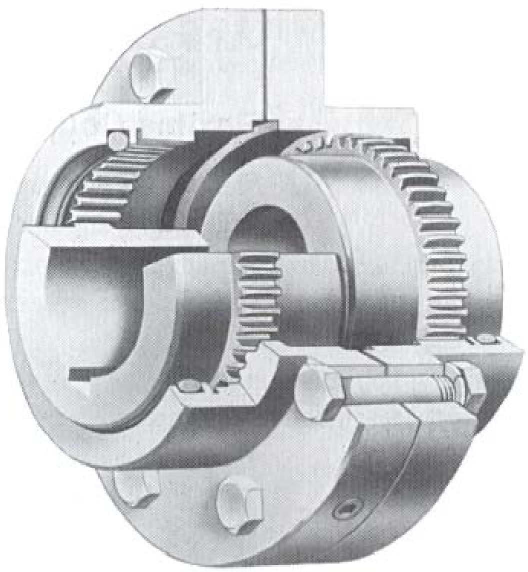
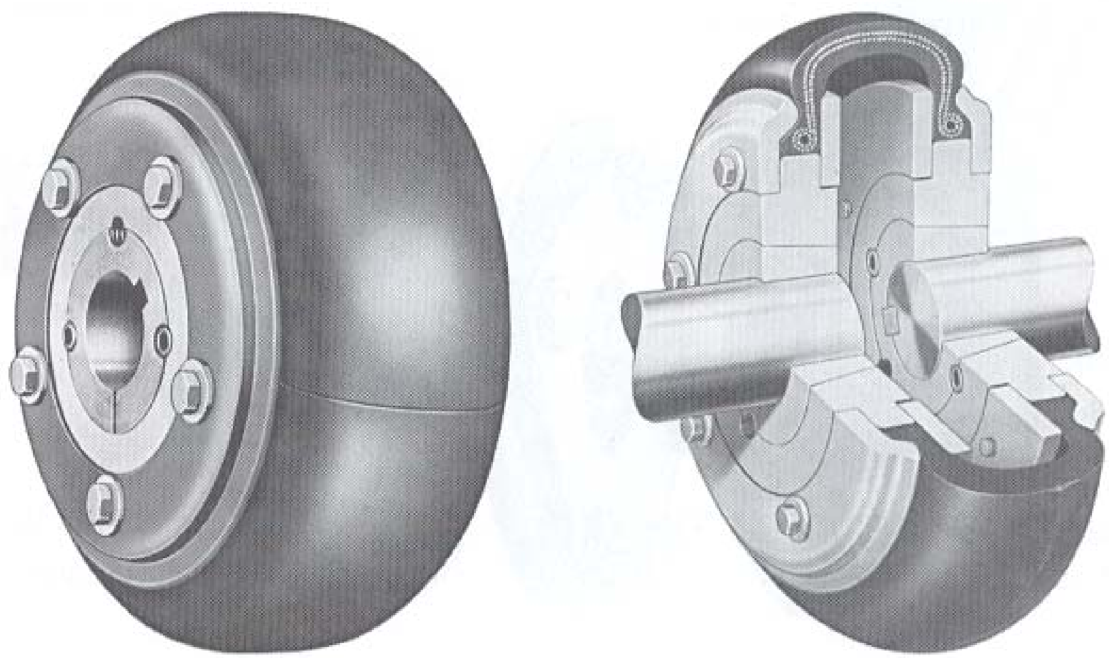


FIGURE 11–19 Gear coupling. Torque is transmitted between crown-hobbed teeth from the coupling half to the sleeve. The crown shape on the gear teeth permits misalignment. (Emerson Power Transmission Corporation, Ithaca, NY)

Flexible Couplings

FIGURE 11–21 PARA-FLEX® coupling. Using an elastomeric element permits misalignment and cushions shocks. (Rockwell Automation/Dodge, Greenville, SC)

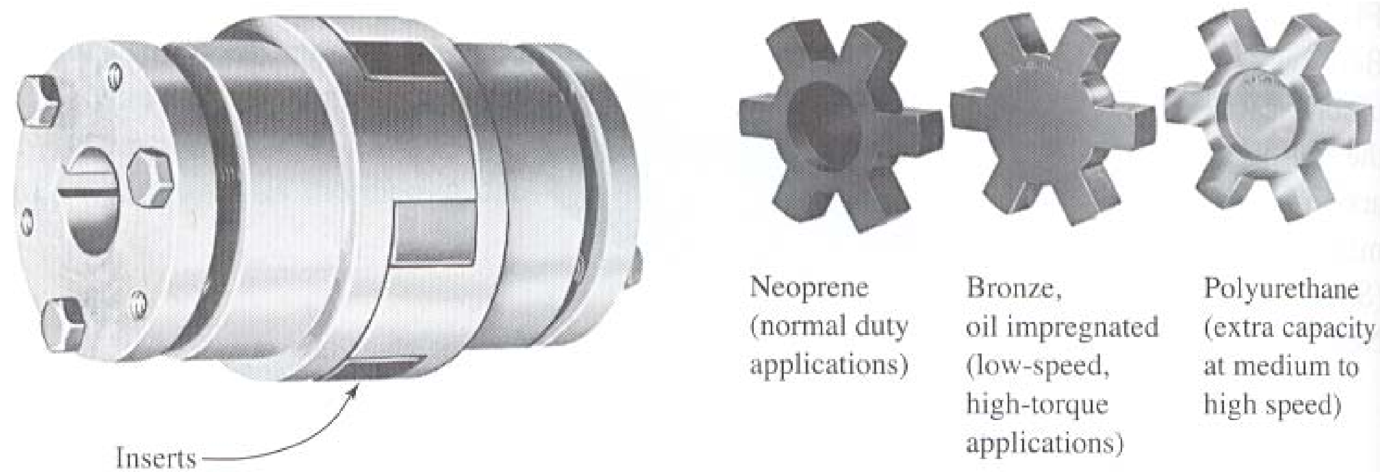


Lord Corp. Products

FIGURE 11–22
Dynaflex® coupling. Torque is transmitted through elastomeric material that flexes to permit misalignment and to attenuate shock loads. (Lord Corporation, Erie, PA)

Flexible Coupling

FIGURE 11–23
Jaw-Type coupling (Emerson Power Transmission Corporation, Ithaca, NY)



(a) Assembled coupling

(b) Types of inserts

Universal Joints

Large shaft misalignments permissible

Key factors in selection are Torque, Angular Speed and the Operating Angle

FIGURE 11–25 Universal joint components (Curtis Universal Joint Co., Inc., Springfield, MA)

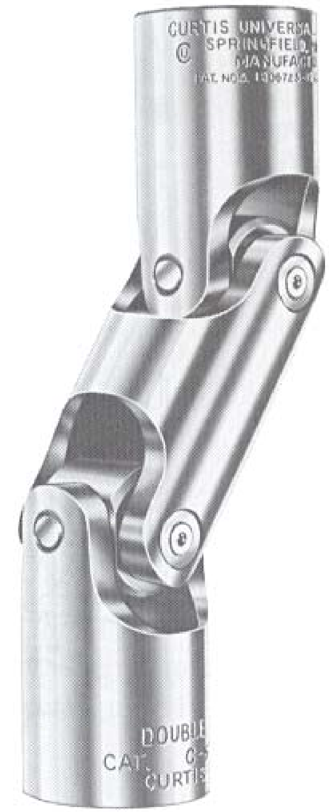
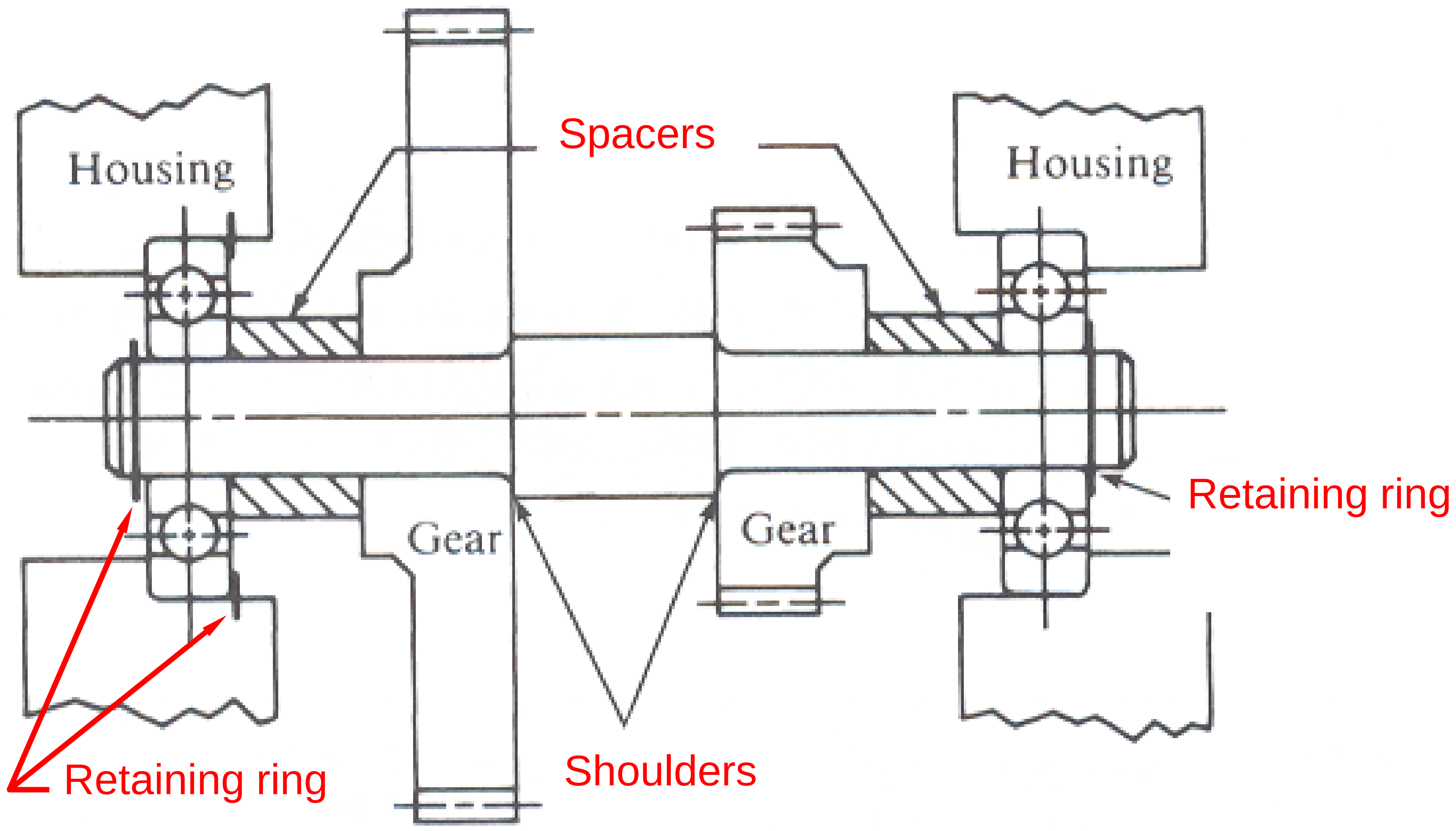


FIGURE 11–26 Double universal joint (Curtis Universal Joints Co., Inc., Springfield, MA)

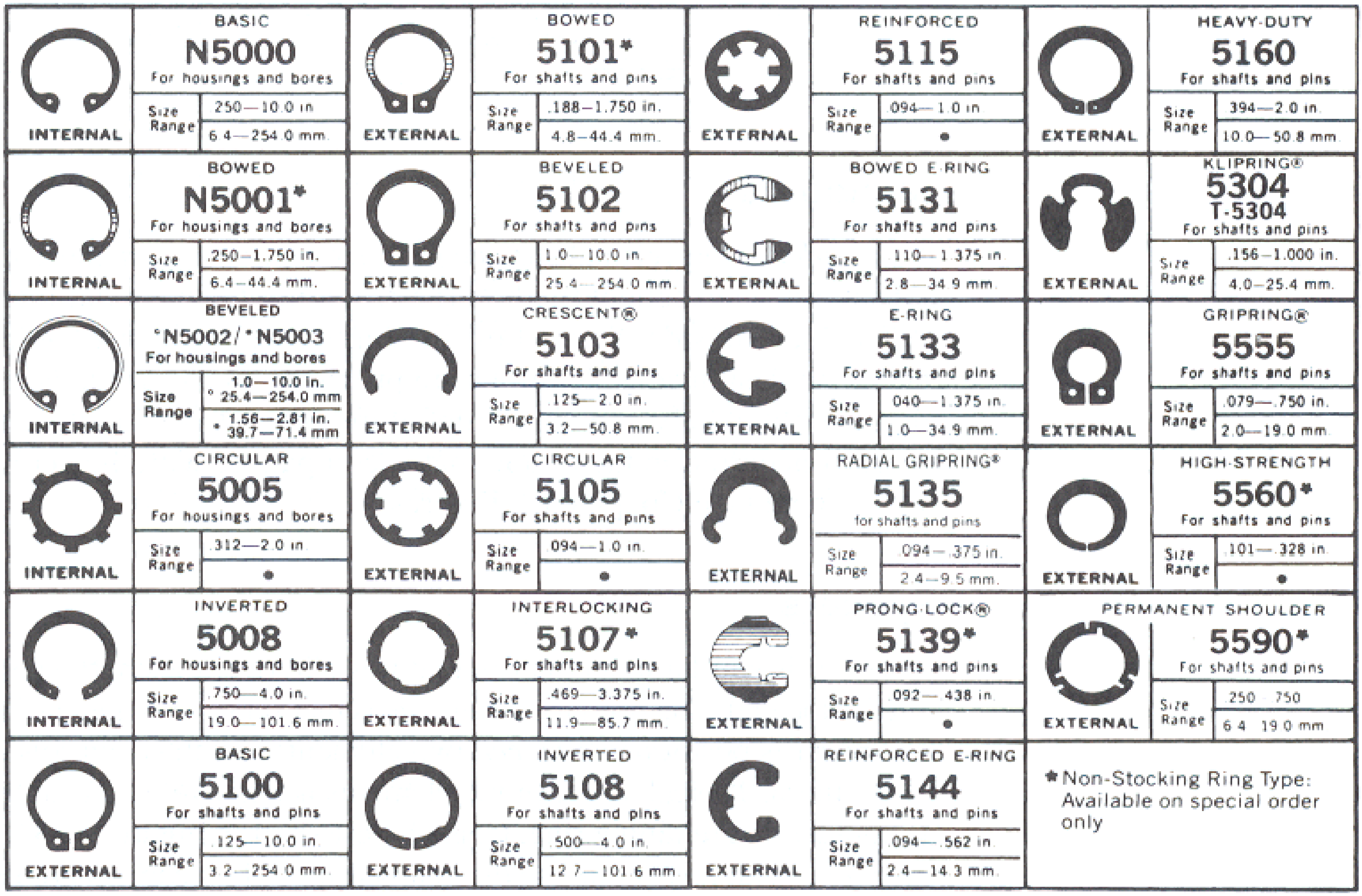
Output not uniform wrt input

Output IS uniform wrt input

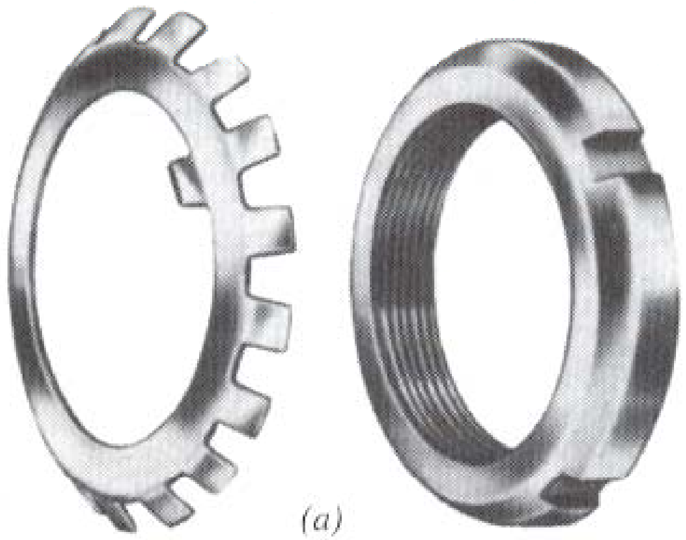
Axial Constraint Methods



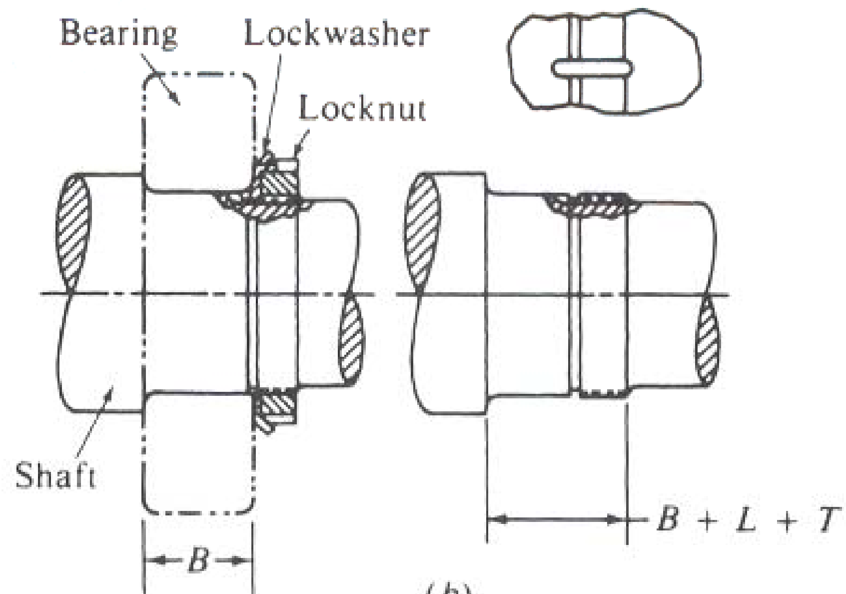
Retaining Rings



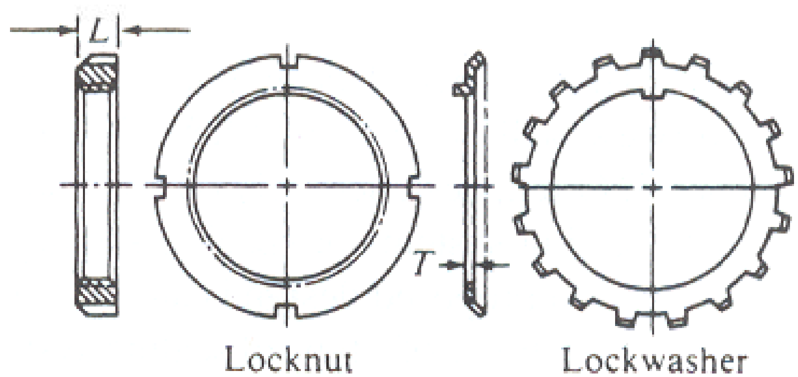
Locknuts



(a)



(b)



(c)