Chapter 11

Keys, Couplings and Seals

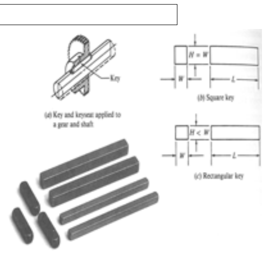
Material taken for Mott, 2003, Machine Elements in Mechanical Design

Keys

- A key is a machinery component that provides a torque transmitting link between two power-transmitting elements.
- The most common types of keys are the square and rectangular parallel keys. With the square being the most common. The rectangular key is most common in either very small or very large shafts.
- The key width is nominally ¼ of the shaft diameter.

Parallel Keys

- The key seats in the shaft and the hub so that exactly ½ of the key bears on the shaft and the other ½ bears on the hub.
- The parallel key is installed on the shaft, before mating the hub. The hub is slide over the key to provide the interface.



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Key Shaft vs. Shaft Diameter

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

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

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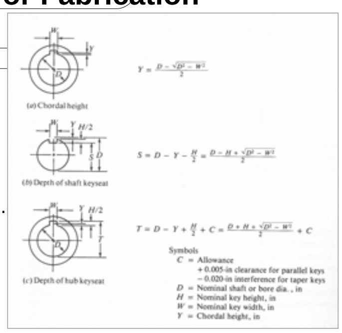
Dimensions for Fabrication

- Keys are usually cut with a end-mill or circular-mill cutters and leave square corners which create stress concentrations.
- Radiused keyseats and chamfered keys can reduce stress concentrations.



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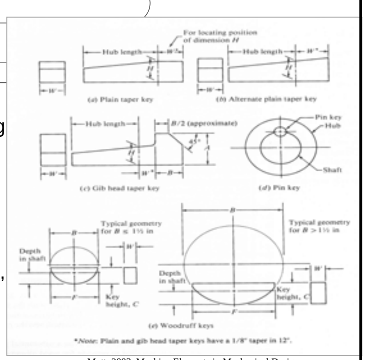
Key Types

- Tapered keys are installed after mating the hub and shaft. The taper extends over the length of the hub.
- Pin keys reduce stress concentration, but requires a tight fit.



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Woodruff Keys

TABLE 11–3 Woodruff key dimensions

Key number	Nominal key size, $W \times B$	Actual length, F	Height of key, C	Shaft keyseat depth	Hub keyseat depth
202	$1/16 \times 1/4$	0.248	0.104	0.0728	0.0372
204	$1/16 \times 1/2$	0.491	0.200	0.1668	0.0372
406	$1/8 \times 3/4$	0.740	0.310	0.2455	0.0685
608	$3/16 \times 1$	0.992	0.435	0.3393	0.0997
810	$1/4 \times 1\frac{1}{4}$	1.240	0.544	0.4170	0.1310
1210	$3/8 \times 1\frac{1}{4}$	1.240	0.544	0.3545	0.1935
1628	$1/2 \times 3\frac{1}{2}$	2.880	0.935	0.6830	0.2560
2428	$3/4 \times 3\frac{1}{2}$	2.880	0.935	0.5580	0.3810

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- Used in light loading applications, where it is desirable to easy assembly and dis-assembly.

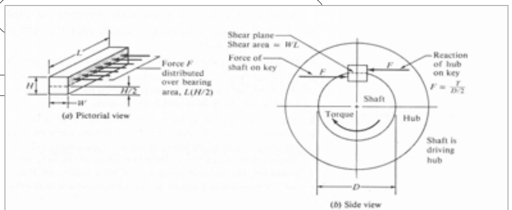
Selection of Key

- The type and size of the key is usually selected after the shaft and hub have been designed.
- The length and material specification is determined through by design analysis, while two factors limit their selection:
  - Width of the hub
  - Distance to adjacent stress concentrations.
- Key is held in place by; set screws, shoulders, retaining rings, or spacers.
- Keys are commonly made of low-carbon, cold-drawn steel; such as AISI 1020CD with 61 ksi, ultimate strength.

Stress Analysis

- There are two basic failure modes for keys transmitting power.
  - Shear across the shaft/hub interface
  - Compression failure due to bearing stress between key and shaft, or hub.

Shear Stress Analysis



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- The torque in the shaft creates a force on the lower left side of the key which in turn creates a reaction force on the upper right side of the key, transmitting the torque into the hub.
- This force couple places the key in shear over the surface A = W*L, where F = T / (D/2)

Shear Stress Analysis

- Shearing stress

$$\tau = \frac{F}{A_s} = \frac{T}{(D/2)(WL)} = \frac{2T}{DWL}$$

- Design stress

$$\tau_d = 0.5 s_y / N$$

- Key Length

$$L = \frac{2T}{\tau_d DW}$$

Bearing Stress Analysis

- Failure in bearing is due to compressive stress that acts of shaft or hub area: L * (H/2), where failure occurs on the surface with the lowest compressive yield strength.
- Design Stress

$$\sigma_d = s_y / N$$

- N = 3 is typical

Bearing Stress Analysis, cont.

- Actual compressive stress

$$\sigma = \frac{F}{A_c} = \frac{T}{(D/2)(L)(H/2)} = \frac{4T}{DLH}$$

- Solving of required length using design stress

$$L = \frac{4TN}{DW\, S_y} = \frac{4T}{\sigma_d\, DH}$$

- Using S_y from material of lesser yield compressive strength, be it the key, shaft or hub.

Parallel-Square Key Design

- Complete shaft design and spec. shaft diameter.
- Select key size, using W=H if shaft size < 6.5".
- Spec. material, usually AISI 1020 CD
- Determine yield strength of key, shaft and hub.
- Check design based on shear stresses in key, and bearing stresses on material with lesser compressive yield strength.
- Complete design details of keyseat and key for fabrication.

Example Problem 11–1 A portion of a shaft where a gear is to be mounted has a diameter of 2.00 in. The gear transmits 2965 lb · in of torque. The shaft is to be made of AISI 1040 cold-drawn steel. The gear is made from AISI 8650 OQT 1000 steel. The width of the hub of the gear mounted at this location is 1.75 in. Design the key.

Solution From Table 11–1, the standard key dimension for a 2.00-in-diameter shaft would be 1/2 in square. See Figure 11–5 for the proposed design.

Material selection is a design decision. Let's choose AISI 1020 CD steel with s_y = 51 000 psi.

A check of the yield strengths of the three materials in the key, the shaft, and the hub indicates that the key is the weakest material. Then Equation (11–5) can be used to compute the minimum required length of the key:

$$L = \frac{4TN}{DWs_y} = \frac{4(2965)(3)}{(2.00)(0.50)(51\,000)} = 0.698 \text{ in}$$

This length is well below the width of the hub of the gear. Notice that the design of the shaft includes retaining rings on both sides of the gear. It is desirable to keep the keyseat well clear of the ring grooves. Therefore, let's specify the length of the key to be 1.50 in.

Summary In summary, the key has the following characteristics:

Material: AISI 1020 CD steel

Width: 0.500 in

Height: 0.500 in

Length: 1.50 in.

Figure 11–5 shows some details of the completed design. A profile keyseat in the shaft is shown.

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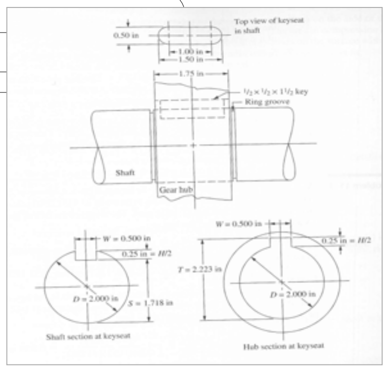
Example



Splines

- A spline can be described as a series of axial keys machined into a shaft, with corresponding grooves machined into the bore of the mating part (gear, sheave, sprocket, etc).
- The splines perform the same function as a key in transmitting torque from the shaft to the mating element.

Splines con't

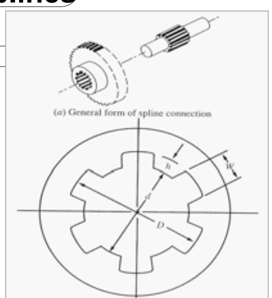
- There are many advantages of splines over keys.
  - Because 4 or more splines are used, as compared with 1 or 2 keys, a more uniform transfer of torque and a lower loading on a given part of the shaft/hub interface result.
- The splines are integral with the shaft, so no relative motion can occur as between a key and the shaft.
- Because of the multiple splines on the shaft, the mating element can be indexed to various positions.

Splines con’t

- Splines can either be straight-sided or involute.
- The involute form is preferred because it provides for self-centering of the mating element and because it can be machined with standard hobs used to cut gears.

Straight-Sided Splines

- Straight splines are made according to the specifications of the SAE and usually contain 4, 6, 10, or 16 splines.



(a) General form of spline connection

(b) Internal spline

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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*.

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Straight-Sided Splines con't

- The torque capacity for SAE splines is based on the limit of 1000-psi bearing stress on the sides of the splines, from which this formula is derived:
  - T = 1000NRh
  - Where N = number of splines
  - R = mean radius of the splines
  - h = depth of the splines

Straight-Sided Splines con't

- The torque capacity is per inch of length of the spline.

$$\mathbf{R} = \frac{1}{2}[\frac{\mathbf{D}}{2} + \frac{\mathbf{d}}{2}] = \frac{\mathbf{D} + \mathbf{d}}{4}$$

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

Straight-Sided Splines con't

- Then,

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

- This equation can be further refined for each of the types of splines by substitution of the appropriate relationships for N and d.

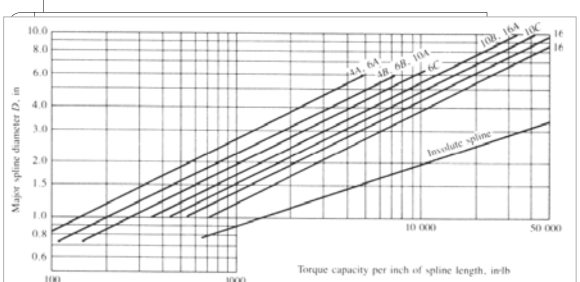
Straight-Sided Splines con't

- For example, for the 6-spline version and the B fit, N = 6, d = 0.850D, and $d^2 = 0.7225D^2$.
- Then,
  $$\mathbf{T} = 1000 * 6\frac{\mathbf{D}^2 - 0.7225\mathbf{D}^2}{8} = 208\mathbf{D}^2$$
- Therefore, the required diameter to transmit a given torque would be $\mathbf{D} = \sqrt{\mathbf{T}/208}$

Straight-Sided Splines con't

- Use the same approach to find the torque capacities and required diameters for the other versions of straight splines.
- The next slide enables an acceptable diameter to be chosen for a spline to carry a given torque, depending on the desired fit, A, B, or C.

Torque Capacity Per Inch of Spline Length



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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	$688D^2$	$\sqrt{T/688}$

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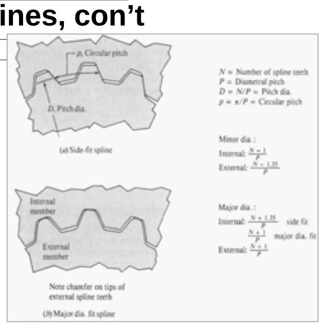
Involute Splines

- Involute splines typically made with pressure angles of 30°, 37.5°, or 45°.
- The major diameter fit produces accurate concentricity between the shaft and the mating element.

Involute Splines, con’t

- In the side fit, contact occurs only on the sides of the teeth, but the involute form tends to center the shaft in the mating splined hub.



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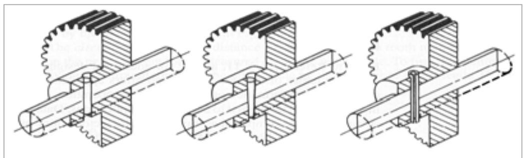
Pinning

- With the element in position on the shaft, a hole can be drilled through both the hub and the shaft, and a pin can be inserted in the hole.
- The straight, solid, cylindrical pin is subjected to shear over two cross sections.
- If there is a force, F, on each end of the pin at the shaft/hub interface, and if the shaft diameter is D, then:
  - T = 2F (D /2) = FD

Pinning con't



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Pinning con't

- With the symbol d representing the pin diameter, the shear stress in the pin is:

$$\tau = \frac{\mathbf{F}}{\mathbf{A_s}} = \frac{\mathbf{T}}{\mathbf{D}(\pi \mathbf{d}^2/4)} = \frac{4\mathbf{T}}{\mathbf{D}(\pi \mathbf{d}^2)}$$

- Letting the shear stress equal the design stress in shear as before,

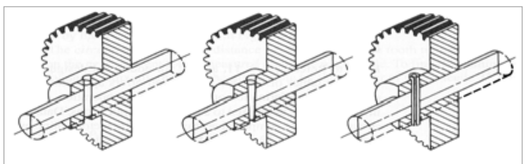
$$\mathbf{d} = \sqrt{\frac{4\mathbf{T}}{\mathbf{D}(\pi)(\tau_\mathbf{d})}}$$

Pinning con't

- Sometimes the diameter of the pin is purposely made small to ensure that the pin will break if a moderate overload is encountered, in order to protect critical parts of a mechanism.
  - Such a pin is called a shear pin.
- One problem with cylindrical pins is that fitting them adequately to provide precise location of the hub and to prevent the pin from falling out is difficult.

Pinning con't

- The taper pin overcomes some of these problems, as does the split spring pin.



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Keyless Hub to Shaft Connections

- Using a steel ring compressed tightly around a smooth shaft allows torque to be transmitted between the hub of a power-transmitting element and a shaft without having a key between the tow elements.
- The Locking Assembly^{TM} employees steel rings with opposing mating tapers held together with a series of fasteners.

Locking Assemblies

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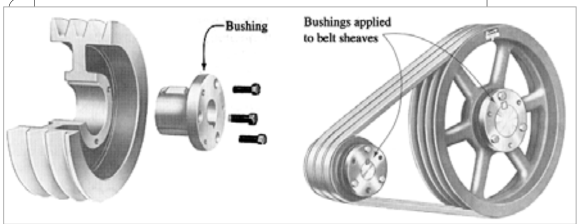
Keyless Hub to Shaft Connections

- With the Locking Assembly™ placed completely within a counter bore of the hub of virtually any kind of power-transmitting element such as a gear, sprocket, fan wheel, cam, coupling, or turbine rotor, the fasteners can be tightened to achieve an excellent torque carrying interface.

Split Taper Bushing

- A split taper bushing uses a key to transmit torque.
- Axial location on the shaft is provided by the clamping action of a split bushing having a small taper on its outer surface.
- When the bushing is pulled into a mating hub with a set of cap screws, the bushing is brought into tight contact with the shaft to hold the assembly in the proper axial position.

Split Taper Bushing



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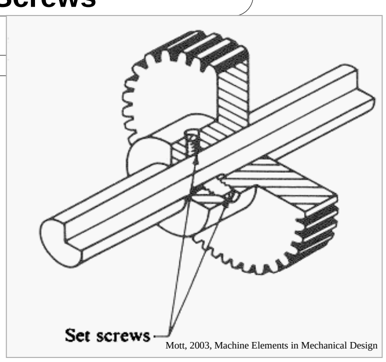
Set Screws

- A set screw is a threaded fastener driven radially through a hub to bear on the outer surface of a shaft.
- The point of the set screw is flat, oval, cone-shaped, cupped, or any of several proprietary forms.
- The point bears on the shaft or digs slightly into its surface.

Set Screws



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Set Screws con’t

- The set screw transmits torque by the friction between the point and the shaft or by the resistance of the material in shear.
- The capacity for torque transmission is somewhat variable, depending on the hardness of the shaft material and the clamping force created when the screw is installed.

Screw Diameter	*Holding Force*
1/4 in	100 lb
3/8 in	250 lb
1/2 in	500 lb
3/4 in	1300 lb
1 in	2500 lb

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Couplings

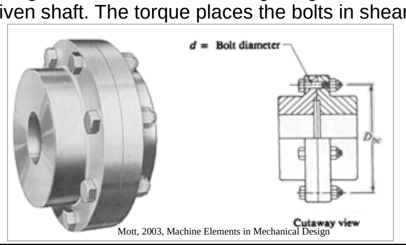
- The term coupling refers to a device used to connect two shafts together at their ends for the purpose of transmitting power.
- There are two general types of couplings: rigid and flexible.

Rigid Couplings

- Rigid couplings are designed to draw two shafts together tightly so that no relative motion can occur between them.
- This design is desirable for certain kinds of equipment in which precise alignment of two shafts is required and can be provided.
- In such cases, the coupling must be designed to be capable of transmitting the torque in the shafts.

Rigid Couplings con't

- The load path is from the driving shaft to its flange, through the bolts, into the mating flange, and out to the driven shaft. The torque places the bolts in shear.



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Rigid Couplings con't

- The total shear force on the bolts depends on the radius of the bolt circle, D_{bc} / 2, and the torque, T.
  - F = T / (D_{bc} / 2) = 2T / D_{bc}
- Letting N be the number of bolts, the shear stress in each bolt is:

$$\tau = \frac{F}{A_s} = \frac{F}{N(\pi d^2/4)} = \frac{2T}{D_{bc}N(\pi d^2/4)}$$

Rigid Couplings con't

- Letting the stress be equal the design stress in shear and solving for the bolt diameter:

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

Rigid Couplings con't

- Rigid couplings should be used only when the alignment of the two shafts can be maintained very accurately, not only at the time of installation but also during operation of the machines.
- If significant angular, radial, or axial misalignment occurs, stresses that are difficult to predict and that may lead to early failure due to fatigue will be induced in the shafts.

Flexible Couplings

- Flexible couplings are designed to transmit torque smoothly while permitting some axial, radial, and angular misalignment.
- The flexibility is such that when misalignment does occur, parts of the coupling move with little or no resistance.
- Thus, no significant axial or bending stresses are developed in the shaft.

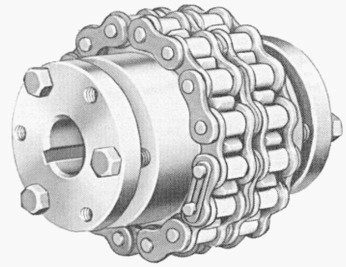

FIGURE 11–16 Chain coupling. Torque is transmitted through a double roller chain. Clearances between the chain and the sprocket teeth on the two coupling halves accommodate misalignment. (Emerson Power Transmission Corporation, Ithaca, NY)

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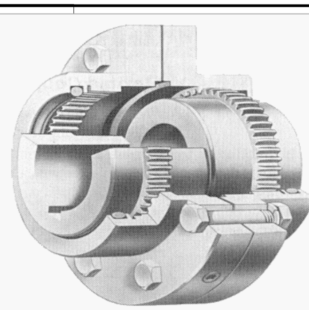

FIGURE 11–19 Gear coupling. Torque is transmitted between crown-hobbed teeth from the coupling half to the sleeve. The crown shape on the gear teeth permits misalignment. (Emerson Power Transmission Corporation, Ithaca, NY)

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FIGURE 11–21 PARA-FLEX® coupling. Using an elastomeric element permits misalignment and cushions shocks.

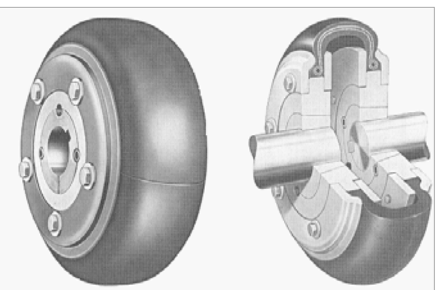

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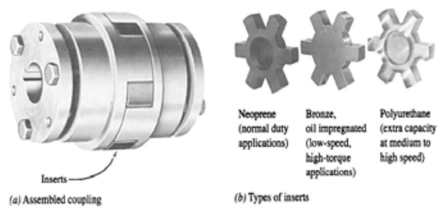
FIGURE 11–23
Jaw-Type coupling (Emerson Power Transmission Corporation, Ithaca, NY)



(a) Assembled coupling (b) Types of inserts

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Universal Joints

- When an application calls for accommodating misalignment between mating shafts that is greater than the 3 degrees typically provided by flexible couplings, a universal joint is used.
- Angular misalignments of up to 45° are possible at low rotational speeds with single universal joints like the one shown on the next slide, consisting of two yokes, a center bearing block, and two pins that pass through the block at right angles.

Universal Joints con't

- Approximately 20 to 30 degrees is more reasonable for speeds about 10 rpm.
- Since universal joints have the disadvantage that the rotational speed of the output shaft is nonuniform in relation to the input shaft.

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)

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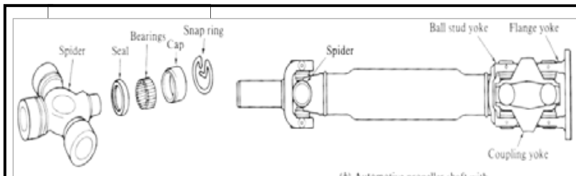
Universal Joints con’t

- A double universal joint allows the connected shafts to be parallel and offset by large amounts.
- Furthermore, the second joint cancels the nonuniform oscillation of the first joint so the input and the output rotate at the same speed.



(a) Automotive universal joint components

(b) Automotive propeller shaft with double Cardan universal joint

- This figure shows a vehicular universal joint used for connecting an engine or transmission to the drive wheels used in some rear-wheel drive cars, light and heavy trucks, agricultural equipment, and construction vehicles.
- The spider assembly contains needle-bearing rollers on each arm.
- The right end shows a ball stud yoke, a flange yoke, and a center coupling yoke that make up a double Cardan universal joint.
- Another style, called a constant velocity joint, or CV joint, is often used as a key component of front wheel drive and all wheel drive vehicle drivelines.

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