CHAPTER 29
COUPLINGS

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GLOSSARY OF SYMBOLS

A	Area or parallel misalignment
b	Bearing spacing
d	Diameter
D	Diameter or distance between equipment
e	Eccentricity
E	Young's modulus or shape factor for maximum allowable stress, psi
F	Force
g	Acceleration due to gravity
h	Height of keyway
I	Second moment of area
J	Polar second moment of area
K_a	U-joint angle correction factor
K_L	U-joint life correction factor
K_s	U-joint speed correction factor
L	Life or length of engagement
ℓ	Length
m	Mass

n	Speed, r/min
N	Number of active elements or bellows convolutions
P	Pressure
PV	Pressure times velocity
r	Radius
R	Operating radius
R_c	Centroidal radius or distance
s	Maximum permissible stroke per convolution for bellows
S	Link length, shape factor, or maximum permissible total bellows stroke
t	Thickness
T	Torque
V	Velocity
w	Width
X	Angular misalignment
Y	Parallel misalignment
α	Rotational position
β	Torsional amplitude
γ	U-joint angle
δ	Deflection or U-joint angle
Δ	Deflection
ζ	Damping ratio
θ	Shaft or joint angle
θ_{eff}	Torsional equivalent angle
τ	Shear stress
ω	Angular velocity

29.1 GENERAL

29.1.1 System Requirements

When selecting a coupling, you have to consider all the system's requirements. It is not enough to know what the driver and load are and how big the shaft is. You must also know how the two halves are assembled and whether there is misalignment, as well as the system's operating range and the operating temperature.

Before you select a coupling, determine the following about the system:

1. *Driver* Type; electric motor, internal-combustion engine, number of cylinders, etc.
2. *Load* Fan, pump, rockcrusher, etc., to determine the inertias.
3. *Nominal torque* T_{kn} Continuous operating torque.
4. *Maximum torque* T_{max} Peak expected on startup, shutdown, overload, etc.
5. *Vibratory torque* T_{kw} Oscillating torque about the nominal $T_{kn} \pm T_{kw}$.
6. *Number of startups per hour.*

7. *Misalignment* Amount and type of misalignment between the driver and the load: parallel, angular, and/or axial.
8. *Type of mounting* Shaft to shaft, shaft to flywheel, blind fit, etc.
9. *Shaft size* Diameter of the shafts for both the driver and the load.
10. *Operating temperature* General operating temperature and whether the drive is enclosed (unventilated).
11. *Operating speed range* The upper and lower limits of the operating range.
12. *Service factor* A "fudge factor" designed to combine many of the above operating conditions and lump them into one multiplier to oversize the coupling in order to accommodate these parameters. Typical service factors are shown in Table 29.1.

29.1.2 Coupling Characteristics

Once the system requirements have been determined, check the characteristics of the coupling chosen to verify the selection. You should be able to check the following characteristics:

1. *Torque capacity*
2. *Bore size* Minimum and maximum bore
3. *Type of mounting* Mounting configurations available for any given coupling
4. *Maximum speed range*
5. *Misalignment* Degree of misalignment that can be accepted in mounting
6. *Flexible material* Capability of material to withstand heat or oil contamination; torsional stiffness

29.1.3 Selecting the Coupling

The first step is to make a preliminary selection based on the torque transmitted and the shaft dimensions. Then verify that the selection will satisfy the requirements for type of mount, degree of misalignment, operating speed, and operating temperature. Don't forget to check for the possibility of resonance.

Not all systems require all these steps. Smooth operating systems, such as electric motors driving small loads, are seldom subject to severe vibration. The natural frequency probably does not have to be checked.

As a simple guideline for determining system requirements for smooth systems, coupling manufacturers have developed the service factor. The service factor is a rough approximation of the temperature requirements, maximum torque, and natural frequency. It is stated as a multiplier, such as 1.5. To be sure the coupling you have selected is adequate, multiply the nominal torque required for the system by the service factor and select a coupling with that torque rating or better.

The service factor is adequate for some systems. Its drawbacks are that it is imprecise and, in severe applications, does not evaluate all the variables. Also, when you are selecting according to the service factor, be careful not to overspecify, getting more coupling than needed. This is not cost-effective.

Perhaps the most important thing to remember in selecting a coupling is that the coupling manufacturer can make a recommendation for you only based on the

TABLE 29.1 Service Factors and Load Classification for Flexible Couplings†

Agitators	
Pure liquids	1.0
Liquids, variable density	1.0
Barge puller	2.0
Beaters	1.5
Blowers	
Centrifugal	1.0
Lobe	1.25
Vane	1.25
Can-filling machinery	1.0
Car dumpers	2.5
Car pullers	1.5
Compressors	
Centrifugal	1.0
Lobe	1.25
Reciprocating	‡
Conveyors, uniformly loaded or fed	
Assembly	1.0
Belt	1.0
Screw	1.0
Bucket	1.25
Live roll, shaker and reciprocating	3.0
Conveyors (heavy-duty), not uniformly fed	
Assembly	1.2
Belt	1.2
Oven	1.2
Reciprocating	2.5
Screw	1.2
Shaker	3.0
Cranes and hoists	
Main hoists	2.0
Reversing	2.0
Skip	2.0
Trolley drive	2.0
Bridge drive	2.0
Slope	2.0
Crushers	
Ore	3.0
Stone	3.0
Dredges	
Cable reels	2.0
Conveyors	1.5
Cutter head drives	2.5
Maneuvering winches	1.5
Pumps	1.5
Elevators	
Bucket	1.5
Escalators	1.0
Freight and passenger	2.0
Evaporators	1.0
Fans	
Centrifugal	1.0
Fans (*cont.*)	
Cooling towers	2.0
Forced draft	1.5
Induced draft w/o damper control	2.0
Propellor	1.5
Induced draft w/damper control	1.25
Feeders	
Belt	1.0
Screw	1.0
Reciprocating	2.5
Generators	
Not welding	1.0
Welding	2.0
Hoist	1.5
Hammer mills	2.0
Kilns	1.5
Laundry washers, reversing	2.0
Line shafting any processing mach.	1.5
Lumber machinery	
Barkers	2.0
Edger feed	2.0
Live rolls	2.0
Planer	2.0
Slab conveyor	2.0
Machine tools	
Bending roll	2.0
Plate planer	2.0
Punch press gear driven	2.0
Tapping machinery	2.0
Other	
Main drive	1.5
Aux. drives	1.0
Metal-forming machines	
Draw bench carriage	2.0
Draw bench main drive	2.0
Extruder	2.0
Forming machinery	2.0
Slitters	1.5
Table conveyors	
Nonreversing	2.5
Reversing	2.5
Wire drawing	2.0
Wire winding	1.5
Coilers	1.5
Mills, rotary type	
Ball	2.0
Cement kilns	2.0
Dryers, coolers	2.0
Kilns	2.0
Pebble	2.0
Rolling	2.0
Tube	2.0
Tumbling	1.5

TABLE 29.1 Service Factors and Load Classification for Flexible Couplings† (*Continued*)

Mixers	
Concrete, cont.	1.75
Muller	1.5
Papermills	
Agitators (mixers)	1.2
Barker mech.	2.0
"Barking" drum spur gear	2.5
Beater and pulper	2.0
Calenders	1.5
Calenders, super	1.5
Converting machines	1.2
Conveyors	1.2
Dryers	1.5
Jordans	2.0
Log haul	2.0
Dresses	2.0
Reel	1.2
Winder	1.2
Printing presses	1.5
Pug mill	1.75
Pumps	
Centrifugal	1.0
Gear, rotary or vane	1.25
Reciprocating	
Pumps (*cont.*)	
1 cyl., single- or double-acting	2.0
2 cyl. single-acting	2.0
2 cyl. double-acting	1.75
3 or more cyl.	1.5
Rubber machinery	
Mixer	2.5
Rubber calender	2.0
Screens	
Air washing	1.0
Rotary stone or gravel	1.5
Vibrating	2.5
Water	1.0
Grizzly	2.0
Shredders	1.5
Steering gear	1.0
Stokers	1.0
Textile machinery	
Dryers	1.2
Dyeing mach.	1.2
Tumbling barrel	1.75
Windlass	2.0
Woodworking machinery	1.0

†The values of the service factors listed are intended only as a general guide. For systems which frequently use the peak torque capacity of the power source, check that this peak torque does not exceed the normal torque capacity of the coupling.

The values of the service factors given are to be used with prime movers such as electric motors, steam turbines, or internal combustion engines having four or more cylinders. For drives involving internal combustion engines of two cylinders, add 0.3 to values; and for a single-cylinder engine add 0.70.

‡Consult the manufacturer.

SOURCE: Ref. [29.1].

information you provide. A little time spent selecting the right coupling can save a lot of time and money later.

Selecting a flexible coupling involves more than meeting torque and shaft size requirements. It is also important to understand the functions of a flexible coupling in the system, the operating requirements of the system, and the characteristics of the coupling selected. Flexible couplings serve four main functions in a drive system:

1. They transmit torque and rotation from the drive to the load.
2. They dampen vibration.
3. They accommodate misalignment.
4. They influence the natural frequency of the system.

The torque-handling capacity of a given coupling design defines the basic size of a coupling. The nominal torque T_{kn} is the coupling's continuous load rating under conditions set by the manufacturer. The maximum torque rating T_{max} is the peak torque the coupling can handle on startup, shutdown, running through resonance, and momentary overloads. As defined in the German standards for elastomeric couplings, Ref. [29.2], a coupling should be able to withstand 10^5 cycles of maximum torque at a frequency of not more than 60 per hour. Vibratory torque ($\pm T_{kw}$) is the coupling vibratory rating at 10 hertz (Hz) for elastomeric couplings. The rotary output of the coupling may be uniform (constant velocity) or cyclic (e.g., Hooke's joint).

All drive systems experience some vibration. Vibration can exceed the limits of design, which can cause system failure. Flexible couplings are one method of dampening the amount of vibration from either the driver or the driven equipment.

When a flexible coupling is used, the vibration is transferred to a material which is designed to absorb it rather than transmit it through the entire drive. Soft materials, such as natural rubber, can absorb greater amounts of vibration than stiffer materials, such as Hytrel† or steel. As a comparison, the relative vibration damping capabilities of Buna N rubber, Hytrel, and steel are shown in the transmissibility chart of Fig. 29.1.

If a system has misalignment, there are two factors to consider. First, you must use a coupling that can operate between two misaligned shafts. Second, you must be sure that the coupling does not exert excessive forces on the equipment because of misalignment. Perfect alignment between the driver and the load is difficult to obtain and maintain over the life of the system. A cost-effective alternative to precise alignment is a coupling that can accommodate misalignment between two shafts. The amount of misalignment a coupling can accept varies. Steel drive plates, for example, can accept only misalignment equal to their machining tolerances, frequently as little as 0.005 inch (in) parallel. Other couplings can accommodate mis-

† Hytrel is a trademark of E.I. du Pont de Nemours.

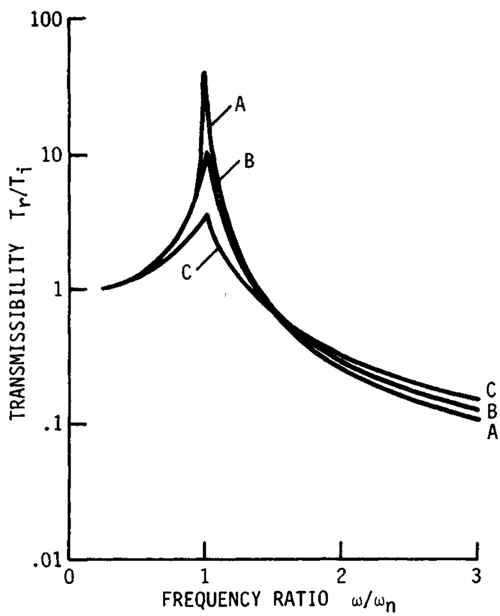


FIGURE 29.1 Effect of damping ratio on torque transmission. A, steel, $\xi = 0.01$; B, Hytrel, $\xi = 0.03$; C, Buna N rubber, $\xi = 0.13$, where T_r is the transmitted torque and T_i the input torque.

alignment up to 45°. The maximum allowable misalignment is a function of the percentage of torque capacity being utilized and the amount of vibratory torque the system is transmitting under perfect alignment.

If there is system misalignment, the material used in the coupling is important. Misalignment may cause radial forces to be exerted on the system. If the radial forces are too great, components such as bearings, seals, and shafts can experience undue stresses and fail prematurely. Different materials exert different radial forces; softer materials typically exert less radial force than stiff materials.

The natural frequency of a system can be altered by changing either the inertia of any of the components or the stiffness of the coupling used. See Chap. 38. Generally, after a system is designed, it is difficult and costly to change the inertia of the components. Therefore, coupling selection is frequently used to alter the natural frequency.

29.2 RIGID COUPLINGS

The solid coupling does not allow for misalignment, except perhaps axial, but enables the addition of one piece of equipment to another. In its simplest form, the rigid coupling is nothing more than a piece of bar stock bored to receive two shafts, as shown in Fig. 29.2. Its torque-handling capacity is limited only by the strength of the material used to make the connection. The coupling is installed on one shaft before the equipment is lined up, and the mating equipment is brought into position without much chance of accurate alignment when the equipment is bolted into position.

The maximum shear stress occurs at the outer radius of the coupling and at the interface of the two bores. This stress can be derived from the torsion formula (see Chap. 49) and is

$$\tau_{\max} = \frac{TD_o}{2J} \tag{29.1}$$

where J, the polar second moment of the area, is

$$J = \frac{\pi}{32}(D_o^4 - D_i^4) \tag{29.2}$$

The coupling must be sized so that, typically, the stress given by Eq. (29.1) does not exceed 10 percent of the ultimate tensile strength of the material, as shown in Table 29.2; but see Chap. 12.

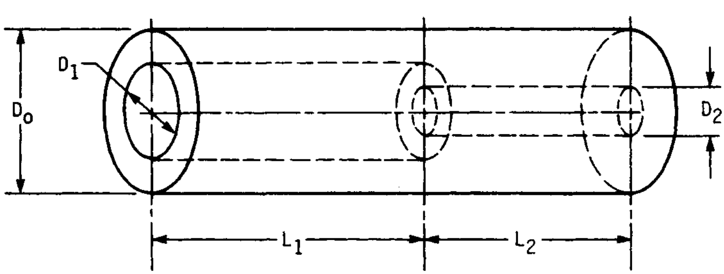


FIGURE 29.2 Schematic view of a rigid coupling.

TABLE 29.2 Maximum Allowable Shear Stress for Some Typical Materials

Material	Stress, psi	Material	Stress, psi
Steel	8000	Powdered iron (Fe-Cu)	4000
Ductile iron (60-45-12)	6000	Aluminum (SAE 380)	4000
Cast iron (Class 40)	4500	Tobin brass	3500

Other factors to consider are the length of engagement into the coupling. The shear stress over the keyway must not exceed the allowable shear stress as given above. Based on Fig. 29.3, the centroidal radius is

$$R_c = \frac{1}{2}\left(\frac{D_o}{2} + \frac{D_i}{2} + h\right) \tag{29.3}$$

The centroid of the bearing area is at radius $(D_i + h)/2$. If the transmitted torque is T, then the compressive force F is $2T/(D_i + h)$. The bearing stress σ_b is

$$\sigma_b = \frac{F}{A} = \frac{4T}{wL(D_i + h)} \tag{29.4}$$

The allowable compressive stress from distortion energy theory of failure is $\sigma_{all} = \tau_{all}/0.577$. Combining this with Eq. (29.4) gives

$$\tau_{all} = \frac{0.577(4)T}{wL(D_i + h)} \tag{29.5}$$

with τ_{all} coming from Table 29.2.

Next, the length of key stock, for keyed shafts, must be examined to keep its shear loading from exceeding the allowable shear stress. Referring to Fig. 29.4, we note that the shear force is $F = T/(D_i/2) = 2T/D_i$. Therefore the average shear stress is

$$\tau = \frac{F}{A} = \frac{2T}{wLD_i} \tag{29.6}$$

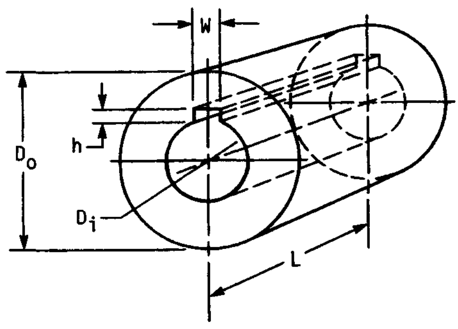


FIGURE 29.3 Portion of coupling showing keyway.

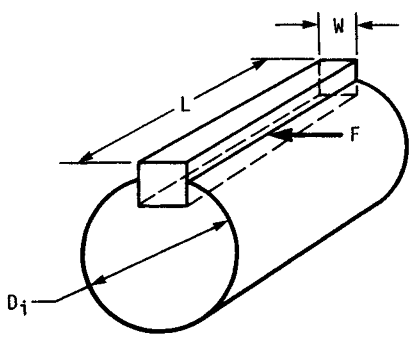


FIGURE 29.4 Portion of shaft showing key.

Both keys must be checked, although experience has shown that small-diameter shafts are more prone to failure of the key and keyway when these precautions are not followed because of their normally smaller key width and length of engagement. As a rule of thumb, the maximum allowable shear stress for some typical materials is shown in Table 29.2.

The *ribbed, hinged,* and *flanged* couplings are shown in Figs. 29.5, 29.6, and 29.7, respectively. These can be analyzed using the same approach as described above.

29.3 FLEXIBLE METALLIC COUPLINGS

29.3.1 Flexible Disk and Link Couplings

In this coupling (Fig. 29.8), misalignment is accommodated by the flexing of steel laminations. Parallel misalignment capacity is virtually zero unless two separated disk packs are used, in which case parallel misalignment is seen in the form of angular misalignment of each pack. This type of coupling can support large imposed radial loads, such as in rolling mills or long, floating shafts. The disk packs can be made from any material and are frequently manufactured from stainless steel for severe service. This coupling requires no lubrication.

The large radial loads imposed by long sections of tubing connecting to widely separated disk packs [up to 20 feet (ft)] are due to the heavy wall section necessary to give the tubing (or shafting) the necessary rigidity to resist whirling due to the

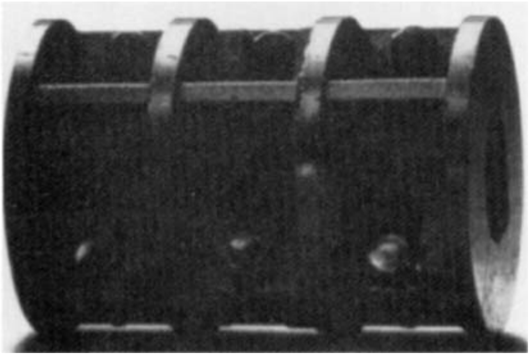


FIGURE 29.5 This ribbed coupling is made of two identical halves, split axially, and bolted together after the shafts have been aligned.

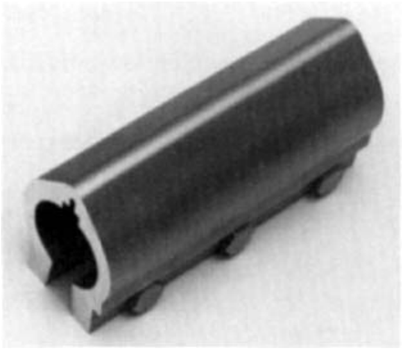


FIGURE 29.6 This hinged coupling is used mostly for light-duty applications. *(CraneVeyor Corp.)*

weight of the tubing (shafting). Specifically, the whirling speed of a uniform tube due to its weight is

$$n_c = \frac{60}{2\pi}\sqrt{\frac{\Delta}{g}} \qquad (29.7)$$

where Δ = static deflection of the tube due to its own weight. See Chap. 50 for deflection formulas, and Chap. 37 for method.

The standard rule of thumb is to keep the critical whirling speed at least 50 percent above the operating speed for subcritical running, or 40 percent below the operating speed for supercritical speeds. This forbidden range of

$$0.6n_c < n_c < 1.4n_c$$

corresponds to the amplification region of a lightly damped resonance curve, as shown in Fig. 29.9. Thus, for a whirling speed of 1800 revolutions per minute (r/min), the operating speed must not be in the range of 1280 to 2700 r/min.

The link coupling in Fig. 29.10 is similar to the metallic disk coupling except that the disk is replaced by links connecting the two shaft hubs. This coupling can be misaligned laterally, considerably more than the disk type. Both the disk and the link type carry torque in tension and compression in alternating arms. Proper bolt torque of the axial bolts holding the links or disks to the hubs is important. Insufficient torque may cause fretting from relative motion between the links or disks. Too much bolt clamping weakens the links or disks at their connecting points as a result of excessive compressive stress.

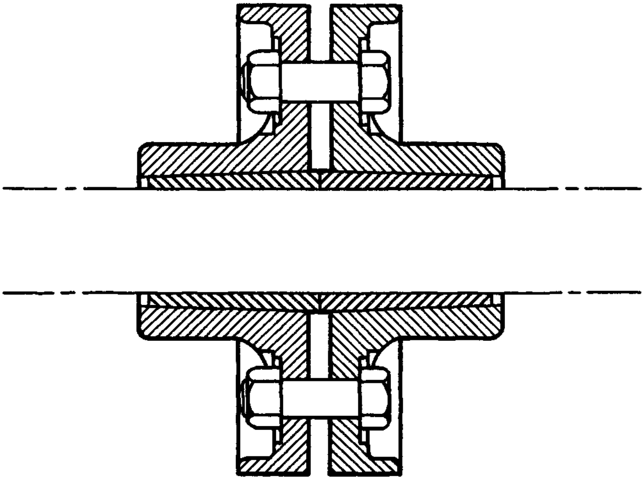
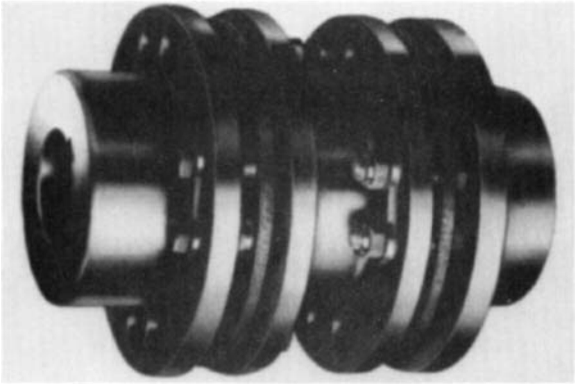
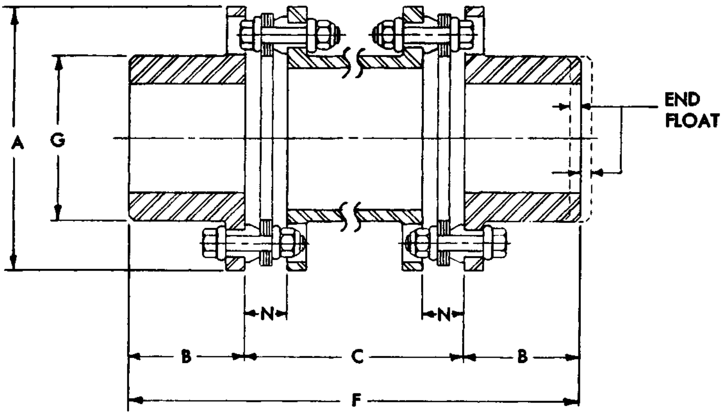


FIGURE 29.7 Schematic view of a flanged sleeve coupling.



(a)



(b)

FIGURE 29.8 (*a*) Flexible-disk coupling; (*b*) cross section. *(Rexnord, Inc., Coupling Division.)*

29.3.2 Chain, Grid, and Beam Couplings

The chain coupling of Fig. 29.11 consists of two sprockets joined by an endless double-roller chain or inverted-tooth silent chain. This type of coupling will accommodate small amounts of angular, axial, and radial misalignment, which is provided by clearances between interfacing surfaces of the component parts.

For maximum service life, chain coupling sprockets should have hardened teeth. The coupling should be lubricated and enclosed in a greasetight case. Chain couplings can be assembled by using unhardened sprockets and operated without lubrication or a cover. This can be hazardous and can result in injury to personnel as well as a short coupling service life. This author has seen many such worn-out couplings. The availability of chain couplings is very good worldwide. Most manufacturers publish horsepower ratings to aid in proper coupling selection.

In the grid coupling (Fig. 29.12) the gears are separated by a specific minimum distance that allows for misalignment (Fig. 29.13). Large axial misalignment is

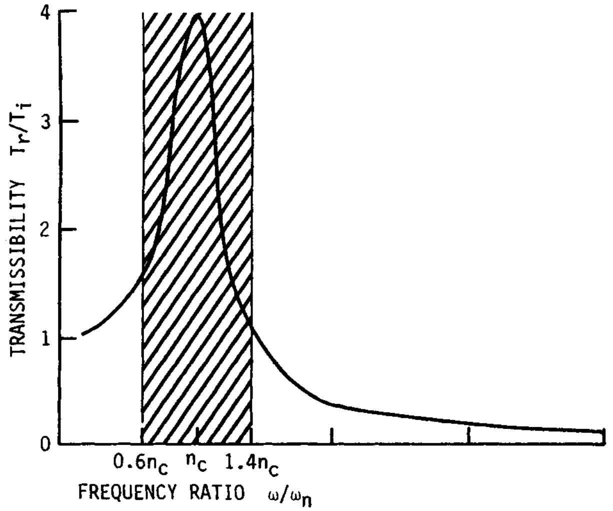


FIGURE 29.9 Lightly damped resonance curve showing forbidden speed range $0.6n_c < n_c < 1.4n_c$.

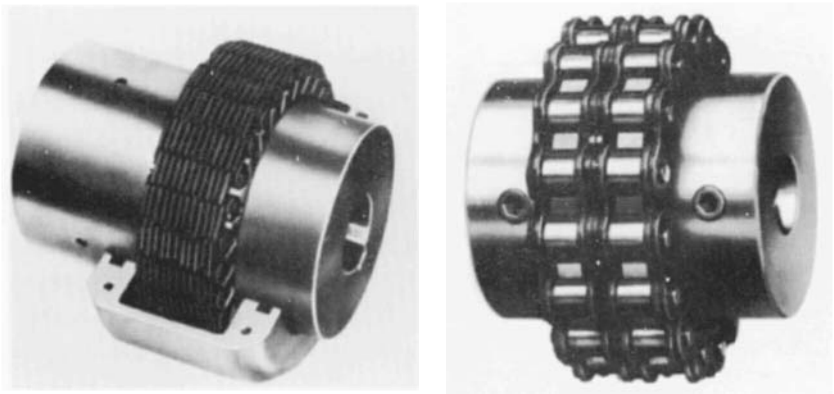
accomplished by sliding of the gear teeth in the rather long grid area. This coupling requires a guard (which is supplied) and lubrication.

The beam coupling is a single bar of metal cut as shown in Fig. 29.14 so that a continuous helical coil is produced. This coupling is torsionally rigid and free from backlash and provides constant velocity. Two designs exist for this type of coupling, one for maximum misalignment, the other for maximum torque (for the package size where axial movement is not a requirement). This coupling requires no lubrication. Speeds to 25 000 rev/min are possible depending on the coupling size.

FIGURE 29.10 Link coupling. *(Eaton Corp., Industrial Drives Operation.)*

29.3.3 Diaphragm and Hydraulic Couplings

As the name suggests, diaphragm couplings are made of a thin diaphragm or multiple thin diaphragms (see Fig. 29.15). Normally the diaphragms are made of metal. The diaphragms may be straight-sided, contoured, tapered, or convoluted; they may have various cutouts in them or take on many other forms. This coupling is connected to one shaft at the periphery [outer diameter (OD)] while the inner diameter (ID) is connected to the shaft or to a spacer piece, which may connect to another



(a) (b)

FIGURE 29.11 (*a*) Silent-chain coupling; (*b*) roller-chain coupling. *(Morse Industrial Products, Borg-Warner Corp.)*

diaphragm(s). This coupling is most often used in pairs (two flex elements), which converts parallel misalignment to angular misalignment between two flex elements. Misalignment is accommodated by stretching (straight, contoured, or tapered diaphragms or unrolling convoluted diaphragms) the diaphragm material. This type of coupling requires no lubrication and is considered torsionally rigid.

The hydraulic coupling consists of two sleeves, one with a tapered OD and one with a tapered ID, which slide over one another, as shown in Fig. 29.16. Oil is forced, under pressure, between the two sleeves to allow the outer sleeve to be positioned at a predetermined position on the inner sleeve. The pressure is released, and the outer sleeve firmly compresses the inner sleeve and shafting. In larger couplings, oil is also forced into a piston chamber to force the outer sleeve into position. To remove the coupling, the area between the sleeves is repressurized, and the outer sleeve can be slid away, releasing the coupling.

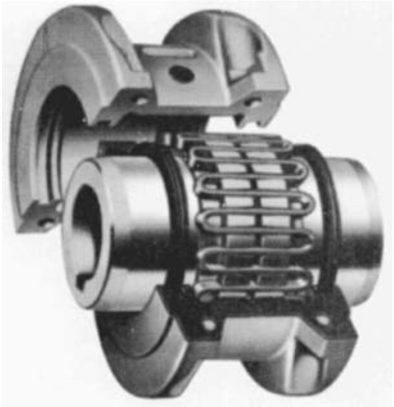


FIGURE 29.12 Metallic grid coupling with cover removed to show grid detail. *(Falk Corp.)*

29.3.4 Gear Couplings

Double-engagement gear couplings, as shown in Fig. 29.17, transmit more power per unit volume and unit weight than any other flexible coupling design, because of their relatively small OD (compared with other types of similar horsepower). The basic design consists of two gear-type hubs (similar to spur gears) loosely connected by an internal-spline sleeve, which could be one piece or two internal-spline mating flanges bolted together.

Clearance between the mating teeth in the hub and the sleeve allows this type of coupling to absorb angular, parallel,

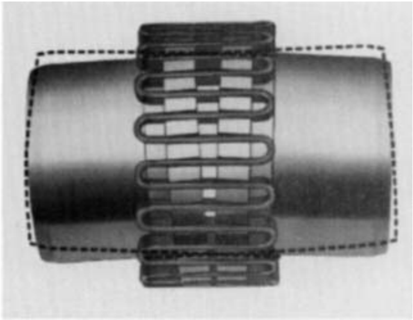


FIGURE 29.13 How the grid coupling accommodates misalignment. *(Falk Corp.)*

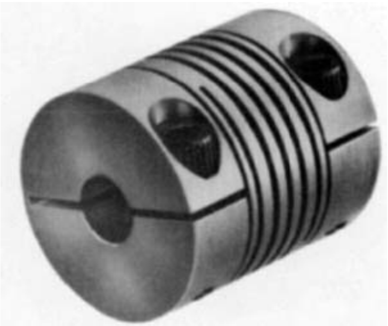


FIGURE 29.14 Beam coupling. *(Helical Products Corp.)*

and axial misalignment, as shown in Fig. 29.18. There is no relative rotation between the gear teeth, as in a normal gear set. Various tooth profiles (including crowned and/or barrel-shaped teeth) or changes in pressure angle allow for different misalignment, life, and load capacities. Straight-tooth couplings allow misalignment of 1° per gear mesh; with barrel-shaped teeth on the hub and straight teeth on the sleeve, 6° per mesh can be allowed.

With perfect alignment, all the teeth in the coupling are in contact, and the load is evenly distributed among them. Misalignment concentrates the load on just a few teeth; the number of teeth under load is a function of misalignment and load. The greater the misalignment (angular and parallel), the fewer the number of teeth in contact and the higher the load per tooth. Barrel-shaped teeth distribute the load over a larger area per tooth and may allow a greater number of teeth to be in contact under misaligned conditions, as shown in Fig. 29.19.

The two gear meshes can be separated by large distances, as shown in Fig. 29.20. In this case, two single-engagement couplings are connected by a floating shaft. For

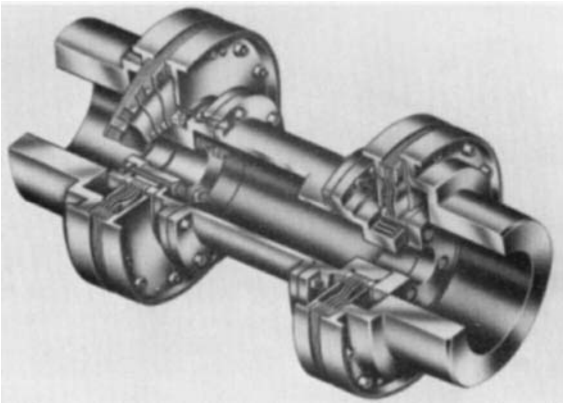


FIGURE 29.15 Cutaway view of diaphragm coupling assembly showing multiple convoluted diaphragms. *(Zurn Industries, Inc., Mechanical Drives Div.)*

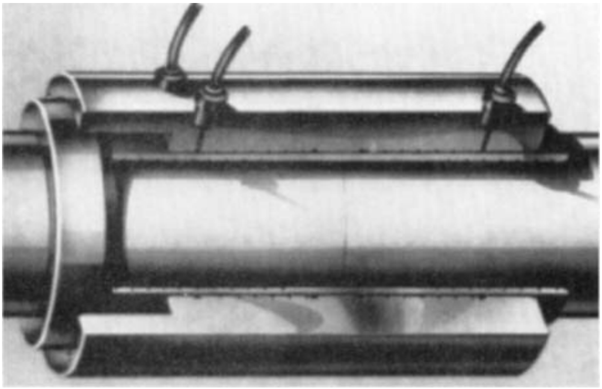


FIGURE 29.16 Hydraulic coupling; cutaway shows oil forced between inner and outer tapered sleeves. Note the oil piston chamber at left. *(SKF Industries.)*

this style coupling, large amounts of parallel misalignment are made possible by converting the angular misalignment capacity per mesh to parallel misalignment.

Parallel misalignment capacity for one single-engagement coupling is virtually nonexistent, however, and these couplings must be used in pairs, as shown in Fig. 29.20, to handle parallel misalignment.

Gear couplings must be lubricated for proper operation. Because of the high contact pressures obtained under misaligned conditions, only extreme-pressure (EP) greases should be used with gear couplings operating at maximum load. At high speeds (over 25 000 rpm), centrifugal effects separate the filler (soap) from the oil in most greases; the filler then collects between the teeth, preventing the oil from lubricating this highly loaded area. To overcome this problem, most high-speed gear couplings use a circulating oil system. The centrifugal effect still separates the fine particles from the oil, even in finely filtered systems. This sludge buildup necessitates cleaning of the teeth at regular intervals to prevent premature coupling failure.

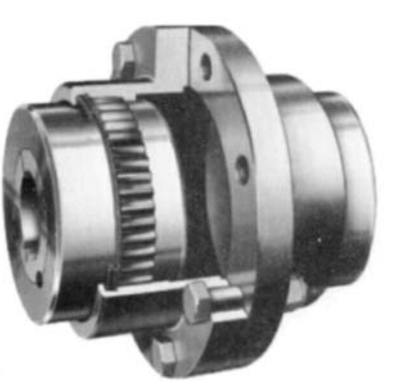


FIGURE 29.17 Cutaway of flange-type gear coupling. *(Dodge Division, Reliance Electric.)*

Gear couplings, while inherently balanced, being machined all over and self-centering, may still require balancing to remove any residual unbalance due to bore runout. The magnitude F of this unbalanced or centrifugal force is

$$F = me\omega^2 \qquad (29.8)$$

where m = mass of the coupling, e = eccentricity, and ω = angular velocity in radians per second. See also Chaps. 37 and 38.

29.3.5 Spring and Flexible Shaft

Flexible shafts are constructed from a casing and a core, which is a series of

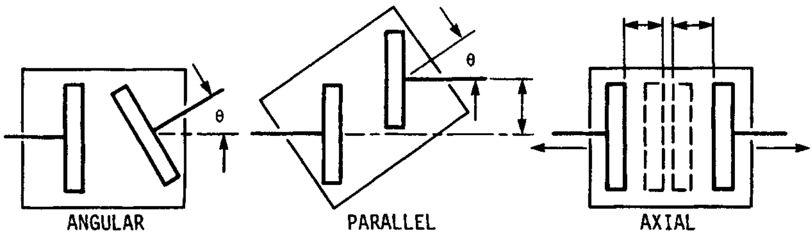


FIGURE 29.18 How double-engagement gear couplings accommodate angular, parallel, and axial misalignment.

multistranded layers of wire successively wrapped about a single central wire. Each wire layer is wound opposite to and at right angles to the layer beneath it to transmit maximum power and retain the greater flexibility.

The casing protects the rotating core from dust and moisture, but does not rotate itself. It is also reinforced to support the core and prevent helixing under torque load. *Helixing* is the tendency for a rope, or wire, to bend back on itself when subjected to torsional stress (Fig. 29.21). The casing also provides a cavity for grease to lubricate the rotating core. The core is attached to the hub on either end and then connected to the equipment.

The power transmission capacity of flexible shafting is limited only by the core construction, minimum radius of curvature of the shafting, and maximum unsupported length.

Flexible shafts are commercially available with ratings up to 1500 pound-inches (lb·in) at 440 r/min. Such a shaft is 1½ in [38 millimeters (mm)] in diameter and has a minimum operating radius R of 24 in (600 mm). In Fig. 29.22, let R be the required

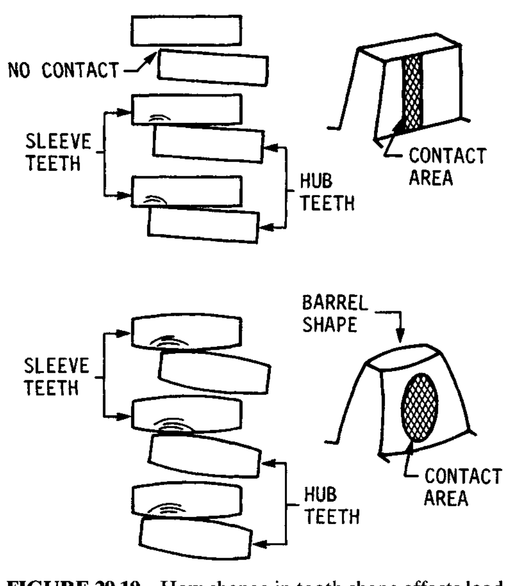


FIGURE 29.19 How change in tooth shape affects load distribution on the teeth of the gear coupling.

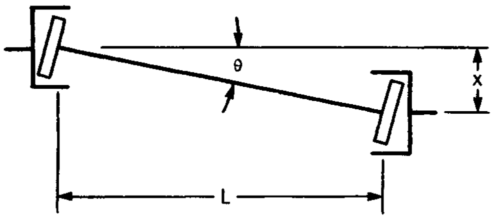


FIGURE 29.20 Diagram shows how parallel misalignment is converted to angular misalignment in each gear coupling mesh. For example, for an extended floating shaft with $L = 12$ in and $\theta = 1°$, the misalignment is $x = L \tan\theta = 0.20$ in. For a standard double-engagement coupling with $L = 2$ in and $\theta = 1°$, $x = 0.03$, which is significantly less.

operating radius corresponding to a misalignment A and a spacing D between equipment. Then the following relations can be derived from Fig. 29.22:

$$R = \frac{D^2 + A^2}{4A} \tag{29.9}$$

$$A = 2R - (4R^2 - D^2)^{1/2} \tag{29.10}$$

$$D = [A(4R - A)]^{1/2} \tag{29.11}$$

$$C_L = \frac{\pi R}{90} \sin^{-1} \frac{D}{2R} \tag{29.12}$$

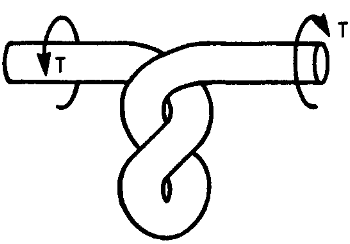


FIGURE 29.21 "Helixed" flexible shaft made of wire or rope.

where C_L = flexible length between the equipment.

If the casing is eliminated from the flexible shaft, a flexible coupling is created with much shorter maximum length (owing to lack of support and antihelixing without the casing). Typical commercial availability of this type of coupling is limited to 50 lb·in and 16-in lengths with an 8-in minimum operating radius.

Another similar coupling, the Uniflex,† consists of three layers of springs, each with three rectangular wires wound around an open-air core. This coupling (Fig. 29.23) also runs without a casing and has maximum speeds of up to 20 000 rpm, depending on size. It is relatively free from backlash and winds up about 1° at rated torque. This design uses spring elements up to 3 in long with rated torque up to 2000 lb·in at up to 4.5° misalignment.

29.3.6 Bellows Coupling

This type of coupling, shown in Fig. 29.24, consists of an all-metal circular bellows attached to two hubs. The design exhibits zero backlash and constant-velocity operation and is torsionally rigid. However, commercial couplings are typically rated to a maximum of 30 lb·in.

† Uniflex is a trademark of Lovejoy, Inc.

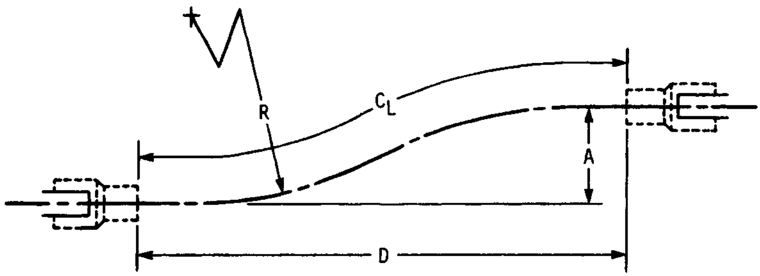


FIGURE 29.22 Maximum parallel misalignment of flexible shafting. *(Stow Manufacturing Company)*

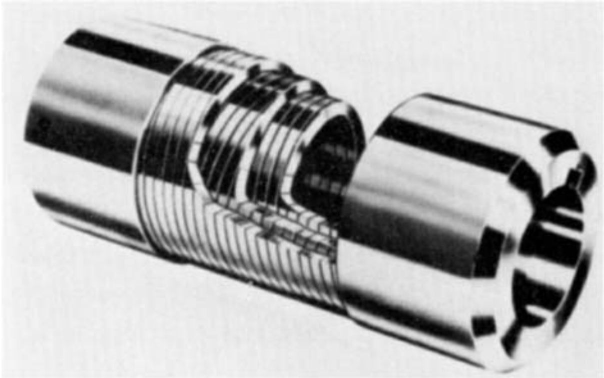


FIGURE 29.23 Uniflex flexible-spring coupling. *(Lovejoy, Inc.)*

The torque rating is obtained from

$$T = \frac{8pkD_o^2}{\ell} \tag{29.13}$$

where T = torque, in · oz
p = pitch, in
k = spring rate per convolution, oz/in
D_o = outside diameter, in
ℓ = length of a single convolution

The windup of the coupling (angular deflection) is measured in seconds of arc per inch-ounce of torque and is

$$u = \frac{0.08\ell}{(D_o + D_i)^3 t} \tag{29.14}$$

where u = windup, seconds/(in · oz)
D_i = inside diameter, in
t = thickness of bellows, in

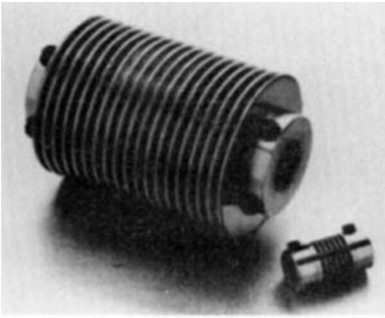


FIGURE 29.24 A bellows coupling. *(Servometer Corp.)*

Equations (29.15) and (29.16), which follow, apply to the determination of the life of this type of coupling. This life is dependent on the flexing motion due to angular and parallel misalignment, as shown in Fig. 29.25. The formula gives the operating misalignment corresponding to 10^5 flexing cycles (5×10^4 r). For 10^8 cycles, derate by 24 percent. Now,

$$X = \frac{71.6NS}{D_o} \tag{29.15}$$

$$Y = \frac{N^2 nS}{D_o} \tag{29.16}$$

where X = angular misalignment in degrees, Y = parallel misalignment, N = number of bellows convolutions, S = maximum total permissible bellows stroke, D_o = outside bellows diameter, and n = r/min.

29.4 FLEXIBLE ELASTOMERIC COUPLINGS

In this type of coupling, it is an elastomeric cushioning material—rather than only a metal part, as in the rigid, flexible metallic, or universal-joint (U-joint) couplings—that is subjected to the dynamic stresses of the operating system. These flexible elastomeric couplings need no lubrication and only periodic visual inspection for maintenance. They are available as compression, precompression, shear, and tension types.

The compression coupling is also known as the jaw, or block, coupling (Fig. 29.26). In this type of coupling, elastomeric cushioning blocks, rollers, or "spiders" are compressed between alternating pairs of fingers on the two hubs of the coupling.

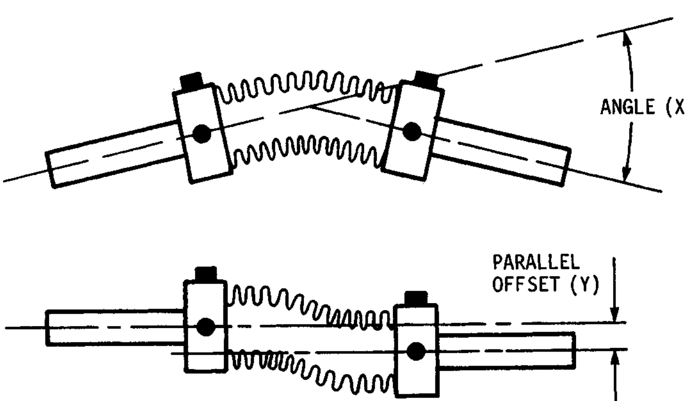


FIGURE 29.25 Bellows coupling with angular misalignment X and parallel misalignment Y. *(Servometer Corp.)*

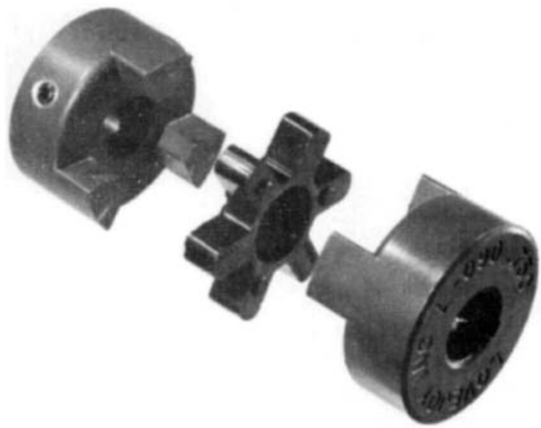


FIGURE 29.26 Exploded view of jaw-type compression coupling showing the two hubs and the flexible spider insert. *(Lovejoy Inc.)*

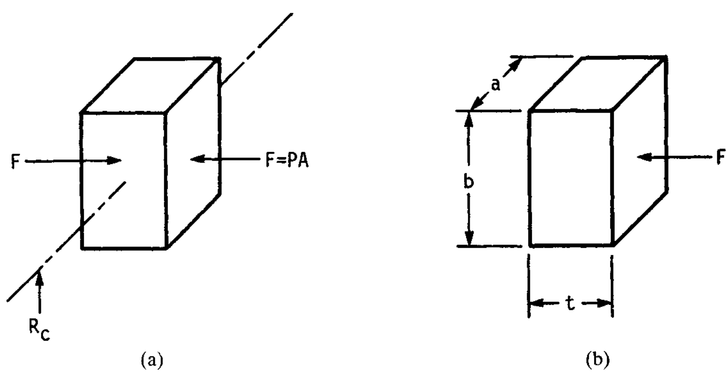


FIGURE 29.27 (*a*) Loading of one cushion at distance R_c from the center of a compression block coupling; (*b*) parameters used to calculate shape factor S for materials such as rubber where load area is ab and force F compresses the material at a thickness t.

The elastomeric elements can be of varying degrees of hardness or different materials to suit load-carrying capacities and temperature and chemical resistance requirements, in addition to torsional stiffness. Overall major dimensions may be altered by changing the number of active elements, element size, and radius on which the load is applied. The loading on one cushion is shown in Fig. 29.27*a*, and the torque rating of the coupling is

$$T = NPAR_c \tag{29.17}$$

where T = torque, N = number of cushions, P = pressure, and R_c = centroidal radius.

From Eq. (29.17) we can see that large couplings with many active elements are capable of handling very large loads. Typically, the design limit for rubber in compression is 300 pounds per square inch (psi), while a plastic like Hytrel is capable of 1100 psi. The recommended load capacity of some elastomers is dependent on the *shape factor S*, which is the ratio of one load area to the total free area of the cushion. See Fig. 29.27*b*. Thus

$$S = \frac{ab}{2t(a+b)} \tag{29.18}$$

As this ratio changes from a thin plate (high S) to a fat block (low S), the maximum allowable stress E decreases, and the deformation of the block (compression) increases, as shown in Fig. 29.28.

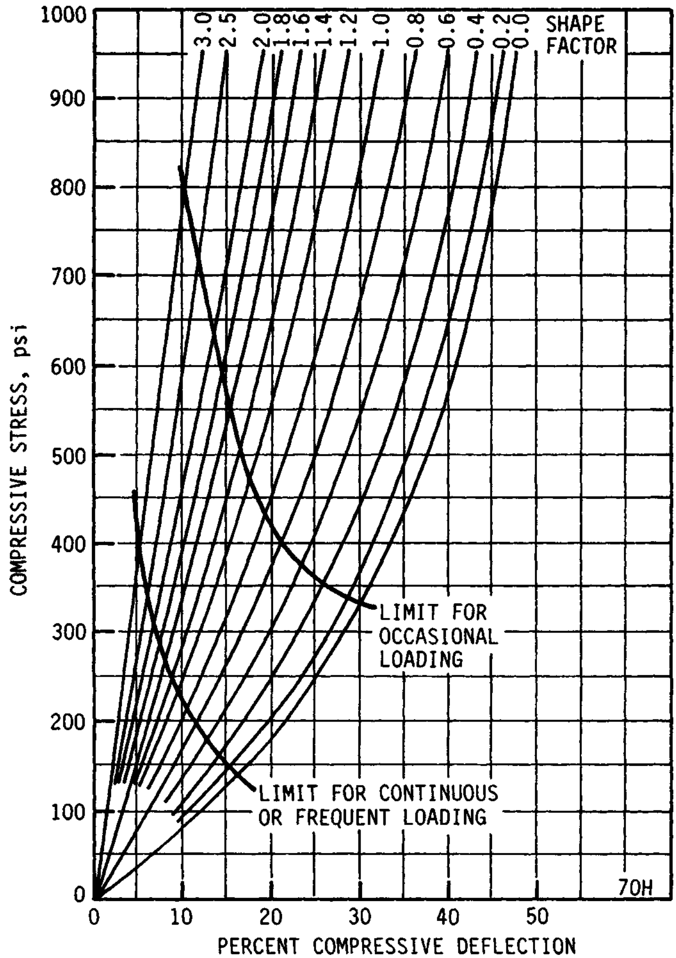


FIGURE 29.28 How the shape factor changes the maximum allowable stress in 70 durometer (Shore A) rubber. *(From Ref. [29.5], p. 77.)*

The basic misalignment capacity of these couplings is determined by the mating tolerances of the two hubs, the elastomeric element, and the particular jaw design. The larger the difference between the cushion and the jaw dimensions, the greater the misalignment that can be accommodated without exerting undue reaction forces on the system. However, this arrangement also leads to noisy operation, pulsating power transmission, and the transmission of shock through the system. A tradeoff between these effects is made in the final design. Alternatively, thick blocks of material can be used and deformed under misalignment. This may cause high restoring forces but decrease noise and pulsation of power owing to loading and unloading of the flexible elements. These couplings exert some axial thrust with increasing torque unless special jaws are used.

The shear unclamped coupling shown in Fig. 29.29 transmits power through shear in the elastomer, which can be rubber or a suitable plastic. Since rubber in shear can be loaded to only 20 percent of the load permitted on rubber in compression (70 versus 350 psi), the shear coupling is correspondingly larger in diameter and has a thicker element cross section than a corresponding compression coupling. The basic rating for this coupling type is

$$T = \frac{\pi t_{\max}(r_0^4 - r_i^4)}{2r_0} \tag{29.19}$$

These couplings may have both internal and external teeth in the hubs which mate to teeth in the flexible element, or the element may be bonded to the internal hubs and friction-fit into the outer hub (Fig. 29.30). For proper operation of the toothed-type coupling, the element must be twisted because of torque; this causes the gear teeth to rest properly. If the coupling is too large for a particular application (high service factor), the teeth will not load properly and the coupling will wear out prematurely. The toothed type of coupling is the double-engagement type and may allow more parallel and angular misalignment than a corresponding compression coupling.

The bonded type is designed to slip when torque exceeds the maximum rating; however, the heat generated by prolonged overload will destroy the coupling. The

FIGURE 29.29 Shear unclamped coupling. *(T. B. Wood's Sons, Inc.)*

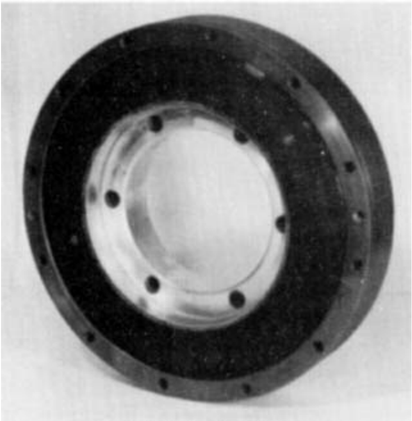


FIGURE 29.30 Bonded type of a shear unclamped coupling. *(Lord Corp., Industrial Products Div.)*

bonded type is torsionally softer than the corresponding toothed type, with windup of 15° to 20° versus 5° to 6° for the toothed type. This coupling is not fail-safe; when the flexible element fails, the driver and driven equipment are no longer connected.

The shear clamped or tire (torus) coupling is designed around a tire-shaped element, as shown in Figs. 29.31 and 29.32. The tire beads are clamped to a hub on either side, and torque is transmitted by shear and tension in the tire body. These couplings may be reinforced or unreinforced (without cords); in either case, the geometry allows very short overall length for a given torque capacity at the expense of diameter, which becomes correspondingly larger. This large diameter can cause some problems resulting from axial pull from the coupling as it is rotated at high speed. Centrifugal forces on the tire cause the two hubs to collapse inward as the speed is increased. The tire shape allows lower restoring force to larger misalignments than the shear unclamped or compression coupling.

The tension coupling shown in Fig. 29.33 is very similar to the link coupling (see Sec. 29.3.1), in that alternate legs (links) are in compression or tension. The links may have reinforcing cords to increase their tensile rating. In this coupling the rating is

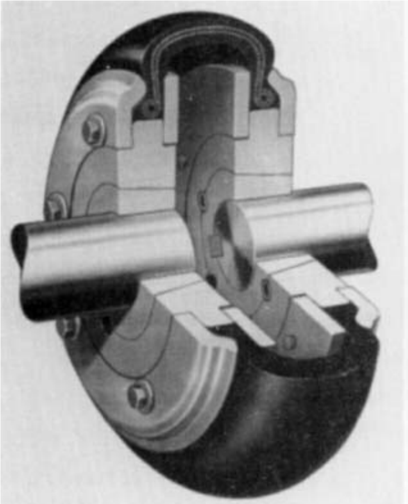


FIGURE 29.31 Tire coupling. *(Dodge Div., Reliance Electric.)*

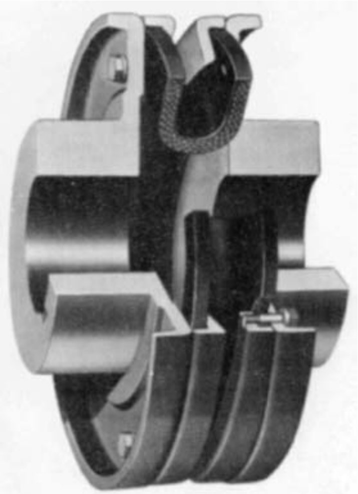


FIGURE 29.32 Torus coupling. *(Falk Corp.)*

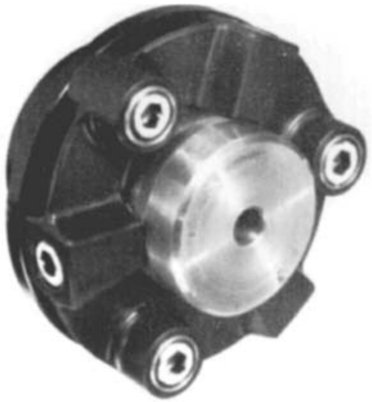


FIGURE 29.33 Tension coupling. *(Lovejoy Inc.)*

determined by the compression buckling of the leg pairs or the tensile strength of the other leg pairs, whichever is less. These couplings may be made of plastic, although rubber treated against ozone, when used with reinforcing cords, is also available.

To eliminate the effect of tension on rubber in a design similar to the one in Fig. 29.33, the rubber element is precompressed by assembling the element into a state smaller than the free state. This allows the "trailing leg" (formerly the tension leg) to stretch almost to neutral but never into tension, which would promote early failure.

In the radially restrained type, a tight metal band is slipped around the OD of the element while it is being held in a radially compressed state. After assembly to the mating hub, the band is snipped off, and the assembly bolts hold the element in precompression. Alternatively, radial bolts are used to draw the rubber element into compression. See Fig. 29.34.

Both these designs use fairly thick cross sections for their flexing elements and are quite compliant in all forms of misalignment. Rubber hardness and type can be changed to alter torque capacity, damping, and chemical-temperature resistance.

Alternatively, axial restraint can be used. In this design, the rubber element is installed between two hubs which are set a fixed distance apart, as shown in Fig. 29.35. The rubber element is then installed by means of bolts into this space, which is smaller than the free state of the rubber. This design is also very compliant, but generates an axial force because of the forced axial compression of the element.

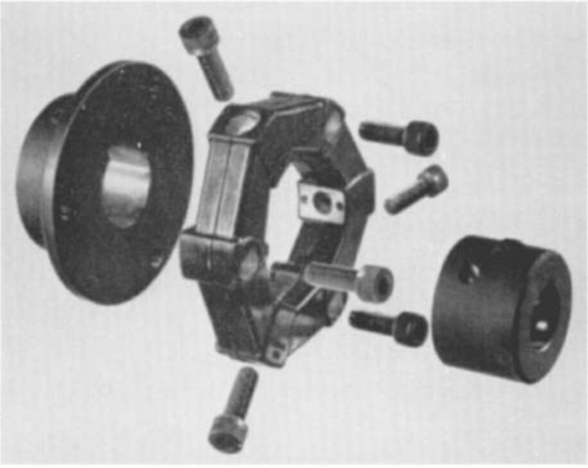


FIGURE 29.34 Precompressed radially restrained coupling. *(Lovejoy, Inc.)*

29.5 UNIVERSAL JOINTS AND ROTATING-LINK COUPLINGS

Another class of shaft connectors is composed of the linkage types, which include universal joints (U-joints) and rotating-link couplings. These couplings all rely on translating misalignment to relative rotation between parts of the coupling. The simplest universal joint is the one for a compass suspension universal joint described by Geronimo Cardano in the sixteenth century and explained mathematically by Robert Hooke in the seventeenth century. The universal joint basically consists of two shaft connections (yokes) which pivot about a pair of centrally located bearings. See Fig. 29.36.

FIGURE 29.35 Precompressed axially restrained coupling. *(Koppers Co. Inc., Power Transmission Div.)*

Shafts A and B are inclined to each other at angle θ (shaft centerlines must intersect); when shaft A is rotated, the center member (spider) causes shaft B to rotate by virtue of the connection. The output speed of shaft B is not constant when the shafts are at an angle. Specifically, the output velocity is

$$\omega_0 = \frac{\omega_i \cos\theta}{1 - \sin^2\alpha \sin^2\theta} \qquad (29.20)$$

where α = rotation position angle and θ = joint or shaft angle. In addition, the change in velocity peaks twice during every revolution. This leads to a second-order harmonic excitation frequency for the drive system. Specifically, the torsional magnitude β of this excitation is

$$\beta = \tan^{-1}(\sec\theta)^{1/2} - \tan^{-1}(\cos\theta)^{1/2} \qquad (29.21)$$

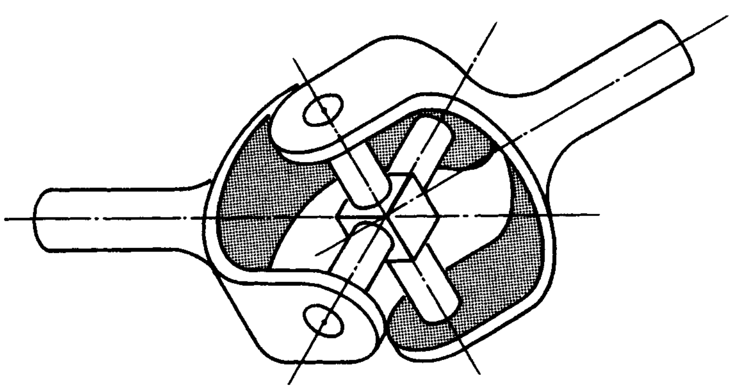


FIGURE 29.36 Basic Hooke's or Cardan universal joint.

The single Hooke's universal joint is not capable of accepting parallel misalignment (shaft centerlines do not meet at pivot member) except for very minor amounts resulting from manufacturing tolerances. Axial misalignment is usually compensated for by using a sliding connection on the input-output shafts or between the two universal joints.

When two universal joints are connected in series, parallel misalignment can be compensated for, because the parallel misalignment is converted to angular in each joint (see Fig. 29.37). By connecting two universal joints in this manner, it is possible to obtain approximately uniform output velocity between *A* and *C*. Basically, if joints *A* and *B* are aligned such that the pivot pins in their input and output yokes, respectively, are in the same plane, then the nonuniform output of joint *A* into shaft *B* is transformed to uniform output in joint *C*. However, if any angle exists between the joint *A* input yoke plane and the output yoke plane of joint *C*, then a nonuniform output will be generated. That nonuniform velocity and angle are, if we refer to Fig. 29.37*c*,

$$\omega_0 = \frac{\omega_i \cos \theta_{\text{eff}}}{1 - \sin^2 \alpha \sin^2 \theta} \tag{29.22}$$

where θ_{eff} = equivalent torsional angle given by

$$\theta_{\text{eff}} = (\delta^2 - \gamma^2)^{1/2} \tag{29.23}$$

Note that δ and γ are the U-joint angles as given in Fig. 29.37*c*.

29.5.1 Pin and Block

In its simplest form, the universal joint consists of two yokes and four pivot pins. The pivot pins ride in holes in the yoke and oscillate as the universal joint rotates. All the bearing takes place between the pins and the yoke, without the use of rolling-element bearings. In this type of joint, a thin film of lubricant or surface coating is all that prevents disaster from occurring, namely, the pin freezing in the journal (yoke) because of galling, corrosion, or adhesive transfer of metal. A first approximation to bearing pressure and velocity is as follows (shown in Fig. 29.38):

$$PV = \frac{Tn\theta}{6875 R_c t} \tag{29.24}$$

where T = operating torque, n = operating speed, θ = joint angle in degrees, t = yoke ear thickness, and R_c = the distance, as shown on Fig. 29.38. Since the velocity is actually nonuniform, the average speed of the pin-yoke oscillation has been used. We can see that the product $n\theta$, speed in revolutions per minute times angle in degrees, becomes a convenient unit to relate to the torque capacity. In fact, that is the typical unit used by most manufacturers for their ratings; see Fig. 29.39. These types of rating curves must be used with care, since a practical limit on angle limits the $n\theta$ parameter and the bearing area limits the PV parameter. A typical load-rating formula, given PV = 1000 psi · in/s, is

$$T = (0.6875 \times 10^7) \left(\frac{R_c t}{n\theta} \right) \tag{29.25}$$

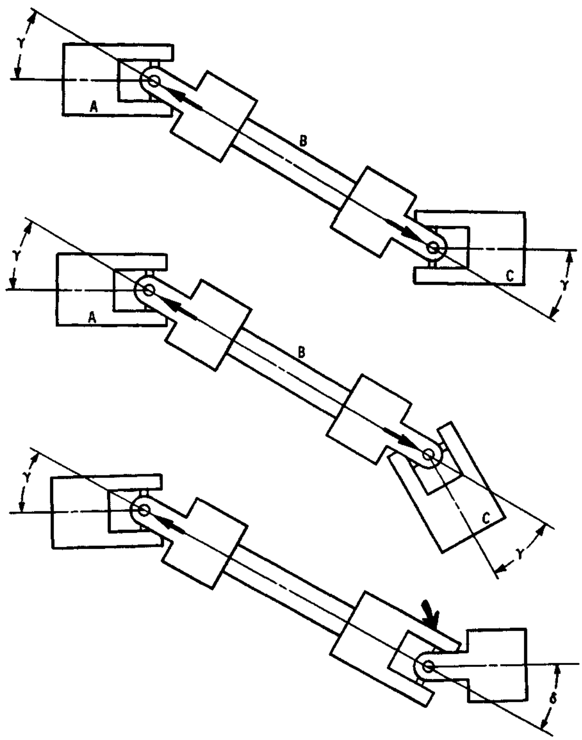


FIGURE 29.37 How two universal joints convert angular misalignment to parallel misalignment. (*a*), (*b*) Correct, yoke ears are aligned and angles are equal. (*c*) Incorrect, yoke ears are not in alignment and angles γ and δ are not equal. A residual angle θ_{eff} results from improper phasing of the two universal joints.

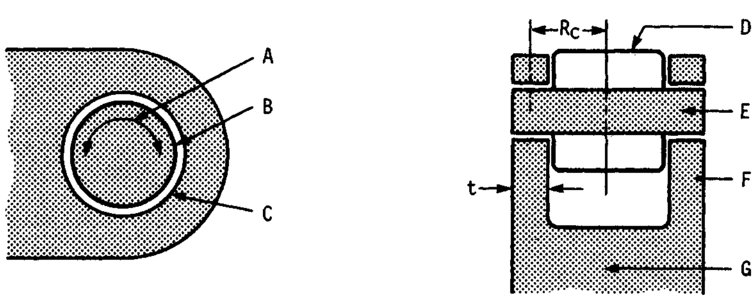


FIGURE 29.38 Plain bearing-type, pin-and-block, universal-joint geometry. A, oscillating pin; B, sliding at yoke interface; C, bore in yoke ear; D, center block; E, bearing pin; F, ear; G, yoke.

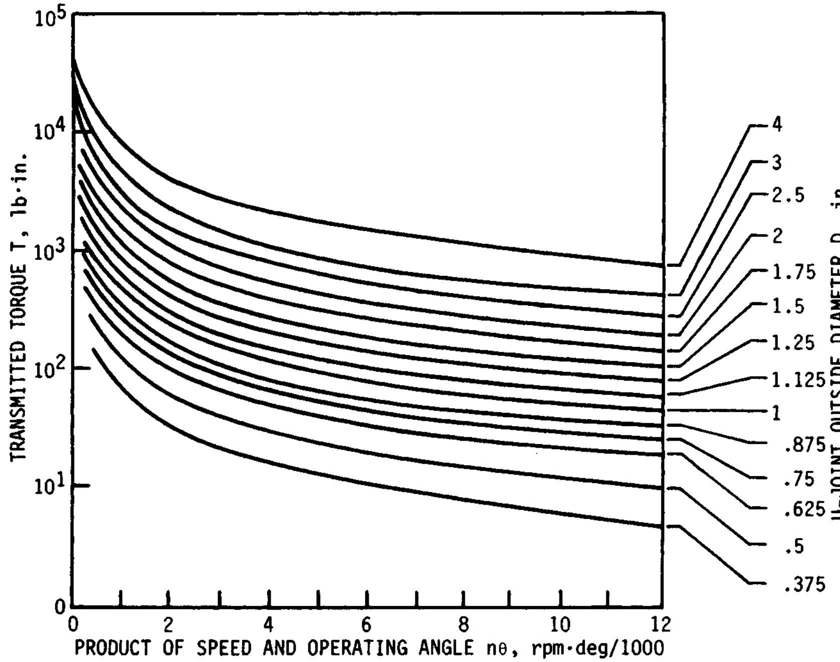


FIGURE 29.39 Torque ratings for pin-and-block universal joint. This chart shows the effect of the speed, operating angle, and OD parameters on the rated torque. These curves are based on a PV value of 1000 psi-in/s.

29.5.2 Needle Bearing

The pin-yoke interface is now changed to a pin and rolling-element bearing. This type of universal joint is also referred to as an automobile type, since it is found in most automobile drivelines. This universal joint also exhibits nonuniform rotational output velocity. A typical needle-bearing universal joint in cross section is shown in Fig. 29.40.

Corresponding to 3000 hours (h) of B_{10} life, the basic torque rating of a needle-bearing universal joint is given by

$$T_R = 4500 K_a K_s d \ell b \tag{29.26}$$

where T_R = basic torque rating, lb·in
K_a = joint angle correction factor
K_s = speed correction factor
d = bearing pin diameter, in
ℓ = length of needle rollers, in
b = bearing spacing, in

The dimensions d, ℓ, and b are shown in Fig. 29.40. The correction factor K_a in Eq. (29.26) is unity when the joint angle γ is 3°. For other angles, use the formula

$$K_a = \frac{1.432}{\gamma^{0.3269}} \tag{29.27}$$

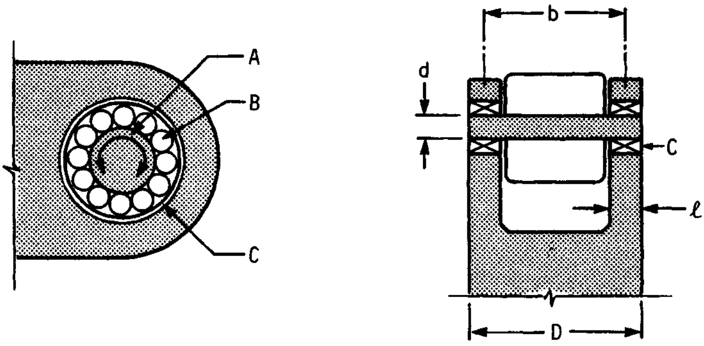


FIGURE 29.40 Geometry of the universal joint with needle bearings. A, oscillating pin; B, rolling motion occurs in the yoke-pin interface; C, needle bearing.

The correction factor K_s in Eq. (29.26) is unity when the speed n is 1000 r/min. For other speeds, use the equation

$$K_s = \frac{9.24}{n^{0.3219}} \tag{29.28}$$

We also define a life factor K_L as the ratio of the actual transmitted torque to the basic torque rating. Thus

$$K_L = \frac{T}{T_R} \tag{29.29}$$

For an expected life L of 3000 h, the life factor is unity. For other lives, use the equation

$$L = \left(\frac{15.5}{K_L}\right)^{2.921} \tag{29.30}$$

where L = expected B_{10} life (see Chap. 27) in hours.

Example 1. A universal joint has an outside diameter $D = 3$ in, an operating speed $n = 4000$ r/min, and a joint angle $\gamma = 5°$. The needle bearings have a needle length of 0.75 in and a bearing pin diameter $d = 0.875$ in. Find the expected B_{10} life if 2400 lb · in of torque is to be transmitted.

Solution. From Fig. 29.40 we find

$$b = D - \ell = 3 - 0.75 = 2.25 \text{ in}$$

Equation (29.27) gives the joint angle correction factor as

$$K_a = \frac{1.432}{\gamma^{0.3269}} = \frac{1.432}{5^{0.3269}} = 0.846$$

The speed correction factor is obtained from Eq. (29.28):

$$K_s = \frac{9.24}{n^{0.3219}} = \frac{9.24}{(4000)^{0.3219}} = 0.640$$

Thus, from Eq. (29.26), we find the basic torque rating to be

$$T_R = 4500K_aK_sd\ell b = 4500(0.846)(0.640)(0.875)(0.75)(2.25)$$
$$= 3598 \text{ lb}\cdot\text{in}$$

Since the actual torque to be transmitted is 2400 lb·in, the life factor is

$$K_L = \frac{T}{T_R} = \frac{2400}{3598} = 0.667$$

So, from Eq. (29.30), the expected life is

$$L = \left(\frac{15.5}{K_L}\right)^{2.921} = \left(\frac{15.5}{0.667}\right)^{2.921} = 9878 \text{ h}$$

29.5.3 Constant-Velocity Universal Joints

Double Cardan Joint. As mentioned previously, two simple Hooke's (Cardan) universal joints can be connected to give relatively constant output velocity (Fig. 29.37). However, if the universal joints are not exactly at the same angle, a small fluctuation in output velocity will occur. Specifically, the torsional equivalent angle $\theta_{\text{eff}} = \sqrt{\theta_1^2 - \theta_2^2}$ for a universal joint when the bearing planes are in the same plane. When the bearing planes are perpendicular, θ_{eff} becomes

$$\theta_{\text{eff}} = \sqrt{\theta_1^2 + \theta_2^2} \tag{29.31}$$

The torsional equivalent angle is used as if only one Cardan joint were in the system.

Tracta. The Tracta joint operates through the sliding action of two internally connected sliding couplings. This type of joint has a fairly high torque and angle capacity for its diameter; however, because of heat buildup from sliding friction, it cannot be used for any extended time at high loads. It can, however, be used satisfactorily at low angles with high loads.

The C-shaped input-output yokes generally have an opening of less than 180° so as to lock into two grooved couplings, which are, in turn, locked together in a tongue-and-groove fashion. See Fig. 29.41.

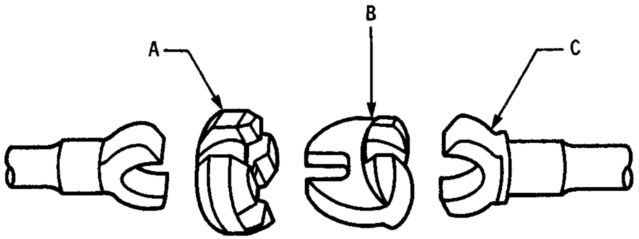


FIGURE 29.41 Exploded view of the Tracta universal joint. A, grooved coupling; B, tongue and groove coupling; C, fork yoke. *(From Ref. [29.6].)*

Rzeppa Universal Joint. This constant-velocity joint uses driving balls and curved axial-groove races; see Fig. 29.42. When the joint is at the same angle θ, the balls on one side of the joint will be farther from the joint centerline than the others. The off-center condition occurs because the groove generation point is displaced to provide camming action necessary for proper ball location and joint operation. Since the outer-race ball grooves act as cam surfaces, frictional locking can occur unless the cam surfaces have a divergent angle of 15° to 17° to prevent this. If the divergent angle were not present, the joint would lock going through zero angle.

Roller and Trunion. If the yoke ears in the simple pin-and-block or needle roller universal joints are elongated to form races, some axial movement is allowed, which decreases greatly the axial force developed by a single rotating universal joint. This joint is not a constant-velocity type. It is also called a *bipot* (see Fig. 29.43).

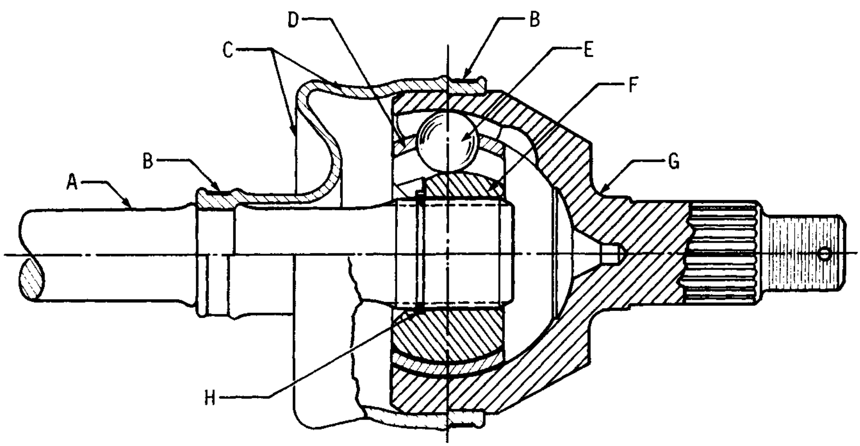


FIGURE 29.42 Rzeppa universal joint in cross section. A, shaft; B, clamp or band; C, boot seal; D, cage; E, ball; F, inner race; G, bell-type outer race; H, retaining ring. *(From Ref. [29.6].)*

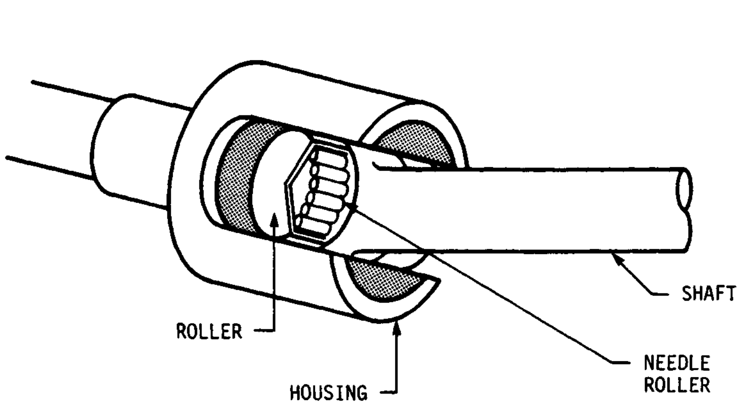


FIGURE 29.43 Roller trunion universal joint.

Tripot. If three equally spaced rollers are used, the joint will transmit constant velocity because of the cancellation of torque couples within the joint. The center of this joint may be free to float (end motion type), in which case the axial force generated by the joint is low and the joint is self-supporting. In the fixed-center type, the joint center is fixed, which causes the joint to orbit about its center; this limits it to low-speed use, because of inertial effects, and places where this orbital motion can be tolerated (such as in conjunction with self-aligning bearings or other universal joints). See Ref. [29.6].

29.5.4 Rotating-Link Coupling

This coupling consists of three disks connected by two sets of three links, as shown in Fig. 29.44. Typically, one set of links is attached to shafts on the driver and center disks, while the other set of links connects the center to the lower disk. All links are of equal length, and all the shafts are equally spaced on the same bolt circle on each hub. The maximum parallel displacement of the input and output shafts is determined by and equal to the link length S. With this arrangement, unless the input and output shafts were misaligned at least a small amount S_{min}, the center disk would be free to swing about the center of the shafts. This would cause an unbalance in the coupling and would result in high vibration and limited life. This effect gives the misalignment range of this coupling of S_{min} to S.

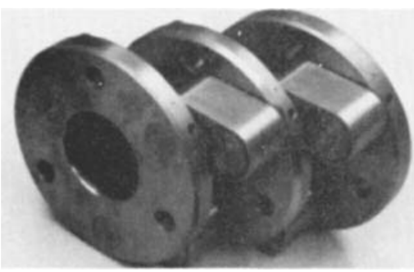


FIGURE 29.44 Rotating-link coupling. *(Schmidt Couplings, Inc.)*

The construction of this coupling causes the links to move parallel to one another and results in constant angular velocity. In addition, because three links are equally spaced about each hub and are of equal size and mass, the sum of all link forces is zero, resulting in smooth operation without imposing side loads.

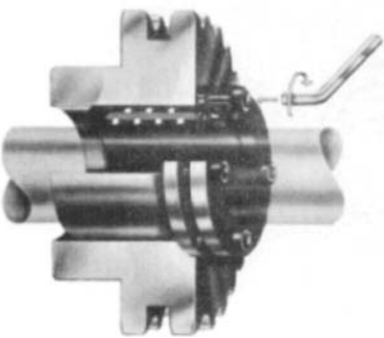


FIGURE 29.45 Pressure bushing. *(FFV Industrial Products.)*

29.6 METHODS OF ATTACHMENT

There are several methods of attaching the coupling hubs to the shafts on equipment. Among these are split-taper bushings, keys and splines, shrink-fit and spline profile distortion, and pressure bushings. The first two items are covered in Chap. 22; shrink fits are discussed in Chap. 19.

Spline profile distortion causes the entire bore area of the coupling hub to deform around the mating shaft. This

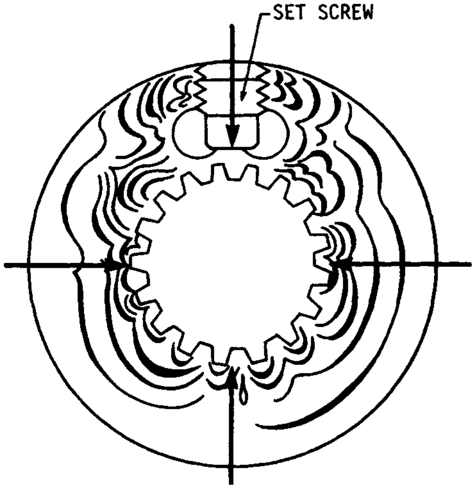


FIGURE 29.46 Drawing of a model of Centaloc spline profile distortion obtained by photoelastic methods showing lines of constant stress. *(Lovejoy, Inc.)*

method of attachment allows a much greater clamping force to be exerted than with a split-taper hub with internal spline. This clamping about the entire spline is important and eliminates fretting corrosion (caused by loose-fitting spline shafts and mating hubs) and point contact obtained with split bushing types, as shown in Fig. 29.46.

Pressure bushings (Fig. 29.45) are similar to the hydraulic coupling except that both the inner and outer surfaces are allowed to deform, compressing onto the shaft and expanding into the coupling hub. The pressure source for this type is usually applied through axial compression of the bladder with a clamping ring.

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