SECTION 6.3
POWER TRANSMISSION DEVICES

6.3.1 PUMP COUPLINGS AND INTERMEDIATE SHAFTING

FRED K. LANDON
DONALD B. CUTLER

COUPLING TYPES USED IN PUMP DRIVE SYSTEMS

A coupling is used wherever there is a need to connect a prime mover to a piece of driven machinery. The principal purpose of a coupling is to transmit rotary motion and torque from one piece of equipment to another. Couplings may perform other secondary functions, such as accommodating misalignment between shafts, compensating for axial shaft movement, and helping to isolate vibration, heat, and electrical eddy currents from one shaft to another.

Rigid Couplings Rigid couplings are used to connect machines where it is desired to maintain shafts in precise alignment. They are also used where the rotor of one machine is used to support and position the other rotor in a drive train. Because a rigid coupling cannot accommodate misalignment between shafts, precise alignment of machinery is necessary when one is used.

TYPES There are two commonly used types of rigid couplings. One type consists of two flanged rigid members, each mounted on one of the connected shafts (Figure 1). The flanges are provided with a number of bolt holes for the purpose of connecting the two half-couplings. Through proper design and installation of the coupling, it is possible to transmit the torque load entirely through friction from one flange to the other, which assures that the flange bolts do not experience a shearing stress. This type of arrangement is especially desirable for driving systems where torque oscillations occur, as it avoids subjecting the flange bolts to a shearing stress.

A second type of rigid coupling, known as the *split rigid,* is split along its horizontal centerline (Figure 2). The two halves are clamped together by a series of bolts arranged axially along the coupling. The rigid coupling and machine shafts may be equipped with conventional keyways, which are in turn fitted with keys to transmit the torque load, or in

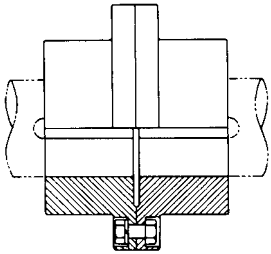

FIGURE 1 Flanged rigid coupling (Kop-Flex)

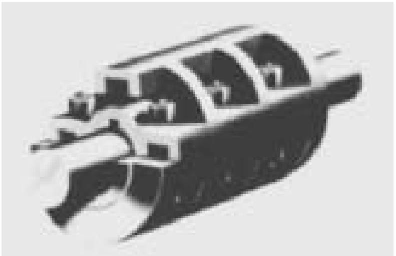

FIGURE 2 Split rigid coupling (Dodge Manufacturing Division, Reliance Electric)

certain cases the frictional clamping force may be sufficient to permit transmitting the torque by means of friction between shaft and rigid coupling. This type of coupling is commonly used to connect sections of line shafting in a drive train.

A variation to the flanged rigid coupling is known as the adjustable rigid coupling (Figure 3). This coupling is designed along the lines of conventional rigid couplings, except that a threaded adjusting ring is placed between the two flanges. This ring engages a threaded extension on one of the shaft ends. By means of this ring, it is possible to position the pump shaft axially with respect to the driver.

APPLICATIONS A common application for rigid couplings in the pump industry is in vertical drives, where the prime mover (generally an electric motor) is positioned above the pump. In such cases, both machines can employ a common thrust bearing, which is generally located in the motor. The coupling flange bolts must be capable of transmitting any down thrust from the pump to the motor. In applications where the thrust from the pump is toward the motor, it is common practice to provide shoulders on the shafts to transmit the axial force.

Many pump drive systems require a rigid coupling that is capable of providing axial adjustment to compensate for wear in the pump impeller or impellers. The adjustable rigid coupling is used for this purpose. The threaded adjusting ring attached to a mating threaded extension of the pump shaft permits vertical positioning of the impeller or impellers. The hub, which is mounted on the pump shaft, is equipped with a clearance lit and feathered key that permit the hub to slide with respect to the shaft. The load capacity of a coupling of this type is generally limited by the pressure on the pump shaft key because there is no possibility of load being transmitted by interference fit.

A few words of caution should be noted about the use of rigid couplings. First, precise alignment of machine bearings is absolute necessary because there is no flexibility in the coupling to accommodate misalignment between shafts. Secondly, accuracy of manufacture is extremely important. The coupling surfaces that interface between driving and driven shafts must be manufactured with high degrees of concentricity and squareness, to avoid the transmittal of eccentric motion from one machine to the other.

Flexible Couplings Flexible couplings accomplish the primary purpose of any coupling; that is, to transmit a driving torque between prime mover and driven machine. In addition, they perform a second important function: they accommodate unavoidable misalignment between shafts. A proliferation of designs exists for flexible couplings, which may be classified into two types: mechanically flexible and materially flexible.

MECHANICALLY FLEXIBLE COUPLINGS Mechanically flexible couplings compensate for misalignment between two connected shafts by means of clearances incorporated in the design of the coupling. The most commonly used type of mechanically flexible coupling is the gear, or dental, coupling (Figure 4). This coupling essentially consists of two pair of clearance fit splines. In the most common configuration, the two machine shafts are

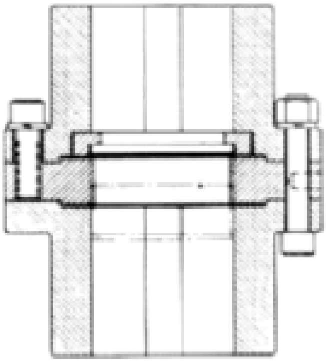

FIGURE 3 Adjustable flanged rigid coupling (Kop-Flex)

FIGURE 4 Gear-type mechanically flexible coupling (Kop-Flex)

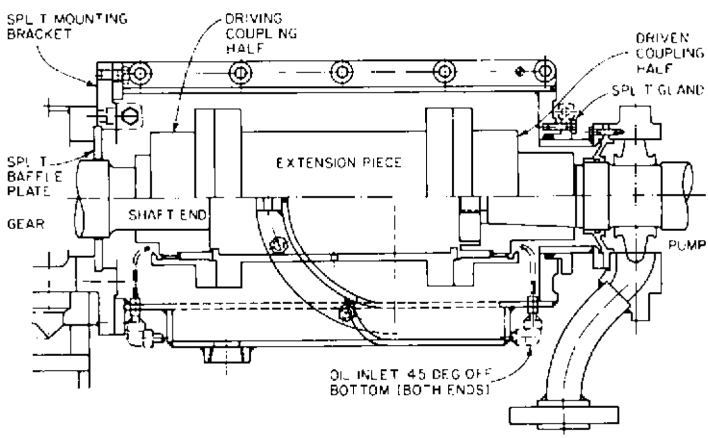

FIGURE 5 Continuously lubricated coupling (Kop-Flex)

equipped with hub members having external splines cut integrally on the hubs. The two hubs are connected by a sleeve member having mating internal gear teeth. Backlash is intentionally built into the spline connection, and it is this backlash that compensates for shaft misalignment. Sliding motion occurs in a coupling of this type, and so a supply of clean lubricant (grease or oil, depending on the design) is necessary to prevent wear of the rubbing surfaces.

If operation cannot be interrupted to lubricate the couplings, constantly lubricated couplings are used, as shown in Figure 5. These consist of an oiltight enclosure bolted at one end to the stationary portion of either the driving or driven piece of equipment. The other end of the enclosure has a slip fit inside a cover that is bolted to the other piece of equipment. Some form of packing is used to prevent loss of lubricant at the slip joint. Oil under pressure is brought through the enclosure and impinges upon the meshing gear teeth of the coupling, the excess being collected at the bottom of the enclosure and returned to the oil reservoir.

A second type of mechanically flexible coupling that sees wide usage, especially in low-cost drive systems, is known as the roller-chain flexible coupling (Figure 6). This coupling employs two sprocket-like members mounted one on each of the two machine shafts and connected by an annulus of roller chain. The clearance between sprocket and roller and, in some cases, the crowning of the rollers provide mechanical flexibility for misalignment. This type of coupling is generally limited to low-speed machinery.

MATERIAL-FLEXIBLE COUPLINGS These couplings rely on flexing of the coupling element to compensate for shaft misalignment. The flexing element may be of any suitable material (metal, elastomer, or plastic) that has sufficient resistance to fatigue failure to provide acceptable life. Some materials, such as steel, have a finite fatigue limit. A coupling made of such material must be operated under conditions of load and misalignment that assure that the stress developed in the coupling element is within that limit. Other materials, such as elastomers, generally do not have a well-defined fatigue limit. In these cases, however, heat developed in the material when the coupling flexes can cause failure if excessive.

One type of material-flexible coupling is the metal-disk coupling (Figure 7). This coupling consists of two sets of thin sheet-metal disks bolted to the driving and driven hub members. Each set of disks is made up of a number of thin laminations that are individually flexible and compensate for shaft misalignment by means of this flexibility. These disks may be stacked together as required to obtain the desired torque transmission capability. This type of coupling requires no lubrication; however, alignment of the equipment must be maintained within acceptable limits so as not to exceed the fatigue limit of the material.

Another example of an all-metal material-flexible coupling is the flexible diaphragm coupling, shown in Figure 8. This coupling is similar in function to the metal-disk coupling in that the disk flexes to accommodate misalignment. However, the diaphragm type consists of a single element with a hyperbolic contour that is designed to produce uniform stress in the member from inner to outer diameter. By more efficiently utilizing the material, the weight is reduced correspondingly, thus making this coupling suitable for high-speed applications.

Material-flexible couplings employing elastomer materials are numerous and their designs are varied. By definition, an elastomer is a material that has a high degree of elasticity and resiliency and will return to its original shape after undergoing large-amplitude deformations. One example of an elastomer coupling is the pin-and-bushing coupling (Figure 9). This design comprises two flanged hub members, one mounted on each machine shaft. The flange of one hub is fitted with pins that extend axially toward the adjacent shaft. The other flange is equipped with rubber bushings, which generally have a metal

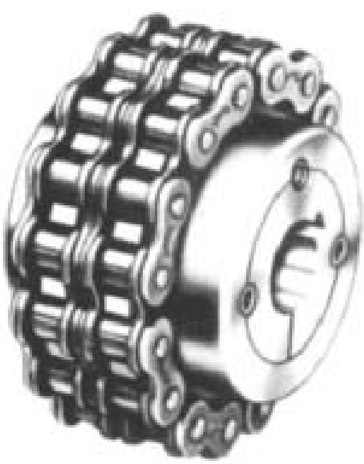

FIGURE 6 Roller-chain mechanically flexible coupling (Dodge Division, Reliance Electric)

FIGURE 7 Spacer metal disk coupling (Thomas Coupling Div./Rexnord)

FIGURE 8 Diaphragm material-flexible coupling (Fluid Power Division, Bendix)

FIGURE 9 Pin-and-bushing elastomer coupling (Ajax Flexible Coupling)

FIGURE 10 S1eeve-type elastomer coupling (T B. Woods)

sleeve at the center. The pins fit into these sleeves and provide transmission of torque through the bushings. Because the bushings are made of flexible material, they can accept slight angularity or offset conditions between the two flanges.

A second group of elastomer couplings employs a sleeve-like element that is connected to a hub member on each shaft and transmits torque through shearing of the flexible element. The flexible element may be attached to the machine hubs by a number of different means: it may be chemically bonded, it may be mechanically connected by means of loose-fitting splines (Figure 10), or it may be clamped to the hubs and held in place by friction (Figure 11). Misalignment between shafts is accommodated through flexing of the elastomer sleeve.

A third group of couplings utilizes an elastomer member that is loaded in compression to transmit load from one shaft to the other. The elastomer material is placed loosely into cavities formed by members rigidly mounted onto the two shafts (Figure 12). Again, the elastomer material deflects to compensate for shaft misalignment, and is usually used when torsional damping is required.

Another type of elastomer coupling that is commonly used on low-power drive systems is the rubber jaw coupling (Figure 13). The heart of this coupling is a "spider" member, having a plurality (usually three) of segments extending radially from a central section. The

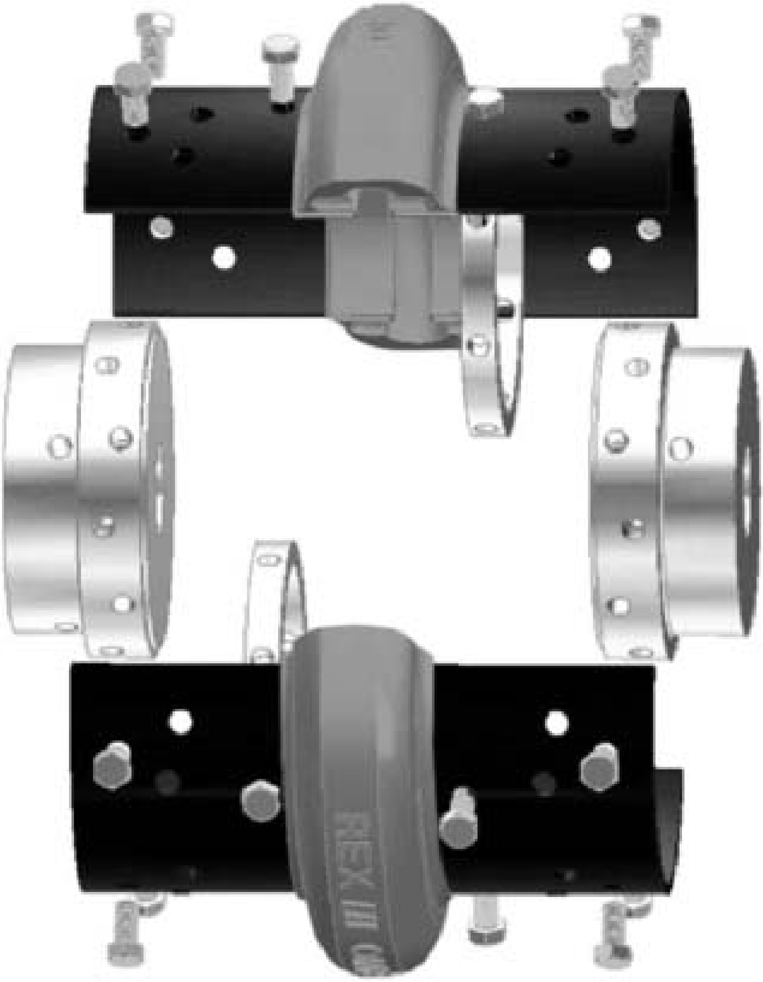

FIGURE 11 Spacer elastomer coupling (Omega Coupling Div./Rexnord)

hubs, which are mounted on driving and driven shafts, each have a set of jaws corresponding to the number of spiders on the flexible element. The spider fits between the two sets of jaws and provides a flexible "cushion" between them. This cushion transmits the torque load as well as compensating for misalignment.

SPRING-GRID COUPLING There is a type of commercially available flexible coupling that combines the characteristics of mechanically flexible and material-flexible couplings. This is the spring-grid coupling (Figure 14). This design has two hubs, one mounted on each machine shaft. Each hub has a raised portion on which toothlike slots are cut. A spring steel grid member is fitted, or woven, between the slots on the two hubs. The grid element can slide in the slots to accommodate shaft misalignment and flexes like a leaf spring to transmit torque from one machine to the other. Unlike most material-flexible couplings, this design requires periodic lubrication to prevent excessive wear of the grid member.

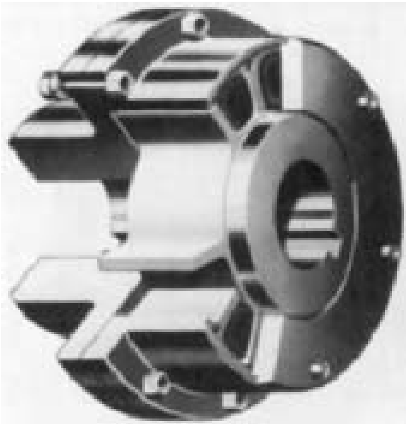

FIGURE 12 Compression-loaded, loosely fitted elastomer coupling (Kop-Flex)

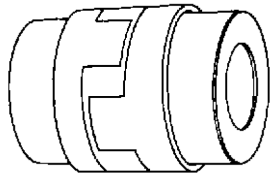

FIGURE 13 Rubber jaw coupling (Lovejoy)

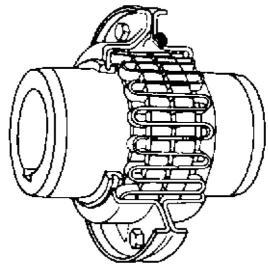

FIGURE 14 Spring-grid coupling (Falk Division, Sundstrand Corp.)

APPLICATIONS The type of flexible coupling most suitable for a particular application depends upon a number of factors, including power, speed of rotation, shaft separation, amount of misalignment, cost, and reliability. In the design of a system, it is the goal of the designer to use the least expensive coupling that will do the job. In low-cost systems, cost alone may be the most important criterion, and the least expensive coupling that transmits the rated power and accepts some small degree of misalignment is generally the choice, albeit at some sacrifice of reliability and durability. On the other hand, high-power, high-speed machinery generally represents a critical piece of equipment for a power station, sewage plant, or other vital process, and in these cases a coupling should be selected that will not compromise the overall reliability of the system.

Low-power pumps (up to about 200 hp, 150 kW) driven by electric motors can usually be coupled successfully by any of the couplings described here. Selection procedures vary from manufacturer to manufacturer, but generally the following data are required: power rating, speed, anticipated misalignment, and type of pump (reciprocating, vane, centrifugal, and so on).

Pumps of the same power range are very often driven by reciprocating engines (diesel, gasoline, natural gas). This is quite common in remote areas, such as at pipeline pumping stations, where a source of electric power is not available. Because this type of prime mover produces a pulsating type of power, it is often necessary to perform a torsional vibration analysis of the drive system to ensure that the normal operating speed is well removed from a speed that may produce a torsional resonant vibration. Such an analysis requires that the torsional stiffness of the coupling be known. It is quite often possible to tune the drive system to avoid operating at a resonant condition by selecting the proper coupling stiffness. The selection data required for a system of this type are the same as listed above. Most coupling manufacturers will, however, assign a higher service factor to an application involving a reciprocating prime mover, to compensate for fatigue effects due to torque fluctuations. In addition, the remote location of many engine-driven pumps indicates a special need to ensure a high degree of reliability of the system.

Another commonly employed prime mover is a steam turbine. These machines, which range from about 100 hp (75 kW) to well over 50,000 hp (37,000 kW) for pump drives, operate very efficiently and economically, providing there is a source of steam available at the installation. Any flexible coupling employed on a machine driven by a steam turbine must be capable of accepting the thermal gradient at the turbine shaft and must also accommodate the axial growths of the turbine shaft as it warms up to operating speed.

Steam turbines are generally high-speed machines (4,000 to as high as 10,000 to 15,000 rpm in some cases) and as such require a relatively high degree of system balance to avoid critical vibration. Elastomer couplings have occasionally been applied successfully to steam turbine drives, but because of the high speeds involved, all-metal couplings are usually employed. The metal coupling most commonly used on this type of drive is the gear (mechanically flexible) design. High-speed machinery requires that the weight of rotating components be minimized to decrease shaft deflections and hence increase the lateral critical speed of the system. The gear coupling is an efficient design for transmitting large amounts of power at high speeds and with minimum weight. However, disk/diaphragm coupling designs that are light, flexible, and do not require lubrication are becoming more popular, especially for higher speed applications.

Where coupling weight and torsional stiffness are critical values to the overall system, special designs may be created that provide the specific values required for satisfactory system operation.

BALANCE For higher speed, higher power applications, coupling balance (or residual unbalance) is an important factor to consider. Elastomeric couplings may have a considerable amount of residual unbalance because of their construction, and they don't lend themselves to balancing. The amount of residual unbalance in metal couplings may be controlled largely by the level of precision to which they are manufactured. When pumps operate at speeds exceeding four-pole motor speeds and low equipment vibration levels are critical to service life, elastomeric couplings may not be a good choice. Such requirements are important to certain pump types, like those required for API Standard 610: "Centrifugal Pumps for Refinery, Heavy Duty Chemical, and Gas Industry Services." For more information on coupling manufacturing and balance classes and requirements, the reader is referred to the API and AGMA standards listed at the end of this section.

LIMITED END FLOAT Many horizontal motor-driven pump systems utilize motors that are equipped with journal, or sleeve-type, bearings. These bearings are intended only to absorb the transient thrust created by the motor rotor during acceleration and deceleration. The coupling for this type of drive should be equipped with suitable provisions for limiting the axial float of the motor rotor to some fraction of its total float. This may be done by positioning the motor in the center of its axial travel and then employing a coupling having a total float that is less than the float of the motor. Any motor thrust is taken by the pump bearing. Gear coupling total float can be limited by inserting a button between shaft ends, as shown in Figure 15. This type of coupling prevents the motor rotor from ever contacting the thrust shoulders on the shaft bearings. Certain types of elastomer and disk couplings having inherent float-restricting characteristics provide centering without any additional modifications (Figures 16 and 17).

VERTICAL OPERATIONS As previously noted, rigid couplings are commonly used on vertical-drive systems where the system characteristics warrant such a coupling. However, many vertical-drive systems require a flexible coupling to accommodate shaft misalignment. It is generally possible to use a nonlubricated coupling, such as one of the many elastomer designs, in a vertical position without modification, provided the shafts are supported in their own bearings and the coupling does not have to transmit a thrust force. Lubricated designs, such as the gear and spring-grid types, usually require some modification to make certain that lubricant is retained in both halves of the coupling.

PUMP DRIVE SHAFT SYSTEMS

Pump drive system arrangements may be classified in one of two categories: those that are close-coupled, having a shaft separation of a fraction of an inch (not to be confused with

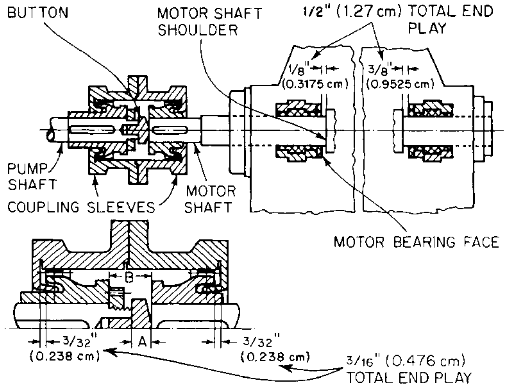

FIGURE 15 Motor-driven coupling end float limited by insertion of button between shaft end (Flowserve Corporation)

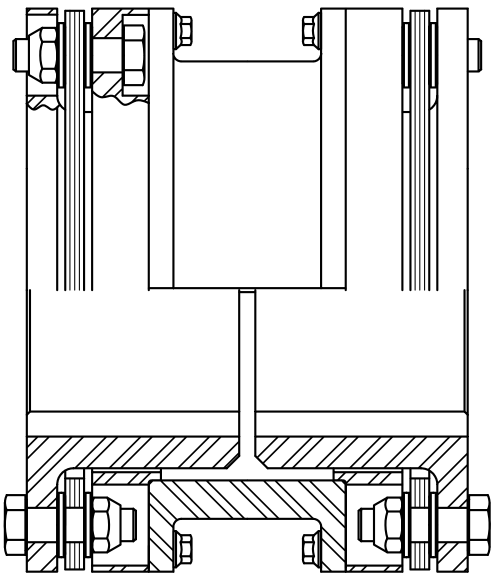

FIGURE 16 Metal disk coupling with flanged, tubular spacer (Thomas Coupling Div./Rexnord)

integral motor pumps not having a flexible coupling) and those that, for one or more reasons, have the prime mover located a substantial distance from the pump. These latter types of systems require a modification to the basic flexible coupling designs previously described.

Spacers One means of accommodating shaft separation in excess of the normal amount provided in a standard coupling is to employ a flanged tubular spacer between the two coupling halves (Figure 18). These components are lightweight and are commonly used with end-suction pumps, where it is possible to remove the pump impeller while the pump

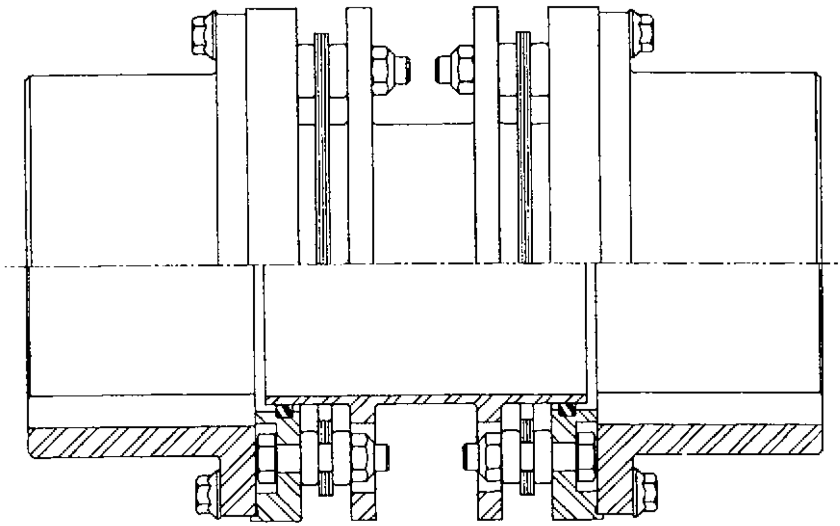

FIGURE 17 Vertical double-engagement gear coupling (Koppers)

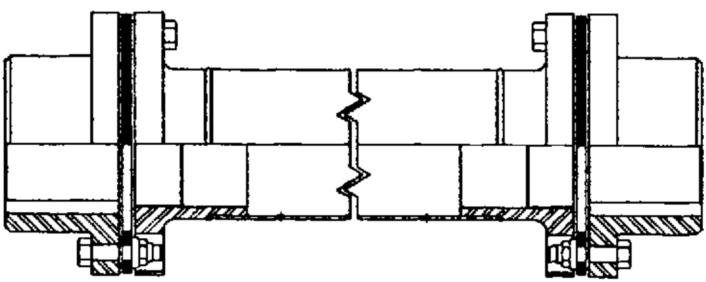

FIGURE 18 Tubular spacer coupling (Koppers).

housing and piping, as well as the prime mover, remain in place (Figure 19). For high-speed drive systems, a lightweight spacer is usually the only practical means of achieving the goal. This spacer may be used to tune the system torsionally (as previously discussed) by varying the body diameter and wall thickness. Spacers can be manufactured in lengths up to several feet, but the cost generally restricts its usage to shorter lengths, except where no other means is suitable.

Floating Shafts Floating shaft couplings have a purpose similar to that of spacer couplings; namely, to connect two widely separated shafts. The basic difference is in the construction of the component. Whereas spacers are normally made with the body having an integrally formed flange to connect the two coupling halves, a floating shaft usually is made by attaching a flange to a piece of solid or tubular shafting by means of mechanical keys or by welding (Figure 20). This type of construction is generally less expensive than a one-piece spacer, especially when very large shaft separations must be spanned. Graphite composite center members are also being used. This material has the advantage of being much lighter and stiffer than standard metal components, allowing longer spans between bearing supports without adding mass.

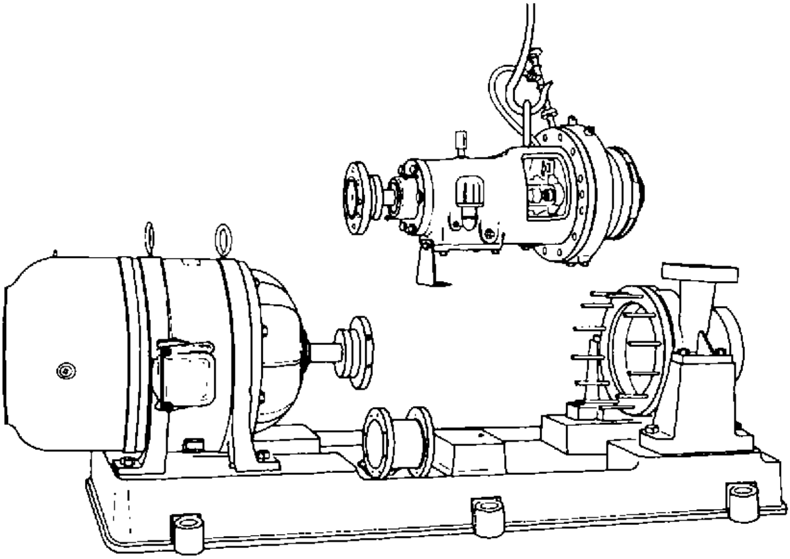

FIGURE 19 Spacer coupling enables end-suction pump to be dismantled without moving piping, pump casing, or driver (Worthington Pump).

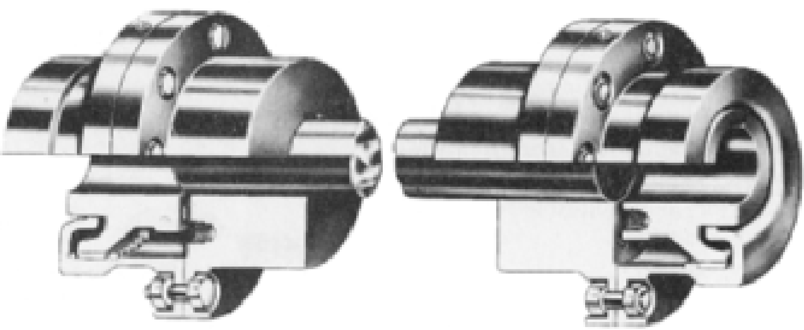

FIGURE 20 Section of floating shaft and couplings (Kop-Flex)

Floating shaft arrangements are widely used on horizontal pump applications of all types and are especially common on vertical pump applications, such as in water pumping and sewage treatment stations. In these latter applications, the pump may be submerged in a pit usually 30 to 50 ft (9 to 15 m) below ground level, whereas the motor (to prevent damage during flooding) is mounted at ground level. Settings as deep as 100 ft (30 m) have been constructed. Such systems require long floating shafts, which are generally made in sections and supported by line bearings at intermediate supports or floors.

Rigid Shafts Vertical centrifugal pumps can be designed to contain their own thrust and line bearings. These pumps employ a flexible coupling at the pump shaft. When vertical intermediate shafting is used, the shaft need be designed only to transmit torque. Some pumps are designed to contain only a single line bearing. These pumps require a rigid coupling at the pump shaft so axial thrust can be carried by the thrust bearing in the driver or gear located above. The constructions of a single oil-lubricated line-bearing pump is shown in Figure 109 of Subsection 2.2.1. In order to carry thrust, the drive shaft, if made in sections, must use rigid intermediate couplings and the coupling at the driver or gear must also be rigid. Intermediate bearings used with rigid shafting must be designed to provide only lateral support, thereby assuring that all the axial thrust is carried by the driver or gear. An intermediate oil-lubricated sleeve-type guide bearing is shown in Figure 108 of Subsection 2.2.1. Pump and intermediate shaft sleeve guide bearings can be either grease- or oil-lubricated, and they can also be antifriction if preferred.

A rigid intermediate shaft connected to a pump provided with a rigid coupling must be designed to carry its own weight, the axial thrust from the pump, and a bending load. Volute-type centrifugal pumps produce a radial reaction on the pump shaft that in turn is transmitted to the vertical shafting through the rigid pump coupling. Intermediate bearings then act as line bearings. The bearing supports must also be designed to resist this bending force. The pump shaft and bearing, intermediate shafts and line bearings, and the driver or gear shaft and bearings must be treated as a single shaft supported at each bearing when analyzing the critical speed of the entire rotor system. If the intermediate line bearing supports are assumed to be nodes in the critical speed calculation, these supports must be absolutely rigid. The supplier of the shafting should, however, confirm what flexibility will be allowed at these supports and give the design forces. The support for the guide bearings must also be designed not to have a natural frequency of vibration within the operating speed range of the pump.

Flexible Drive Shafts Universal joints with tubular shafting (Figure 21) can be substituted for flexible couplings whenever it is necessary to (*a*) eliminate the need for critical alignment, (*b*) provide wider latitude in placement of pump and driver, and (*c*) permit large amounts of relative motion between pump and driver. This type of shafting can be used horizontally as well as vertically to provide a short spacer arrangement or a large separation between pump and driver, such as that required for deep settings (Figure 22).

Flanges are furnished to fit pump and driver shafts to the universal joints, which are splined to allow movement of the shaft. An intermediate steady bearing is required at each joint, which must also support the weight of a section of shafting (Figure 23). Because of the spline, pump thrust cannot be taken by the driver, and it is necessary to use a combination pump thrust and line bearing. Because of the universal joint, the intermediate bearing or bearings take no radial load from the pumps and therefore act only as a steady bearing or bearings for the shaft.

The shaft must be selected to transmit the required torque and be of a length and diameter that will have a critical speed well removed from the operating speed range of the pump. The support for the guide bearing must not have a natural frequency of vibration within the speed range. Shafts of this type are often pre-engineered and stocked. Selection charts are available from the manufacturer to make a proper size and length selection. Standard tubular, flexible drive shafts have limited torque-carrying capacity.

Design Criteria

TORSIONAL STRESS The torsional stress in the shafting may be calculated by the following equations:

$$S_s = \frac{16T}{\pi D_o^3} \qquad \text{for solid shafting}$$

$$S_s = \frac{16T}{\pi D_o^3\left(1 - \dfrac{D^4}{D_o^4}\right)} \qquad \text{for tubular shafting}$$

where S_S = torsional shear stress, lb/in^2 (N/m^2)

T = transmitted torque, in · lb (N · m)

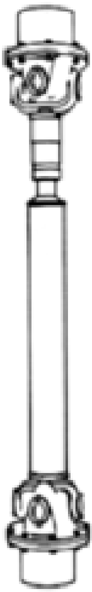

FIGURE 21 Tubular intermediate shafting and universal units (H. S. Watson)

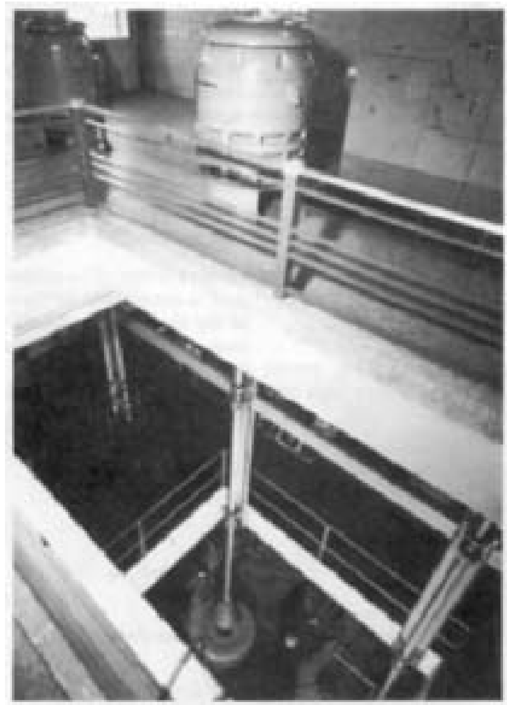

FIGURE 22 Sections of flexible shafts and intermediate guide bearings used to transmit torque from motor located above flood elevation to pump located below ground (Flowserve Corporation)

D_o = shaft outer diameter, in (m)
D = shaft inner diameter (tubular shaft only), in (m)

The allowable shear stress depends upon the material being used and whether it is subjected to other loads, such as bending or compression. The design safety factor on the shafting should be equal to or greater than those of the other components in the drive train.

CRITICAL SPEED The critical speed of a drive shaft is determined by the deflection, or "sag," of the shaft in a horizontal position under its own weight. The less the sag, the higher the critical speed. In practical terms, a long, slender shaft will have a low critical speed and a short, large-diameter shaft will have a very high critical speed. The deflection of a simply supported shaft is calculated as follows:

$$y = \frac{5wL^4}{384EI} = \text{ shaft deflection, in (m)}$$

noting that

$$I = \frac{\pi D_o^4}{64} \qquad \text{for solid shafting}$$

$$I = \frac{\pi(D_o^4 - D^4)}{64} \text{ for tubular shafting}$$

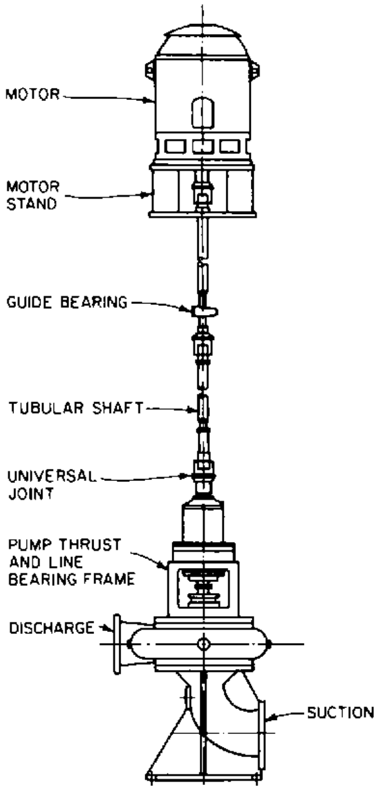

FIGURE 23 Vertical pump with flexible shafting (Flowserve Corporation)

where w = weight (force) of shaft per unit length, lb/in (N/m)
L = length between bearing supports, in (m)
E = Young's modulus, lb/in^2 (N/m^2)
I = moment of inertia, in^4 (m^4)

Knowing the natural deflection of the shaft, it is possible to calculate the first critical speed from the equation:

$$N_{\text{crit}} = 187.7\sqrt{\frac{1}{y \text{ (in inches)}}} = 946\sqrt{\frac{1}{y \text{ (in millimeters)}}}$$

which expresses critical speed directly in revolutions per minute of the rotating shaft.

In practice, the critical speed should be placed well away from the operating speed of the shaft. Actual operating conditions, such as imbalance and clearance in bearings and couplings, tend to reduce the critical speed from its theoretical value. The designer has several options from which to choose to attain the desired critical speed: (1) vary the shaft diameter, (2) use tubular shafting to increase stiffness and reduce deflection, and (3) vary the bearing span.

Varying the diameter of the shaft has a dramatic effect on shaft deflection because the deflection decreases inversely with the cube of the diameter. The weight of the shaft will increase proportionately to the square of the diameter and may consequently impose excessive loads on hearings. When this is the case, tubular shafting may be substituted for solid shafting. The ratio *I/w* is much higher for a tubular shaft than for a solid shaft of the same diameter, so the shaft deflection is again reduced and critical speed increased.

The bearing span may be varied when the size and weight of a single length is broken into sections, with a single engagement coupling used to connect each section and acting as a hinge joint. A self-aligning bearing is placed on each section immediately adjacent to the coupling. Each section may then be treated as an individual shaft, which will result in a substantial reduction in shaft size. The rigidity of the support at each hearing must be taken into consideration when calculating the critical speed of the shaft.

Dynamic Balance Drive systems that operate at high speeds and connect two widely separated shafts must be given special consideration with regard to dynamic imbalance. Care must be taken to ensure that the long, slender shaft used in such systems is not bent, either in manufacture or in installation. These systems quite often require shafting that has been dynamically balanced after manufacture to compensate for these normal errors.

REFERENCES AND FURTHER READING

1. American Gear Manufacturers Association, ANSI/AGMA 9000. “Flexible Couplings—Potential Unbalance Classification.” 1500 King Street, Suite 201, Alexandria, VA, 22314.
2. American Petroleum Institute Standard 610. “Centrifugal Pumps for Refinery, Heavy Duty Chemical, and Gas Industry Services.” 8th ed., 1220 L Street, Washington, DC 20005, 1995.
3. American Petroleum Institute Standard 671. “Special Purpose Couplings for Petroleum, Chemical, and Gas Industry Services.” 3rd ed., 1220 L Street, Washington, DC 20005, October, 1998.