14

COUPLINGS, TORQUE TRANSMISSION, AND TORQUE SENSING

14.1 COUPLING OVERVIEW*

High-reliability and low-maintenance requirements are almost always among the key demands of process machinery users. This explains why coupling selection deserves considerable attention. Major requirements for high-performance couplings are high-torque and high-speed capacity, low weight and small envelope size, low overhung moment, and low residual unbalance. The coupling must be capable of transmitting the system design torque at maximum continuous speed for extended periods. It must be able to handle speed and load transients at defined misalignment conditions with minimum reactions on the drive system.

There are several major factors that in combination, determine the continuous duty rating of a coupling:

- High speeds may limit the coupling diameter. This in turn sets the tooth pitch diameter and tooth loading of gear couplings (Fig. 14.1) or bolt pitch diameter of nonlubricated couplings (Figs. 14.2 through 14.4).
- The torque that can be transmitted by gear couplings of a given pitch diameter and tooth length is a function of allowable contact pressure and the relative sliding velocity between hub and sleeve teeth.
- The torque that can be transmitted by metal disk or diaphragm-type high-performance couplings is a function of allowable stresses in the flexing members.

* Based on application summaries provided by Michael Calistrat & Associates, Missouri City, Tex.; product information courtesy of Lucas Aerospace Company, Utica, N.Y., and the various contributors listed as sources of illustrations.

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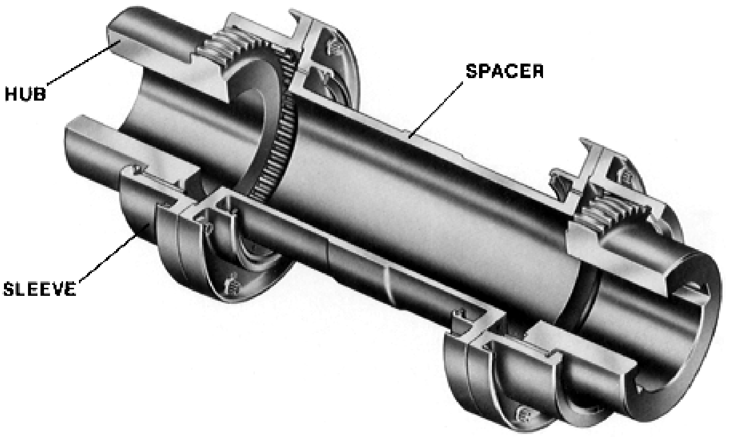

FIGURE 14.1 Gear coupling requires lubrication. *(Zurn Industries, Erie, Pa.)*

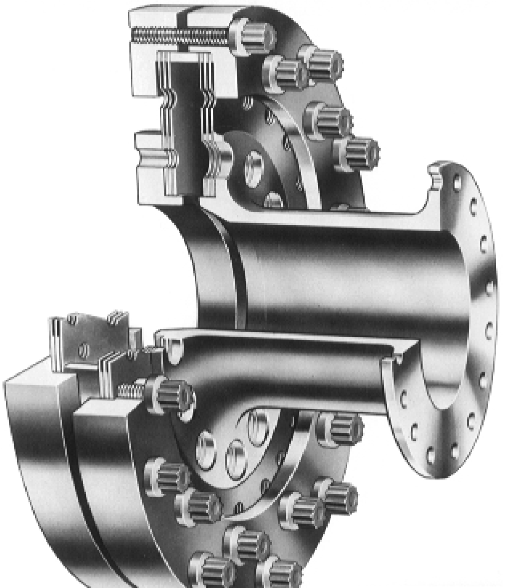

FIGURE 14.2 Nonlubricated Flexxor coupling. *(Coupling Corporation of America, Jacobus, Pa.)*

- Running misalignment determines the relative sliding velocity between the gear teeth for a given pitch diameter and speed of rotation. Similarly, running misalignment affects the stress levels in flexing members of nonlubricated couplings.
- Tooth hardness determines the allowable contact pressure. Heat-treated alloy steel is suitable for many applications, but increased surface endurance may be obtained by suitable hardening procedures, regardless of coupling type.

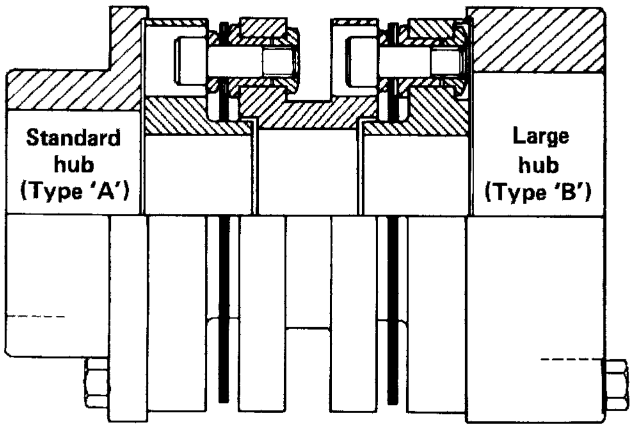

FIGURE 14.3 Disk pack coupling. *(Flexibox, Inc., Houston, Tex.)*

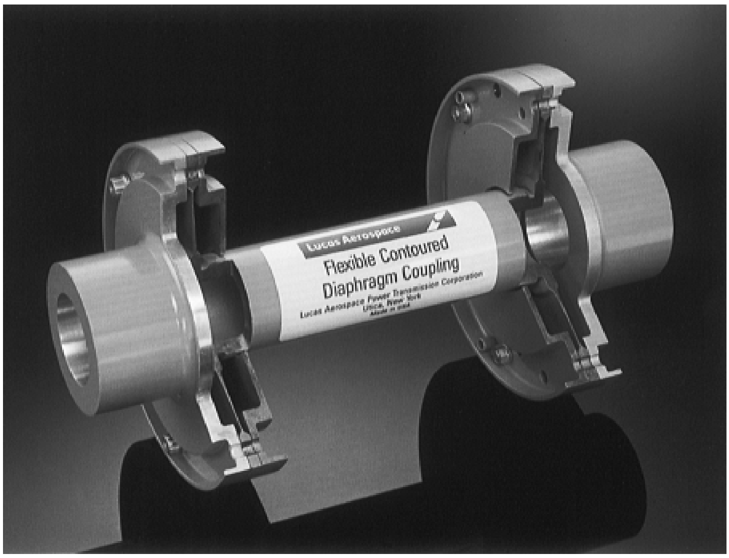

FIGURE 14.4 Flexible contoured diaphragm coupling. *(Lucas Aerospace Company, Utica, N.Y.)*

14.1.1 Low Overhung Moment

The effect of coupling overhung moment is felt not only in machine bearing loads but in shaft vibrations. The advantage of a reduction in overhung moment is not only to reduce bearing loads but to minimize shaft deflection, which results in a reduction in the amplitude of vibration. The reduction in coupling overhung moment produces an upward shift in shaft critical speeds. This change in natural frequencies results in an increase in the *spread* between natural frequencies. For many applications, reduced overhung moment is an absolute necessity to enable the system to operate satisfactorily at the operating speed required.

Low overhung moment is generally achieved with a conventional gear coupling configuration, which consists of gear meshes between a shaft-mounted hub and sleeve-spacer assemblies (Fig. 14.1).

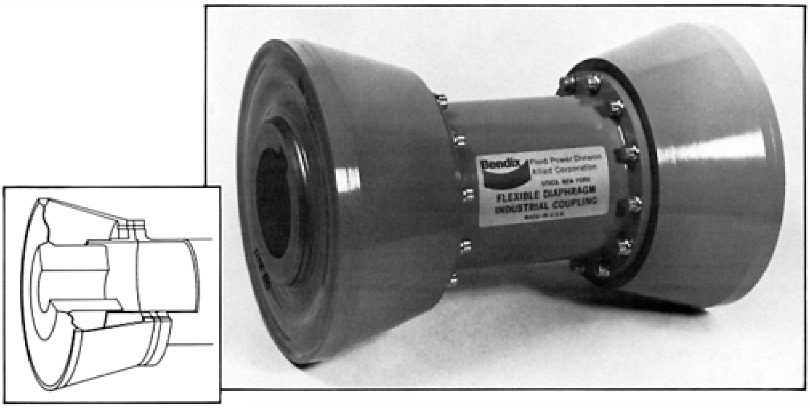

FIGURE 14.5 Low-moment-type diaphragm coupling. *(Lucas Aerospace Company, Utica, N.Y.)*

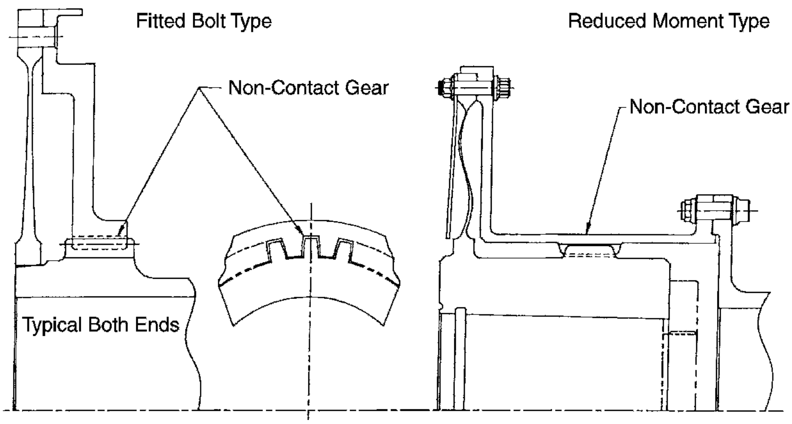

FIGURE 14.6 Contoured diaphragm safety backup coupling. *(Lucas Aerospace Company, Utica, N.Y.)*

Diaphragm couplings can achieve similar results with geometries, as illustrated in Fig. 14.5. In either case, the heavy coupling sections are placed as far back as possible so that the resultant gravity force due to the weight and center of gravity of the hub and the distributed weight of the sleeve–spacer–sleeve assembly applied at the centerline of the hub teeth is the least distance from the centerline of the machine bearings.

The diaphragm safety, or noncontacting gear type of coupling configuration (Fig. 14.6) usually consists of a spool or distance piece extension having internal gear teeth spaced between the teeth machined on the periphery of shaft-mounted coupling hubs.

The disadvantage of the concentration of the weight of the spool piece at points between the connected shaft ends can often be offset by using shorter extensions of the shaft from the machine bearing and by using a smaller and lighter-weight coupling because of the inherent large bore capacities of rigid hubs.

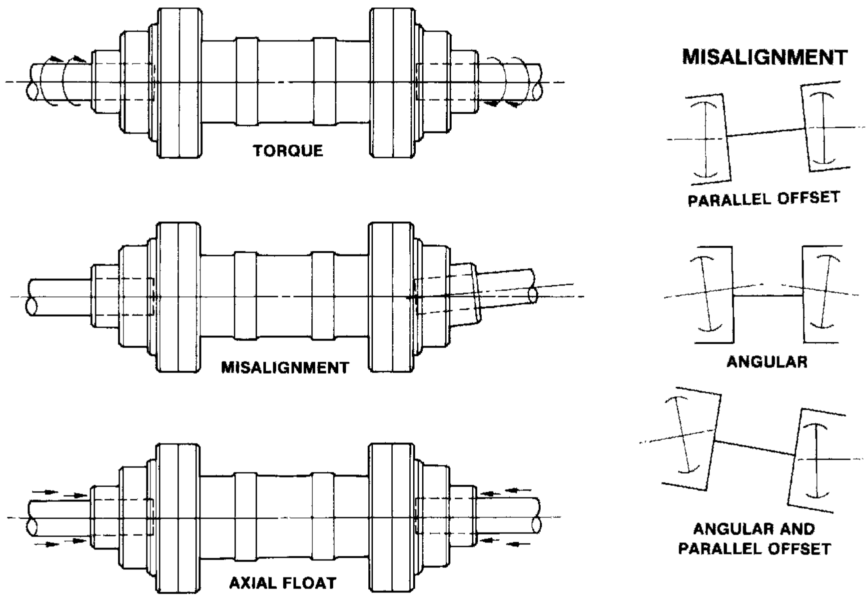

FIGURE 14.7 Functions of a coupling. *(Zurn Industries, Erie, Pa.)*

14.1.2 Low Residual Unbalance Desired

A gear-type high-performance coupling consists of a number of components fastened and hinged together by the tooth meshes. The minimum number of components for a double-engagement coupling is three. Angular and offset misalignment can thus be accommodated (Fig. 14.7). Three-piece couplings include the continuous sleeve, or flangeless coupling, and the marine, or spool, configuration. Five components—two hubs, two sleeves, and a spacer—are used in most other configurations.

In diaphragm couplings (Figs. 14.4 through 14.6) torque is transmitted from the driving shaft hub to the diaphragm rim through the body-fitted bolts and then through the contoured profile to the rigid diaphragm hub electron beam welded to the spacer tube. From here, the torque passes to the opposite diaphragm hub and through the contoured profile to the integral rim and again through the rim body-fitted bolts to the driven hub and output shaft.

Contributing factors to the total residual unbalance of a coupling are:

- *Balance correction tolerance*
- *Balance machine accuracy:* machine sensitivity and driver error
- *Arbor assembly unbalance:* mandrel and bushing unbalance
- *Mounting surface runout:* mandrel runout, bushing-to-mandrel clearance, bushing bore-to-mounting surface runout, and arbor bushing-to-hub bore (or sleeve rabbet) diametral clearance
- *Pilot surface runout:* hub bore-to-hub body pilot OD in cases where metal must be removed from the pilot OD after balancing to provide assembly clearance
- *Pilot surface diametral clearance:* hub-to-sleeve pilot clearance and sleeve-to-spacer diametral clearance
- *Hardware unbalance:* bolts, nuts, retaining rings, etc.

The implication of the aforementioned is that the straightness, concentricities, minimum clearances, and dynamic balance of the tooling are more important than the final correction tolerance in achieving an actual minimum of residual unbalance.

14.1.3 Long Life and Maintainability

The main mode of failure of a gear coupling is, in most cases, wear or fatigue of the tooth surfaces, due to a lack of lubrication, incorrect and water-contaminated lubricant, or excessive surface stress.

Assuming correct lubrication, long life of a gear coupling is attained by proper surface treatment of the teeth. General practice is to make the gear elements of high-performance couplings from chrome–molybdenum steel, or chrome–molybdenum–nickel steel that is heat treated to a core hardness of about 300 BHN.

Diaphragm and disk pack couplings are subjected to potential distress primarily when sensitive surfaces are nicked or scratched, or whenever the flexing metal parts are either pulled apart or pushed together because of hub installation errors, unexpected thermal growth, or movement of coupled shafts.

The principal advantage of nonlubricated metal disk and diaphragm-type high-performance couplings is derived from the fact that neither requires lubrication. Gear couplings, on the other hand, will suffer quickly whenever proper lubrication guidelines are violated. As already mentioned, gear-type couplings require lubrication because of the relative sliding motion between the teeth of the hub and sleeve. This sliding motion is alternating and is characterized by small amplitudes and relatively high frequencies. For example, a gear coupling on a 3-in. shaft turning at 10,000 rpm with an angular misalignment of 2 min has an alternating motion with a frequency of 167 cycles/s and a peak-to-peak amplitude of 1.7 thousandths of an inch.

Even with optimum lubrication, such a condition would probably cause fretting corrosion. Fortunately, the load on each tooth is not constant but varies twice per revolution from maximum to minimum. The ratio between the maximum and minimum force on the tooth is a function of misalignment and tooth geometry; above certain conditions the minimum becomes zero, and this means that temporarily there is actual separation between the teeth. On the other hand, as the misalignment decreases, the force on the tooth tends to remain constant, but the amplitude of the oscillation decreases also.

For use in oil-lubricated couplings, antirust and antifoaming additives and antioxidants are not beneficial. On the contrary, in the case of continuous oil flow lubrication, such additives are often retained within the coupling, causing serious problems. EP additives are not detrimental, but laboratory tests could not prove that they are advantageous in high-performance couplings. It should be noted that these couplings are designed to work under relatively low contact pressures (less than 4000 psi), and extreme pressures could be developed only during the break-in period. For this reason only, EP additives are recommended when lubricating gear couplings.

Very few high-performance couplings are grease lubricated. There are two main reasons for this: the fact that grease-lubricated couplings must be serviced more often than continuously lubricated couplings, and the fact that most of the greases available today are not resistant to centrifugal forces (and high-speed couplings certainly develop very high centrifugal forces). The 3-in. shaft coupling turning at 10,000 rpm would develop a centrifugal force equivalent to 8400 g. It is worth noting that industrial centrifuges cannot develop more than 10,000 g.

Under such high forces, greases tend to separate into their oil and soap constituents. Unfortunately, the soaps are heavier than oils, so that the teeth of the gear coupling at the larger diameter are contacting grease that has an excessive percentage of soap.

14.1.4 Continuous Lubrication Not a Cure-All

Turbomachinery installations that still apply gear couplings are likely to depend on continuous lubrication from the main oil system. The viscosity of this oil is no doubt chosen to satisfy compressor and driver bearing requirements; it is probably too light for optimized lubrication of gear couplings. Perhaps even more damaging is the fact that much of the wear product or water contamination carried by the lube oil ends up being centrifuged out in the coupling. Consider the following.

A coupling requiring an oil flow of 3 gal/min will have a total oil circulation of 1,576,800 gal/yr. If we assume a nearly perfect oil purity of only 2 parts of dirt per million and if the coupling centrifuge effect separates all of this dirt, then in one year the coupling would accumulate 3 gal, or approximately 12 L of sludge!

From the foregoing, we may conclude that

- Grease-packed couplings are acceptable for high-speed machinery only if frequent maintenance downtime can be tolerated.
- Continuously oil-lubricated couplings should be designed so that sludge is not allowed to accumulate in oil retention dams and similar discontinuities.
- The lube oil supply must be virtually free of solid contaminants and, especially, free of water. (Refer to Section 15.2 for a discussion of state-of-the-art water removal methods.)
- The user should give serious consideration to nonlubricated turbomachinery couplings.

14.1.5 Contoured Diaphragm Coupling

The design of a well-proven high-speed high-power nonlubricated coupling is centered principally about the contoured diaphragm (Figs. 14.4 through 14.6). It is this special hyperbolic contouring of the diaphragm that permits the accommodation of torque, speed, axial deflection, and simultaneous angular misalignment while maintaining uniform shear and low tensile stress. Two diaphragm profiles are employed. A straight profile is used on some models (Fig. 14.6, left side). A wavy profile that has the same hyperbolic contour and allows radial freedom of the inner hub and outer rim is used on other models (Fig. 14.6, right side).

As shown in Fig. 14.8, the diaphragm bending stress, resulting from angular and parallel misalignment, is a fully reversing cyclic fatigue stress. This stress occurs at the rate of one cycle per revolution. The diaphragm is designed to distribute this stress across the area of its profile. In addition, the transition from optimal fatigue stress in the profile to outer rim and inner hub should be controlled by generous radii.

Figure 14.9 represents a plot of fatigue data generated by actual life-cycle testing of diaphragms. Fatigue life of the diaphragm material AMS 6414 (vacuum melted 4340) is 70,000 psi at 10^7 cycles. Conservative design criteria restrict imposed bending stress to a maximum of only 35,000 psi.

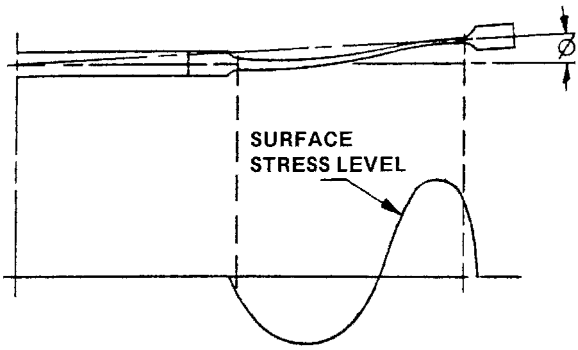

FIGURE 14.8 Diaphragm bending stress is a fully reversing cyclic fatigue stress. *(Lucas Aerospace Company, Utica, N.Y.)*

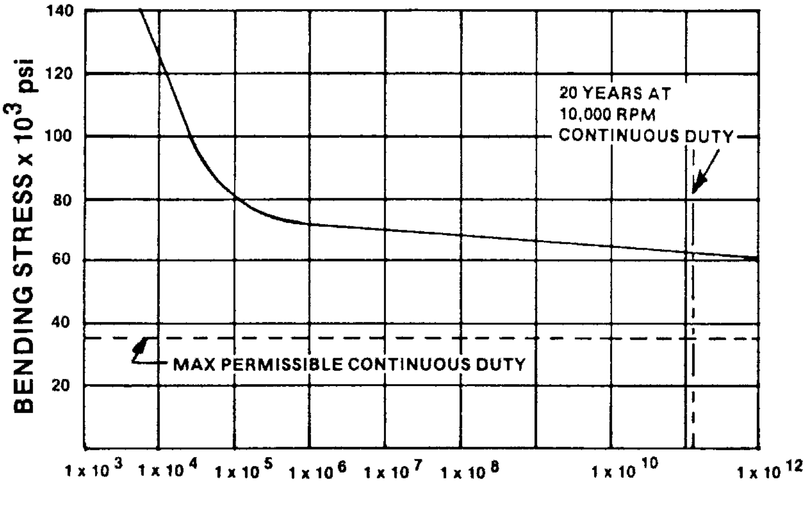

FIGURE 14.9 Fatigue life curve depicting life-cycle testing of contoured diaphragms. *(Lucas Aerospace Company, Utica, N.Y.)*

A modified Goodman diagram is shown in Fig. 14.10. The combined mean stress (steady-state axial or torsional) and combined alternating stress (cyclic axial or torsional and bending) for a given operating condition are plotted on the constant life (modified Goodman) fatigue diagram. Manufacturers such as Lucas Aerospace require that all continuous and short-term operations must have the plotted operating point fall within the area under the dashed curve. Any point within this area has a minimum cyclic factor of safety of 2.0.

For corrosion protection each flex unit is coated with multiple layers of Sermetel W, which is an inorganically (chemically) bonded aluminum coating offering a sacrificial method of corrosion protection. In this manner anytime an area of base material does become exposed to a hostile atmosphere, the Sermetel coating which is more chemically reactive than steel will be the only surface to corrode. This is referred to as *anodic corrosion protection*, which is highly successful.

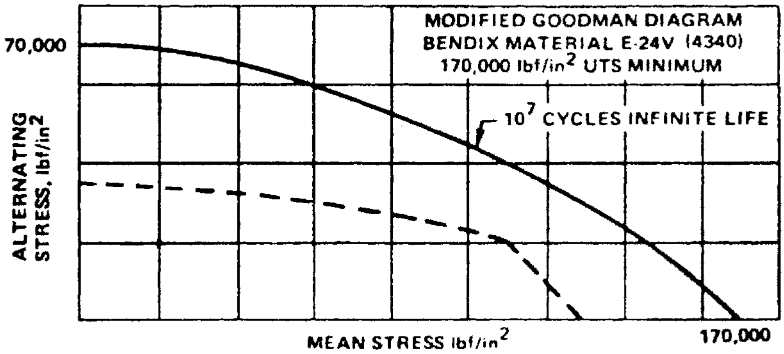

FIGURE 14.10 Modified Goodman diagram for high-performance contoured diaphragm couplings. *(Lucas Aerospace Company, Utica, N.Y.)*

Covering this layer are several coats of chemically resistant epoxy paints. Not only does the epoxy provide protection from direct contact with corrosives, but it also is very tough and helps guard the diaphragm from damage due to abrasion.

14.2 COUPLING RETROFITS AND UPGRADES*

Gear coupling replacements are desirable on many older machines. A common approach is to replicate the geometry of the gear coupling so that (1) an advantageous dry coupling fits the original space without major guard modifications, and (2) the mass–elastic characteristics comprising weight, center of gravity, overhung moment, WR^2, and torsional stiffness are replicated to avoid changes in torsional or lateral critical speeds. This approach assumes that adequate margins exist—with the original coupling in place—between running speed and lateral, torsional, or axial resonant frequencies. Rotor dynamics analysis should always be considered as a means to ensure an informed choice of coupling when retrofitting couplings on high-speed machinery.

Since connected machines function as spring systems in series, the potential exists for torsional resonance. Couplings, being the more accessible components within the system, are often modified to "tune" the overall system such that potentially dangerous torsional vibrations are attenuated. Both the flex-element configuration and, more commonly, the coupling spacer section can be redesigned to produce significant changes in total torsional stiffness. The spacer sections illustrated in Fig. 14.11 are common hybrids. Figure 14.11*a* represents a typical spacer section designed either to increase torsional stiffness or, alternatively, to increase lateral stiffness and increase whirling speed on long spans (typically, cooling tower fan drives). Figure 14.11*b* shows a quill-shaft spacer section designed to minimize torsional stiffness.

The disk coupling carries torque load as a tensile stress in the tangential link. Diaphragm couplings carry torque as a shear stress between outer and inner diameters. Steel is typically twice as strong in tension as in shear which means that for a given torque capability, the disk coupling is typically the lowest overall major diameter. In terms of flexibility, the

* Contributed by Michael Saunders and David Matt, of FlexElement Texas, Inc., Houston, Tex.

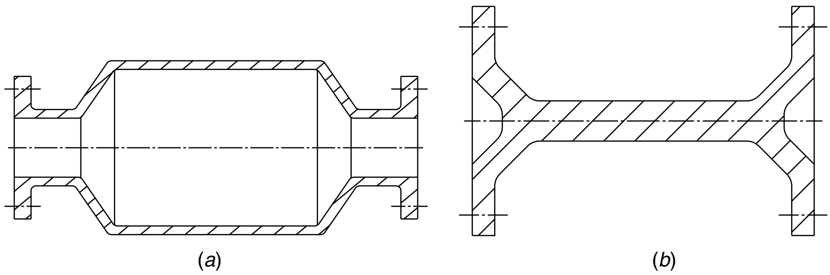

FIGURE 14.11 Coupling spacer sections designed for different radial and torsional response characteristics. (*FlexElement Texas, Inc., Houston, Tex.*)

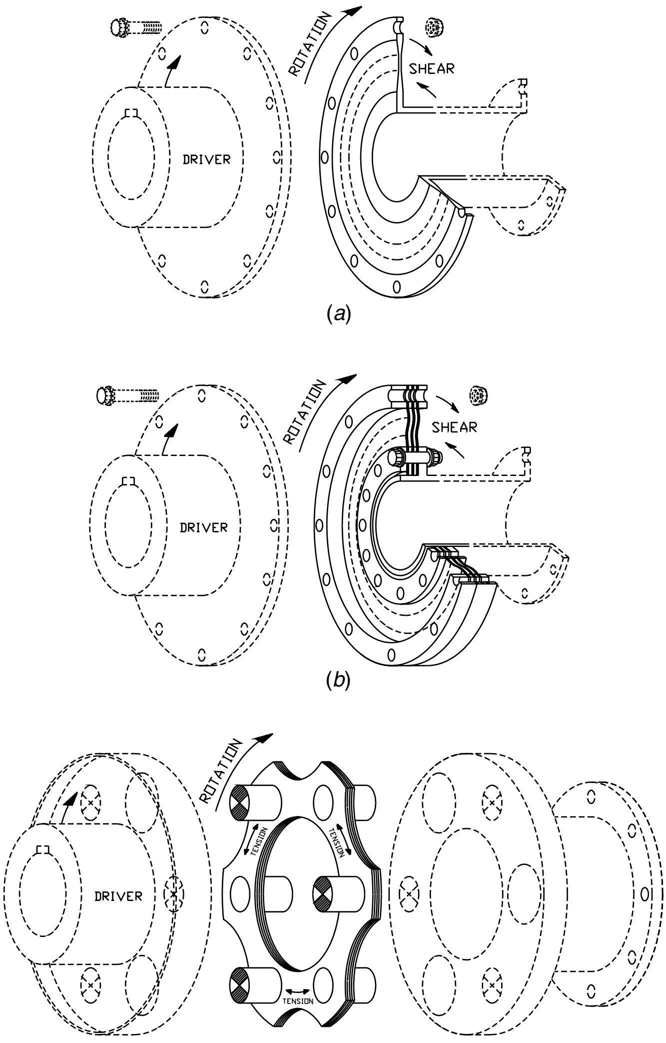

FIGURE 14.12 (*a*) Single diaphragms; (*b*) multiple diaphragms; (*c*) coupling FlexElements designed for specific retrofit applications.

disk coupling is stiff in the directions in which it needs to be stiff (torsionally and radially) and soft in the directions in which it needs to be soft (angularly and axially).This combination of favorable mass–elastic characteristics suggests that its overall rate of acceptance and range of application will continue to increase.

A comparison of coupling flexible elements designed for special retrofit applications is illustrated in Fig. 14.12. Clearly, the configuration or geometry and sometimes even the material selection of coupling components will have to accommodate certain rotor-dynamic parameters. Retrofit applications deserve close attention and cannot be left to chance.

14.3 PERFORMANCE OPTIMIZATION THROUGH TORQUE MONITORING*

Fouling deposits on blading or nozzles cause performance deterioration of steam and gas turbines. In steam turbines, evidence of these deposits is frequently not discovered until the steam flow increases to a point where no additional steam can be passed through the unit. The turbine can no longer carry an assigned load under these conditions. In gas turbine air compressors, fouling deposits reduce the amount of air available for efficient combustion. Excessive fuel consumption or reduced load-carrying capacity may result. Similarly, process gas compressors often polymerize. The resulting flow impediment can seriously influence process operation and mechanical performance. Online torque measurement systems provide an easy method of measuring produced power. Comparing fuel consumption, load conditions, and torque enables you to decide whether further steps need to be taken.

Turbomachinery performance can be restored by judicious application of onstream cleaning methods. Abrasion cleaning and solvent cleaning (water washing) are the two principal approaches. Literature, which can be obtained from original-equipment manufacturers, provides ample details of suitable procedures and their relative merit. The problem, to date, has been to figure out conveniently and accurately when to initiate an onstream cleaning process.

In purely economic terms, onstream cleaning should begin when the cumulative cost of power lost because of fouling since the last cleaning cycle equals the cost of the cleaning procedure. This is where the online torque measuring system comes into play. Installed to monitor torque at the coupling, the device also shows related speed and power. Peak torque, speed, and power values are also provided. The indicator can be connected to a computerized monitoring system, strip chart recorders, or tied to a process control computer or programmable logic controller. This additional equipment allows accessing of and correlation with steam or fuel flowmeters and heat rate tables.

Axial compressor fouling continues to be a common and persistent cause of reduced gas turbine efficiency. A 1% reduction in axial compressor efficiency accounts for approximately a 1½% increase in heat rate for a given power output. Even compressor stations not subjected to industrial pollutants or salty atmospheres are frequently prone to fouling. Torque measurements provide an early indicator of changes in efficiency.

Performance deterioration of gas turbines can be detected by combining turbine fuel flow rate with power output. Monitoring systems should incorporate readouts of power. In a

* Developed and contributed by Bently-Nevada, Minden, Nev. Additional information provided by Torquetronics, Inc., Allegany, N.Y., and by Indikon/Metravib Instruments, Inc., Cambridge, Mass.

computer system, this value can easily be compared to produced power per specific rate of fuel consumed.

Turbine manufacturers provide test stand–verified performance curves. These data are helpful in determining the degree of performance deterioration by comparing actual (fouled) condition and ideal (clean) condition specific fuel consumption rates. For efficiency optimization an operation should monitor the average fuel wastage, or average turbine efficiency, at regular time intervals.

Continuous torque sensing devices can provide valuable information in other areas as well. With torque limitations on one or both of the coupled shafts, a torque indicator can serve as a constraint control. Torque sensing can allow process optimization for computer-controlled compressors where several levels of refrigeration are available. Some turbocompressors can be configured to get desired flow and head by such methods as varying speed, varying stator blade angle, and varying guide vane angle. On large axial compressors, a combination of stator blade adjustment and speed variation of about 10 to 15% improves the part load efficiency of the compressor and increases the stable operating range. When given sufficient attention, appreciable differences in energy consumption may result, and savings of power may be realized by using torque data.

Increased energy input to the driver due to performance decay of the driver or driven equipment can be detected effectively by torque measurements. The case of a gas turbine driving a centrifugal compressor best illustrates how the issue can be resolved by measuring torque. High driver fuel consumption and high-coupling power shows that the driven machine is more highly loaded, mechanically deficient, or internally fouled. Using the dynamic torque signals and diagnostic and analytical instrumentation, procedures are available to figure out which of these three possible causes is most probable. For example, high driver fuel consumption and normal coupling power would show that the most probable cause of the efficiency decay is turbine fouling. Torque measurements and subsequent action can thus reduce energy waste in compressors incorporating antisurge controls. Equally important: Torque sensing may pinpoint causes of failure.

Although many methods exist for determining how a component failed, torque measurements may show what caused it to fail and provides clues as to how to avoid repeat failures. Looking at broken pieces can tell you how something failed. Comprehensive maintenance records can help you predict when something will fail again. Yet these methods do little to identify the cause of failure or prevent failure recurrences.

It can be difficult to figure out the cause of failure with insufficient accurate information on system loading during machine operation. Is it a running overload problem? Is it a resonance-related frequency or vibration-related problem? These and other questions can be answered by measuring torque on a running system. The data should reveal both the steady state and dynamic torque.

Bently-Nevada's TorXimitor torque-sensing system is depicted in Figs. 14.13 and 14.14. It consists of two parts: a stationary component that surrounds, but does not make contact with, a rotating system. The rotating system contains strain gauges and electronic circuitry installed on the coupling spacer or spool piece. Transformer-coupled strain gauge sensing is also employed in MCRT torque meters (www.himmelstein.com). The stationary component is installed on a mounting plate that is attached to a pedestal on the machine baseplate or equipment platform.

A similar approach is embodied in Torquetronics meters. The Torquetronics system measures torque as the shaft twists between a pair of toothed flanges made integral with a coupling spacer. A pair of phase-displaced sinusoidal signals are generated by multiple

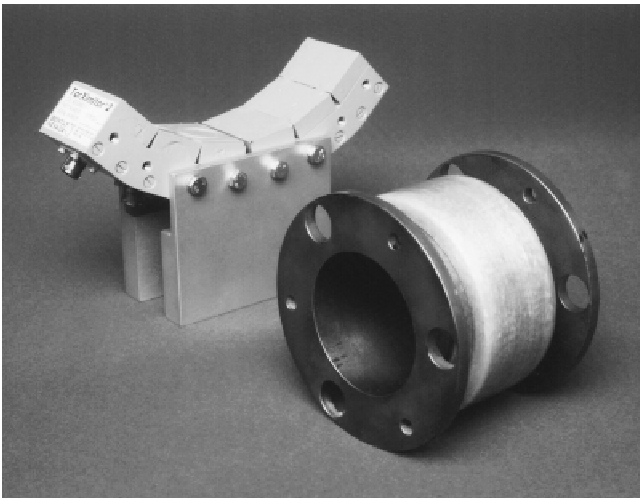

FIGURE 14.13 TorXimitor torque-sensing device. *(Bently-Nevada Corporation, Minden, Nev.)*

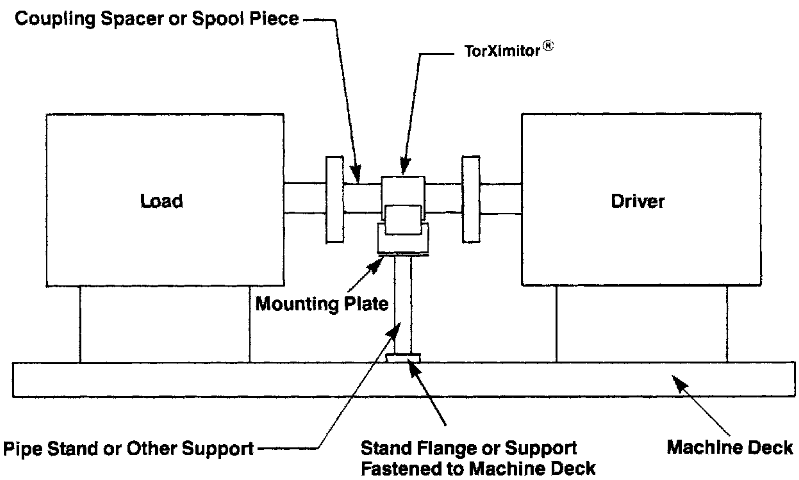

FIGURE 14.14 Schematic representation of TorXimitor torque-sensing device. (*Bently-Nevada Corporation, Minden, Nev.*)

pickups in the form of internally toothed rings surrounding fully encapsulated circumferential coils that are energized to provide a low-level toroidal flux path. Clearly, the torque required to twist the shaft through one tooth pitch must correspond to exactly 100% phase displacement.

The toothed pickup rings are permanently fixed to each end of a rigid stator tube that is supported clear of the rotating shaft from the coupled machines and often replaces the existing coupling guard. Refer to Figs. 14.15 and 14.16. The readout unit, which is a specialized digital phase meter, measures speed and the phase displacement of the two signals and converts them to torque, speed, and power in engineering units. The integrity of the

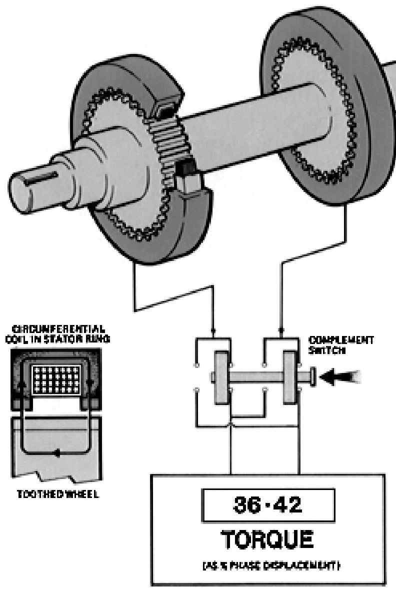

FIGURE 14.15 Torquetronics torque metering principle. *(Torquetronics, Inc., Allegany, N.Y.)*

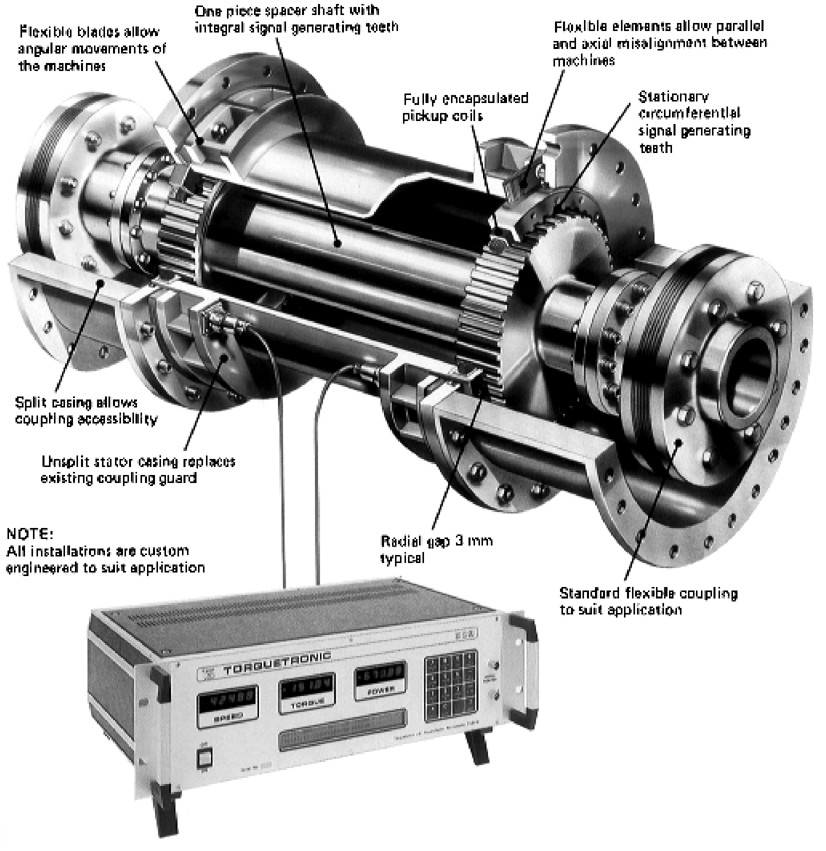

FIGURE 14.16 Cutaway view of Torquetronics torque-sensing components. *(Torquetronics, Inc., Allegany, N.Y.)*

readout unit can be proved by a complementary switch that temporarily crosses the inputs to the phase meter. If reading + complement add up to 100% exactly, all must be correct since it is most unlikely that equal and opposite errors will exist in two measurements that are seen quite independently by the readout.

Our last example describes the Indikon torque meter system. This instrumentation package can also incorporate online continuous ("hot") alignment monitoring. Figures 14.17 and 14.18 illustrate both principles.

Indikon uses a rotary transformer (Fig. 14.19) whose functioning depends on electromagnetic induction between a primary and a secondary winding, just like an ordinary transformer. In this case, however, the secondary is attached to the rotating shaft and the primary is fixed relative to the machine frame. The air gap between primary and secondary is sufficiently large to accommodate worst-case misalignment. Submersion in oil or exposure to oil mist does not affect transformer operation.

An on-shaft calibration circuit periodically generates a millivolt per volt calibration signal that goes through the same shaft electronic circuits as the torque signal. By measuring the ratio of these two signals, the effects of changes in strain gauge bridge voltage, shaft electronic circuit characteristics, and transformer coupling are eliminated, since they affect both signals equally. Zero errors due to shaft displacement are avoided by using different power and signal frequencies.

As in all strain gauge transducers, accuracy is a function of full-scale stress level. This determines whether the magnitude of the full-scale bridge unbalance signal is large enough to reduce to insignificance any residual effect of temperature on bridge balance. In general, a stress level of 15,000 psi permits a system accuracy within the range 0.20 to 1.0% of full scale. Shaft temperatures up to a maximum of 250°F are permitted.

Acceleration levels up to 20,000 *g* are permissible in those cases where the electronics package can be located along the axis of the coupling or torque shaft. In cases where the electronic package must be located on the shaft surface, a maximum of 15,000 *g* is allowable.

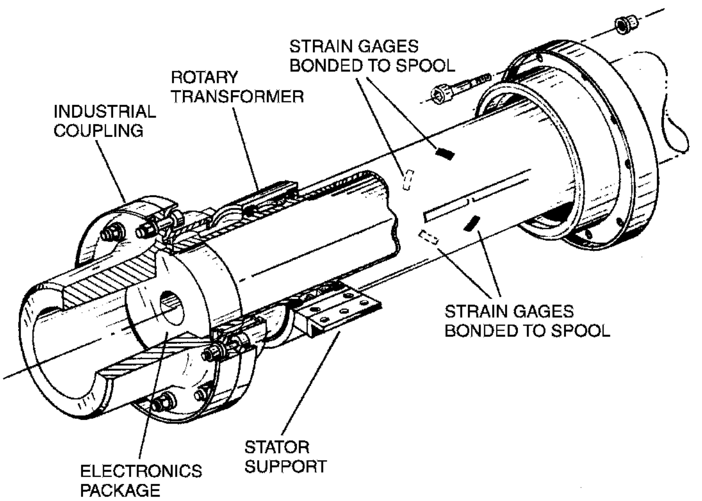

FIGURE 14.17 Indikon torque meter components. *(Indikon/Metravib Instruments, Inc., Cambridge, Mass.)*

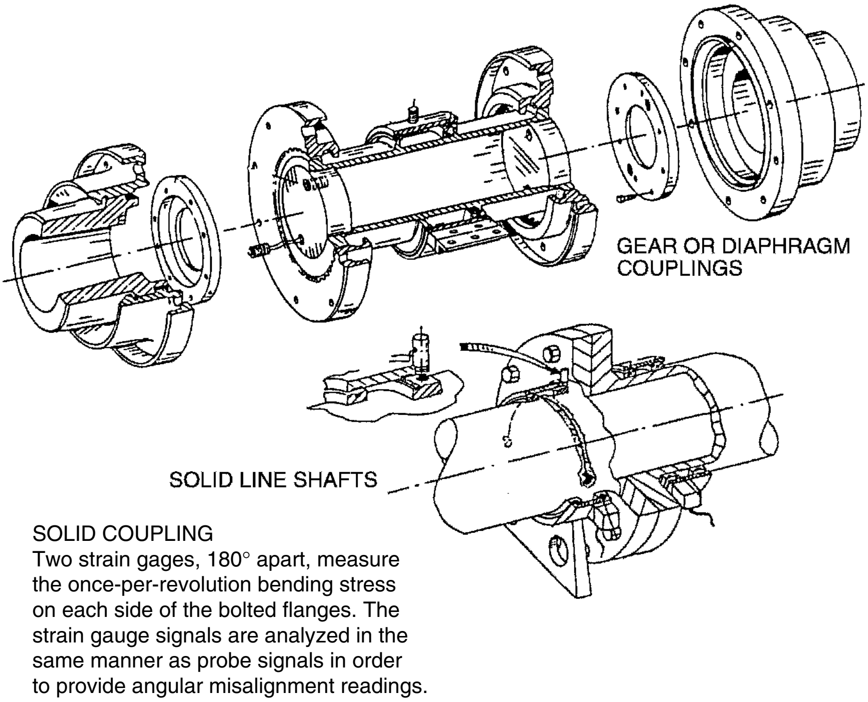

FIGURE 14.18 Online continuous hot alignment monitoring elements associated with turbomachinery couplings. *(Indikon/Metravib Instruments, Inc., Cambridge, Mass.)*

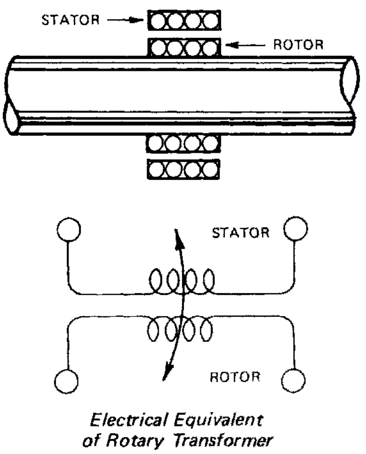

FIGURE 14.19 Rotary transformer. *(Indikon/Metravib Instruments, Inc., Cambridge, Mass.)*

Torque pulsations or fluctuations over the frequency range 10 to 500 Hz produce an analog output signal for recording purposes or for viewing on an oscilloscope.

Wherever possible, the MCRT torque meter makes use of existing couplings or shafts. If stress levels have to be increased to attain the desired accuracy, this can be accomplished by machining a short reduced section on the coupling or shaft. The resulting change in overall torsional stiffness is usually less than 10%.

In new applications, couplings with suitable stress levels can be provided by coupling manufacturers. The preferred location for the shaft electronics package is along the shaft axis. Where this is not possible, the circuits are contained in a package distributed around the shaft circumference.

Indikon's hot alignment indicating system measures coupling misalignment under actual operating conditions and displays digitally the *X* and *Y* mils/in. and mils readjustments required for realignment. It is applicable to both gear- and diaphragm-type couplings. Solid couplings require a somewhat different approach, using strain gauges.

The basic components of the system are shown in Fig. 14.18. They include inductive proximity probes, rotating with the coupling, which measure the amplitudes of the once-per-revolution variations in gap due to misalignment at each end. A marker probe at 12 o'clock detects when the shaft is in a reference position, as determined by a slot or a thin metallic target.

By comparing the phase of the misalignment signal with the phase of a reference voltage derived from the marker probe, the *X* and *Y* components of the misalignment signal are obtained and displayed digitally on the indicator. Digital displays for either *X* parallel and *X* angular misalignment, or *Y* parallel and *Y* angular misalignment, are provided, as determined by a selector switch.

Continuous analog outputs for all four quantities are provided to permit the recording of the growth of the machine into its hot aligned condition. This system can also be used in place of rim and face dial indicators to adjust cold alignment offsets to correct for the misalignment measured under hot conditions.

For gear-coupling applications, an auxiliary system is available to measure mean sliding velocity and to actuate an alarm when conditions develop that could lead to gear surface failure.

The indicator resolves the misalignment signal into its parallel and angular components by making use of two fundamental considerations:

- If the once-per-revolution signals from the probes at both ends are equal, but opposite in phase, the misalignment is parallel. The signal from the probe at one end is then resolved into its *X* and *Y* components.
- If the misalignment is angular, the vector sum of the probe signals is not zero, and the difference signal is analyzed to obtain its *X* and *Y* components.