Mission-critical couplings

HOW TO SELECT, OPERATE AND MAINTAIN THEM

JOE CORCORAN,
KOP-FLEX, EMERSON POWER TRANSMISSION

Couplings are a critical part of a turbomachinery train. They transmit the torque of connected equipment while accommodating the inevitable misalignments — angular and parallel offset — and axial displacements that occur as machine shafts move relative to each other.

There are two types of couplings used in turbomachinery applications — general-purpose and special-purpose, also known as high-performance couplings. General-purpose couplings are found on process pumps and other low-speed (less than 3,600 rpm) equipment. High-performance couplings are found on mission-critical trains, such as high-speed (5,000 - 20,000 rpm), gas turbine-driven, centrifugal compressor trains. They are relatively more expensive than general-purpose couplings and are engineered specifically for the application. They usually operate continuously for five years, often in corrosive environments at temperatures up to 500 ºF.

Turbomachinery users should know the correct selection, operation and maintenance of couplings. They can learn a lot about couplings at meetings such as the Texas A&M Turbomachinery Symposium held in Houston, Texas. This symposium hosts discussion groups where users express their concerns in the presence of peers, experts and manufacturers.

At the 2004 Turbomachinery Symposium, the discussion session on couplings and alignment was attended by over 50 turbomachinery users. There was a lively exchange of ideas and views. Below is a summary of the Q&A at the session that will help you understand the issues related mainly to high-performance couplings.

Examples of unbalance

Balancing a coupling is critical for the long life of equipment in almost all turbomachinery trains. Centrifugal forces from unbalance lead to vibration. Too much unbalance causes excessive vibration on its own, or it may be that the forcing function excites a train-lateral resonance.

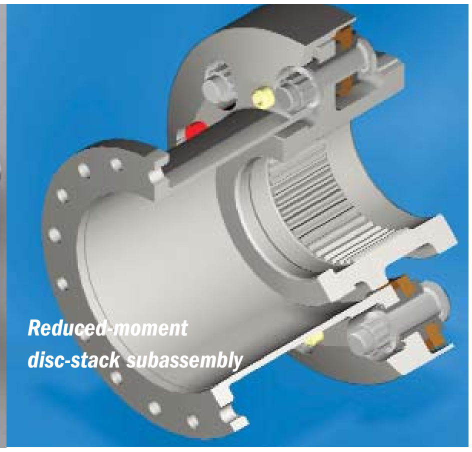
Since the force from unbalance grows with the square of the running speed, the higher the running speed, the more important balancing becomes. For example, an unbalance of 2 g-in (50.8 g-mm) at 3,000 rpm exerts a force of 1.1 lbs, while the force from the same unbalance at 10,000 rpm is 12.5 lbs.

Also, a well-balanced coupling running excessively out to the center of rotor (shaft) rotation has the effect of an unbalanced coupling. For example, a coupling balanced to low levels — a 50 lb coupling balanced in a single plane to 0.28 g-in — runs out to the center of rotation 0.0005 inches (Total Indicator Reading) when mounted. This "error" may be due to the eccentricity of the rotor mounting surface, a runout on the coupling mounting surface, or a combination of the two. This shift of the coupling from the

Figure 1: Once the coupling assembly is placed on the balance machine it can be assembly check balanced or assembly balanced

Coupling Speed	Less Sensitive machine	More Sensitive Machine
0-1,800 rpm	No balance	Component only
1,800-3,600 rpm	Component only	Component only
3,600-5,000 rpm	Component only	Component and assembly check
5,000-10,000 rpm	Component and assembly check	Component and assembly balance
Over 10,000 rpm	Component and assembly balance	Component and assembly balance

Table: Guidelines for coupling balance



Figures 2, 3: Disc couplings accommodate flexure from the metal between adjacent bolts attached to opposite flanges

running center is the equivalent of 5.7 g-in (137 g-mm) — over 20 times the level to which the coupling was balanced.

When do you balance?

You must determine what effect the unbalance has on equipment operation. Is your machine robust and running at low speed, and is it relatively insensitive to coupling unbalance? Or is it high-speed, running in close proximity to a rotor critical speed and sensitive to unbalance? Have you had a history of unbalance related vibration with the machine?

The American Gear Manufacturers Association publishes a handy guide, AGMA 9000-C90 "Flexible Couplings - Potential Unbalance Classification" that has reference charts on the required degree of coupling balance relative to the coupling weight, speed, and other factors. The International Standards Organization's ISO 1940 "Balance Quality of Rotating Rigid Bodies" is sometimes used as a guide. But this document is not as useful because it deals with rigid rotors, not flexible couplings.

The American Petroleum Institute's high-speed turbomachinery specifications require the use of API 671 "Special-Purpose Couplings for Petroleum, Chemical, and Gas Industry Service" for the type of coupling and the degree of coupling balance. These special-purpose couplings need to be well balanced and light-weight.

Here's why lighter is better: Irrespective of balance, the mass of a coupling on the end of a rotor shaft or flange directly affects the rotor's proximity to its lateral critical speeds *[1]*. Higher mass at a particular location on the shaft lowers the rotor critical speeds. So, for example, if a rotor is operating below its second critical, as is typically the case for a high-speed centrifugal compressor, a lighter coupling allows more margin between the rotor (with coupling) critical speed and operating speed.

Both the magnitude and the location of the coupling mass affect the rotor critical. The farther away the coupling portion's center of mass is to the rotor support bearing, the lower the rotor critical speed. In summary, for high-performance couplings the mass and location of the mass affect the sensitivity of the corresponding rotor to unbalance.

Choosing a balance method

Component balance means that the individual components or subassemblies of a coupling are balanced on a balance machine without subsequently assembling the entire coupling parts and placing the assembly on the machine. To achieve a better degree of balance, the balanced components can be assembled and placed on the balance machine (Figure 1), and the assembly can then be assembly check balanced, or assembly balanced.

An assembly check means that no corrections can be made to the assembly but it must not have an unbalance beyond check limits when spun on the balance machine. Assembly balance means that balance corrections are made to the assembled coupling (usually by material removal). The unbalance tolerance for the assembly balance is lower (by a factor of 5-10) than the tolerance for the assembly check.

Be careful, though, when evaluating an assembly or assembly check balance. Because of the small amounts of unbalance we are discussing, and because of the effect of error from mounting surfaces mentioned above, a subsequent reassembly of an assembly balanced or assembly checked coupling will not necessarily have the exact same unbalance readings if placed on a balance machine and checked again. The closeness to the original readings will depend on the manufacturer's machining practices and tolerances, and the accuracy of the balance machine. Normally, an assembly balance will have a lower unbalance when subsequently reassembled.

So when do you decide between component balance, assembly check, and assembly balance? Component balance is suitable only for lower speed (3,600 rpm or below), less critical and sensitive applications. It is less expensive by about 10-40% than assembly check or assembly balance. See table.

Assembly balance is reserved for critical applications where the best balance is required. This usually means higher speeds (over 10,000 rpm). However, the parts of an assembly-balanced coupling should always stay together and should be field-assembled in the exact same position as on the balance machine during the assembly balance. Match marks at mating surfaces help achieve this.

An assembly check balance allows for some interchangeability between components of one coupling and components of a second identical coupling. However, it does not guarantee that the assembly check tolerance will be met in a random field assembly unless components are exchanged during the balance operations and multiple checks are done, which are, of course, more expensive.

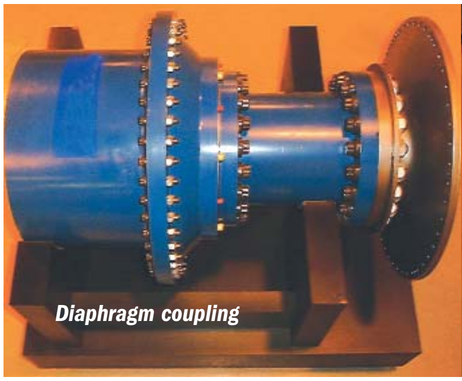
To make the assembly check more reliable, many equipment manufacturers specify assembly checking and match marking, with instructions to the field to keep coupling components together in the same orientation and only mix other coupling components in an emergency. The more sensitive applications usually require this or an assembly balance.

The table is only a guide. You can have large-diameter couplings operating at lower speeds that still require a high degree of balance. Use the additional guidelines in the AGMA and API publications mentioned earlier to be sure, or consult the coupling manufacturer.

Disc vs. Diaphragm

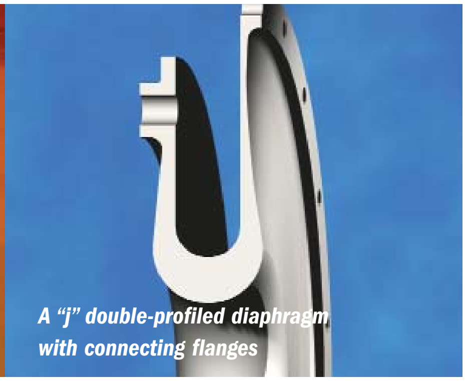
New turbomachinery applications mostly use flexible-element couplings of the disc-pack or diaphragm type. Although they have many similarities, diaphragm and disc couplings are not the same. Both work well in most cases, but sometimes one is preferred to the other. Metallic flexible-element couplings — diaphragm and disc couplings — rely on the flexure of metal to accommodate misalignment and axial displacement of shaft ends while transmitting torque. But the couplings accommodate this flexure differently.

Diaphragm couplings accommodate flex-



Diaphragm coupling

End view of a single diaphragm



A "j" double-profiled diaphragm with connecting flanges

Figures 4, 5, 6: Diaphragm couplings accommodate flexure from the metal between its outside and inner diameter

ure from the metal between the coupling's Outside Diameter (OD) and Inner Diameter (ID). Disc couplings accommodate flexure from the metal between adjacent bolts — the flex elements — that are attached to opposite flanges (Figures 2 and 3). Optimizing the flex elements produces drastically different capacities and characteristics in diaphragm and disc couplings of the same OD.

In disc-pack couplings, the individual disc elements must be thin to be flexible. Stacks of elements provide parallel load paths, and the diameter of the bolt circle is an indicator of the amount of torque to be carried. The amount of misalignment is related to the chord length between bolts and the thickness of the discs and disc packs. Since the discs are almost always put together into packs, one of the benefits of this design is multiplicity. If one or more discs fail — usually the outer discs as they are the most stressed — the rest can still carry the load until the equipment is shut down.

The other major, metallic flexible-element coupling is the diaphragm type. It comes as a single element (Figures 4, 5, and 6), or the elements can be attached together for multiplicity. But even these multiple-diaphragm couplings transmit torque from OD to ID in a radial direction. There are various styles and diaphragm profiles, but the torque transmission principle is the same.

Figure 7: A coupling spacer blackened by "smoking" hot oil from nearby bearing housing. The oil got heated by an overheated coupling guard

In both types, the flexing element is the heart of the coupling. These elements are made of high-strength stainless steel or high-strength alloy steels that are coated for corrosion protection. High-performance discs can also be coated to reduce or eliminate the fretting caused by relative movement of the individual discs under angular misalignment.

Selecting coupling types

In low-speed, general-purpose applications, diaphragms are typically not used because they are more expensive. In those applications a general-purpose disc coupling is more suitable.

High-performance disc couplings are used in a variety of special-purpose applications. For example, the multi-disc design that has reduced moments (Figure 3) is ideal for high-speed centrifugal compressor applications where mass location is crucial to eliminate potential lateral vibrations. Since the torque is transmitted circumferentially from bolt to bolt through the discs, the discs can slide over the hub and shaft end where the disc pack is closer to the equipment bearings. The mass is closer to the support bearings.

The disc-pack will generally also be smaller in diameter than a diaphragm type, which must transmit torque from OD to ID, and therefore will have less windage related problems (See next section on coupling guards). It will travel at lower surface speed and therefore will not heat up or shear as much air in the coupling guard.

Large gas and steam turbines (e.g., GE Frame 6, LM 6000, TP&M FT8) are ideal applications where the diaphragm coupling's diaphragm can bolt directly to the turbine flange thereby giving the best center of gravity location relative to the turbine bearing. Though many times only a single diaphragm is used per coupling end, a well-designed diaphragm is quite reliable, and can handle large amounts of axial travel. Some single diaphragms can accommodate +/- 1 inch (25.4 mm) of axial displacement. In addition, some generator or motor applications with small misalignment and axial displacements use a thick power-dense diaphragm with less, unneeded misalignment and axial travel capacity.

Other variations include a double diaphragm, with two identical profiles per end, or even multiple diaphragms in a pack per end, which are used for high axial misalignment along with high torque loads. There are also "J" diaphragms, which have two different profiles per end. These variations can all work well in gas turbine applications if properly designed.

Coupling guards

The coupling guard's main function is safety. It prevents a worker from inadvertently getting too close to the rotating coupling.

Even though dry couplings have been around for years, the associated coupling guards are sometimes a nuisance. They sometimes get overheated and can exhaust oil spray and even smoking oil into the surrounding area.

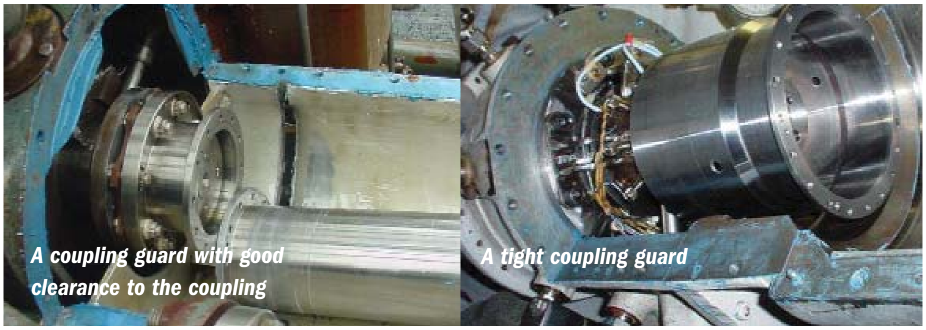
Dry-disc and diaphragm couplings which do not have the cooling effect of continuous oil lubrication for gear couplings, are being increasingly used. And the guards for these couplings must be carefully designed so that they do not get too hot. The coupling shears the air inside the guard and imparts energy to the air, which gets hot if not cooled or exchanged making the guard hot. So coupling guards should be designed to have surface temperatures cool enough to protect personnel. The upcoming fourth edition of API 671 will recommend a limit for the guard temperature.

Further, any large-diameter coupling in close proximity to a bearing housing seal will suck oil through the seal into the guard. This is because the surface speed at the larger diameter is less than at the shaft, causing a pressure differential and therefore a vacuum at the shaft level. Since gear couplings tend to be smaller in diameter for the same torque and run at lower speeds, oil suction is less of a problem. When large-diameter, dry couplings run at higher speeds close to the bearing seals, more oil can be sucked in, and this oil can get hot if heat exchange and cooling do not occur. The oil smokes and "cooks" on the coupling, turning it black (Figure 7).

The best guard design

It is extremely important to design the guard with as much clearance around the coupling as possible. This is not always easy to do because of constraints such as piping, but it is the most effective way to reduce windage related problems. Note in Figure 8 the generous clearance around the coupling and the distance from the coupling to the housing. This coupling is not an anti-windage design (the disc pack is uncovered). In contrast, Figure 9 shows a tight fitting coupling incorporating special guard and coupling features to eliminate excess oil ingestion and windage.

If the coupling needs to operate in a tight guard at high-speed, consult papers such as "The Design of Coupling Enclosures," available on the Kop-Flex web site (http://www.emerson-ept.com/eptroot/kopflex/default.asp) , or other manufacturers' reference materials to design a properly functioning guard. Also note that if the guard temperature cannot be lowered to acceptable levels for safety precautions, and it is still acceptable as far as the oil is concerned, an expanded metal guard similar to Figure 10 can be placed over the first guard. TI



Figures 8, 9: Clearance between coupling and guard helps avoid windage-related problems

Footnote

[1] Rotor critical speeds are rotating speeds which correspond to the natural frequencies of a rotor. They are associated with potentially high vibration amplitudes unless these vibrations are avoided or suppressed. The first critical is the first fundamental natural frequency, the second is the second, and so on.

Figure 10: Expanded metal coupling guard protects personnel

Author

Joe Corcoran is manager of High Performance Engineering for Kop-Flex, Inc., a division of Emerson Power Transmission. The group he heads selects and designs couplings, and processes orders and inquiries for high-performance couplings and torquemeters. Previous to his 19 years at Kop-Flex, he was an operations engineer responsible for two 80 ton-per-day Union Carbide-Linde oxygen plants at Baltimore.