Couplings - Balancing Tutorial & "New Developments in Gas Turbine Couplings"
by Joe Corcoran & Christian Wolford

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Introduction

In recent years both industrial frame and aeroderivative gas turbines have advanced significantly. Power has increased as much as 200% within the same envelope. Designed for reliability and application flexibility, gas turbine technology continues to evolve. Improvements that were years in the making are now commonplace thanks to computer based designs and the development of advanced materials. There are constant demands for lighter weight, low maintenance designs that are capable of continuous operation of up to 5 years.

As advancements in gas turbine have occurred, so have the flexible couplings. Whether mechanical drive or generator, disc or diaphragm type, today's couplings must meet the stringent requirements of the latest gas turbine designs and application. Early gas turbines that required flexible couplings eventually approached gear coupling limitations in the late 60's and early 70's. Because of the short intervals between inspection, gear coupling maintenance and vibration caused by coupling unbalance, advancements in flexible membrane couplings were necessary to provide reliable, long term operation of gas turbines.

This tutorial compares the characteristics of various types of flexible gas turbine couplings: diaphragm and disc primarily, with some coverage of the declining gear coupling installed base. Rigid or quill shaft couplings are not covered. Analysis of these different coupling types will show that they do not react in the same way while performing their function of transmitting torque and accommodating machinery misalignment. Proper selection and sizing, service and safety factors, rotor dynamic influences and environmental conditions address the importance of the flexible coupling in the turbomachinery train. Balance methods, the reactive forces that each type can produce on the equipment, advances in manufacturing, materials and design tools are reviewed. Special features such as torquemeter systems, fail safe back-ups, and overload protection devices will also be covered. A review of failure modes is also included.

Gas turbine technology has steadily advanced since its inception and continues to evolve. Significant research and development has resulted in ever smaller gas turbines for the same power output. Computational fluid dynamics (CFD) and finite element analysis (FEA) computer designs along with material advances have allowed higher compression ratios and temperatures, more efficient combustion, better cooling of engine parts and reduced emissions. Major advances in seals (mechanical and dry gas) and bearings (magnetic and foil) have dramatically extended maintenance intervals while state of the art electronics have allowed for continuous monitoring and operation of machinery trains for periods lasting years with minimum downtime.

These turbines, first used primarily in aerospace and military applications, are now found everywhere in industrial and marine applications. The same technology that powers jet aircraft is the basis for land- based drivers. For example, a popular turbine manufacturer's gas generator, used to power the Boeing 747 and other aircraft, also powers various power turbines mechanical drive, turbocompressor and generator applications (figure 1).

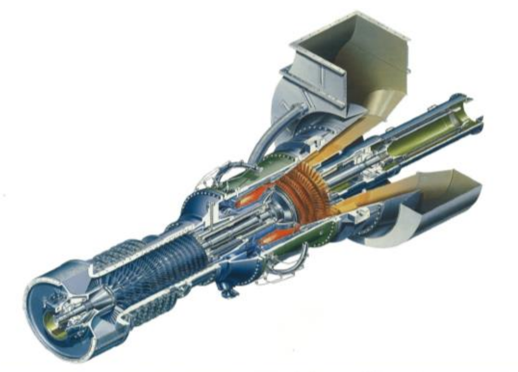


Figure 1

Most gas turbines are considered "critical equipment" and therefore require "special purpose" couplings. Critical equipment is loosely defined as those machines whose power is greater than 1000KW, whose reliability is essential to plant operation and whose failure would cause a unit to shut down. Special purpose couplings are defined by those machines that are expected to operate for an extended period of time, typically 5 years, unspared and requiring no maintenance. Because these machines produce high power they are often sensitive to certain loads. Angular and axial forces caused by misalignment that would seem insignificant to robust steel mill machinery can shorten the life of these critical machines. As a result of that sensitivity, speed and power, coupling requirements are very demanding.

Until the introduction of flexible membrane couplings in the 70's, flexible gear couplings were all that was available for gas turbine applications. Gear couplings transmit torque and accept misalignment by meshing and sliding of gear teeth (figures 2 &3).

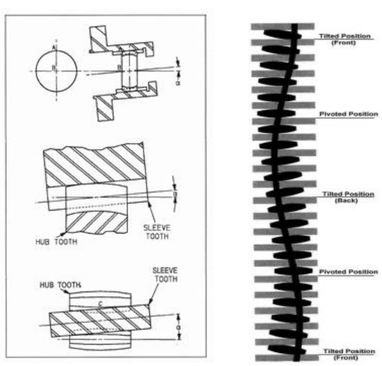


Figure 2 &3

Although "special purpose" gear couplings were designed and built to tight tolerances, to accept misalignment they must have radial clearances which results in higher unbalance and vibration (Figure 4).

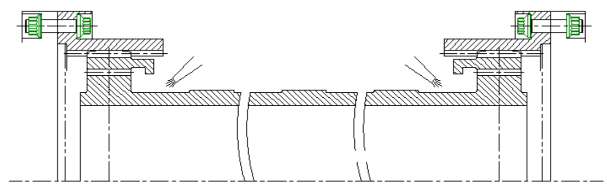


Figure 4

Gear couplings wear and as they do unbalance increases. This reduces their reliability and raises questions about how long they can operate before inspection and maintenance is required. They also put high axial force and bending moments on the equipment when misaligned as the force to slide the mating teeth is a function of torque. This can shorten the life of other machinery components like bearings and seals. So, to extend the period that turbomachinery could operate before inspection intervals, flexible couplings had to advance in technology and design.

Unlike today's flexible membrane couplings, gear coupling failures were often a result of lubrication problems. Foreign materials like dirt and dust can migrate into the lubrication while metal particles from wear mix to form sludge that can "lock up" or rigidize a gear coupling (Figure 5). Other lubricant issues include inadequate supply and centrifugal separation of the base and oil. In all these cases, serious damage can result from improper maintenance (Figure 6). Special purpose gear coupling manufacturer's generally recommend inspection intervals of 18 months which is impractical and very costly. As gas turbine technology advanced in the early 70's, power, speed and operating temperatures increased and gear coupling technology was at its limit. The coupling industry responded with the development of metallic membrane couplings. Since then disc and diaphragm couplings have become the standard for turbomachinery. Although they offer similar advantages, they are not the same.

Figure 5 & 6

Diaphragm Couplings

By definition, diaphragm couplings accommodate flexure from the metal between the couplings outside diameter (OD) and inside diameter (ID) (Figure 7).

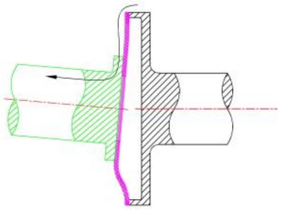


Figure 7

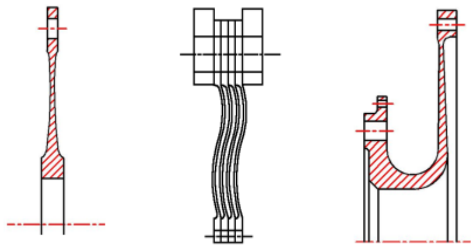
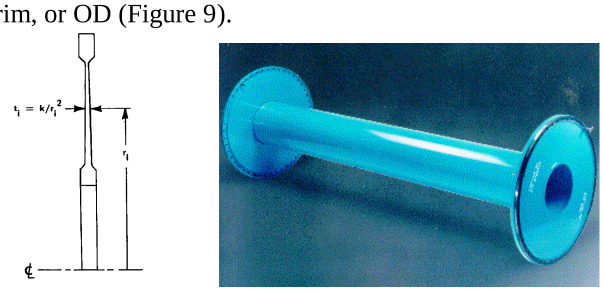


Figure 8

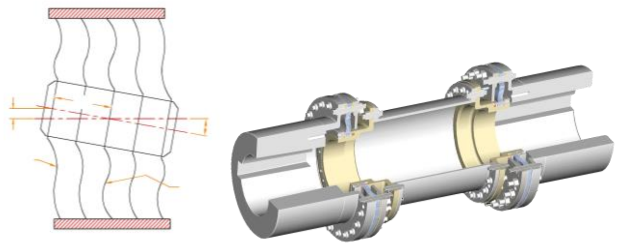
Diaphragm types can be classified as couplings that utilize a single or a series of “plates”, or “diaphragms”, for the flexible members. All shapes have some type of profile modification that helps reduce size, increase flexibility and control stress concentrations (Figure 8). A contoured diaphragm coupling typically uses a single “plate” machined from a solid disk of heat-treated alloy. The diaphragm is contoured so that it has a nearly uniform torsional shear stress throughout the profile, which is therefore thicker at the hub, or ID, and thinner near the rim, or OD (Figure 9).



Figures 9 &10

The purpose of contouring the profile is to keep the diaphragm as thin as possible consistent with the transmitted torque, while bending and axial stresses are designed to be as low as possible. A thicker diaphragm has greater torque capacity, but is not as flexible and vice versa. Compared to gear couplings, bending moments are less than half and axial forces can be as low as 10% because they are determined by the spring stiffness of the element and independent of torque. The Bendix Corporation (now Goodrich Aerospace) first patented the single element contoured diaphragm coupling (Figure 10) after years of research and development. This aerospace proven technology yielded a most reliable and lightweight approach to transferring torque and misalignment. This design incorporates radial and axial welds to connect the diaphragm I.D. to the spacer piece. In 1967, Bendix supplied the first contoured diaphragm coupling for industrial frame gas turbines and has supplied well over 120,000 contoured diaphragm couplings over the past 4 decades.

In 1971, Zurn Mechanical Drives (now Ameridrives Couplings), developed a multiple convoluted diaphragm design utilizing a series of thin stainless steel plates rather than one thick one (Figure 11 & 12).



Figures 11 &12

This unique design provides greater flexibility and lower stresses as the stresses are proportional to the material thickness cubed. The Ameriflex multiple design offers the safety feature of redundancy and therefore a gradual predictable failure mode. Building on the experience of the original design, which separated the diaphragms with rings at the inside and outside diameter to prevent fatigue failure by fretting, today’s design integrates the fillers between diaphragms by machining the flexing area of slightly thicker diaphragms (Figure 13).

This reduces the manufacturing process and therefore cost. The ability to customize material thickness, convolution depth, number of diaphragms and O.D. flange bolt pattern enables gas turbine manufacturers to optimize and tune the design in a power dense coupling.

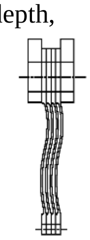


Figure 13

In 1995, Kop-Flex introduced a single, contoured diaphragm coupling that is bolted to the spacer (Figure 14).

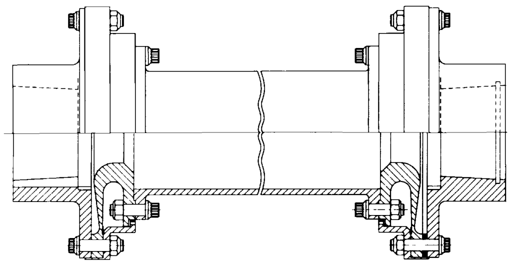


Figure 14

Figure 16

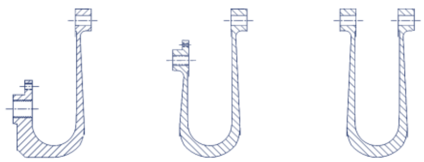


Figure 15

Popular on industrial frame gas turbines, this simplified design is often preferred over the welded design. Available, in “I”, “J” and “U” shapes the element can be customized for misalignment and axial stiffness requirements (Figure 15). Eliminating the weld reduced the lead time and offers the advantage of a field interchangeable flex-element and therefore less costly spares.

Profiles of several diaphragm designs have developed over the past 15 years. In an effort to meet the stringent requirements of turbomachinery manufacturers, diaphragms may be designed for a particular turbine, gearbox or generator where one application may require more torque and another lower axial stiffness. This is possible because the stresses on these diaphragms are well known by FEA and modern machine tools are capable of highly precise machining of these components. Diaphragm materials vary by manufacturer but are all made from high strength material. Some are made from corrosion resistant 15-5 and 17-4PH stainless steel while the standard material for welded designs is 4300 alloy steel which is corrosion protected with special coatings. Machined diaphragms are usually shot-peened to reduce the residual stresses imposed during manufacturing and to prevent the development of surface crack initiation points. Diaphragm couplings are typically marine style which allows the mounted hub to vary in bore size considerably without affecting the diaphragm diameter (Figure 16). This allows the size to be based on torque and misalignment requirements rather than the rotor shaft diameter.

Most diaphragm couplings have machined guards to protect the element from nicks and scratches that would act as stress risers on the diaphragm. These guards also act as an anti-flail device to captivate the spacer in the unlikely event of a failure.

Machined guards can also be used to pilot the flanges, diaphragms and shim sets as required by many specifications to meet balance repeatability requirements and is the preferred method to using body bound bolts for centering components (Figure 17). They also make field interchangeability of components more practical for some designs.

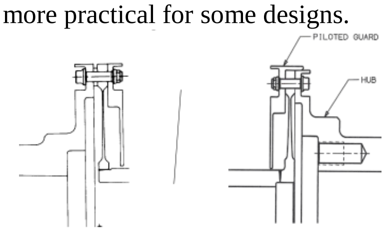


Figure 17

Large gas and steam turbines are ideal applications for diaphragm couplings. .Many of these machines interface with the coupling by an integral flange. If the flanges are designed to bolt directly, the overhung moment is reduced and mounting is simple since there is only one set of fasteners. If the flanges do not mate, a plate or short spacer is necessary (Figure 18). Diaphragm couplings can handle a large amount of axial travel often required on turbines. The contoured and convoluted diaphragm designs can operate as well under high misalignment as zero deflection since there are no fretting corrosion possible. Diaphragm couplings have been designed to accommodate over $\pm$ 25mm.

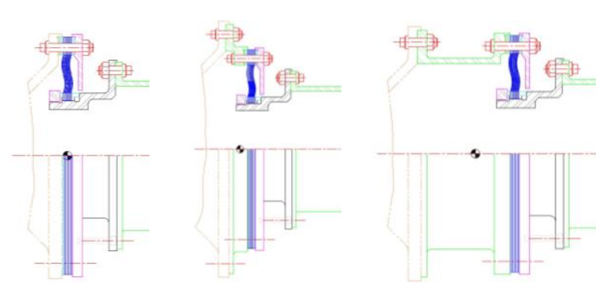


Figure 18

For those applications that require a low overhung moment due to rotor dynamic concerns, the diaphragm coupling can be designed in the reduced moment configuration. Whether welded, bolted or splined, diaphragms are mounted to the hub in this configuration (Figure 19). Because the hub is inverted and must fit under the diaphragm, reduced moment diaphragm couplings limit the hub bore and may not be the best solution when compared to a disc type.

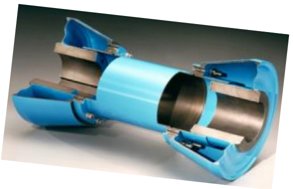


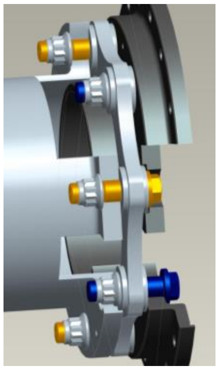
Figure 19

The diaphragm type will generally fail from either over-misalignment or over-torque. Failures from angular misalignment start as cracks in the diaphragm web, as stress from angular misalignment is alternating with each revolution. Axial misalignment can contribute to the stress and failure, though it does not stress the diaphragm in an alternating fashion (Figure 20). Torque overload will cause one or more ripples in the diaphragm (Figure 21) or a complete failure in shear (Figure 22)

Figure 20, 21 & 22

Disc Couplings

Disc couplings by definition transmit torque from a driving to driven bolt, tangentially, on a common bolt circle. These couplings are made up by a series of thin, laminated disc assembled in a “disc pack”. Misalignment is accomplished by deforming or bending of the material between the bolts (Figures 23 & 24).



Figures 23 & 24

The disc profile has evolved from round to polygon to scalloped and vary by manufacturer (Figure 25). The circular is the least popular as the lines of force tend to try to straighten out the curved segment resulting in excessive compressive and tensile stresses. They are more often used on API and ANSI pumps. The other shapes keep the force lines within the boundaries so the element has mainly tensile stresses. Since fatigue would occur at the bending location, the wider cross section is next to the bolts where it does the most good.

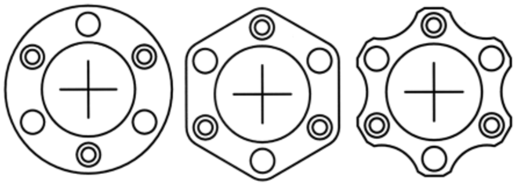


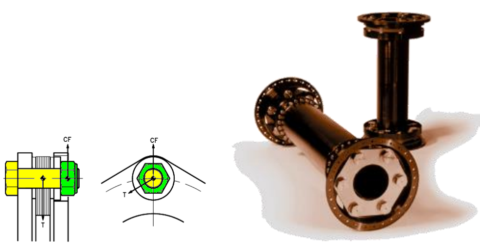
Figure 25

The 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 disc and disc packs. Misalignment limits are a function of the bending that can be accommodated while under tension. The unit will allow axial movement due to thermal growth of the connected machines. There is, of course, a limit so disc couplings are often pre-stretched so they can operated closer to the centerline of their deflection. An important feature of disc couplings, and those that work in the same manner, is the low reactionary load that is transmitted to the coupled machines compared to gear couplings.

As is the case for other types of couplings, additional considerations are required to properly design and select special purpose couplings. In order to save space and reduce costs the coupling designer will select disc packs to closely match the application and in some cases specifically to meet the requirements of a particular machine. The design will include bolt circle diameter, number and size of bolts, and the number of discs needed. Individual discs are factory assembled in a unitized "disk pack" and are not to be disassembled.

Designing for strength is a function of the disc pack material and the shape of the discs at critical points such as bolt attachments. Typically, high performance disc are made from cold rolled stainless steel; generally 300 series. For special applications such as environmental issues that cause corrosion, discs can be made of Monel, Inconel, PH stainless and others.

Ruston Gas Turbines was the first to use disc couplings on gas turbines. The Turboflex polygon shaped disc coupling, manufactured today by Bibby Transmission, was designed for use on Ruston engines. These couplings are generally more power dense than other discs because of the blade design and large, oversized fasteners to tightly clamp the disc pack. Bolts are sized such that, for normal operating conditions, drive through the discs and flanges is in friction (Figure 26 & 27).



Figures 26 & 27

The scalloped disc is optimized to give it a more uniform stress distribution across the flexing plane. Recently, further optimization has seen discs scalloped on both outside and inside diameter (Figure 28). Two manufacturers apply a coating, similar to Teflon, to each disc to minimize or eliminate the fretting under high misalignment (Figure 29).

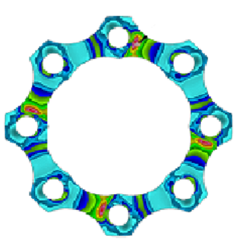


Figure 28

Figure 29

Turbocompressor manufacturers generally prefer to use disc couplings between compressor bodies and from gear to compressor since many of today's machines require the use of a reduced moment coupling. Because today's turbocompressor rotors are lighter in weight and turn faster, the coupling has become an appreciable percentage of the rotor mass system and therefore sensitive to the overhung moment of the half-coupling. It is estimated that over 50% of new turbocompressor applications require the use of a reduced moment coupling. Disc couplings have a significant advantage over diaphragm in the reduced moment design because they can accept a larger shaft due to their torque transmission path. Since the torque is transmitted circumferentially, the discs pack can be located over the hub and accept a larger shaft compared to the diaphragm design (Figure 30). Disc couplings are typically smaller in diameter, lighter in weight, and impose lower forces on the connected equipment for the same given torque due the multiplicity of design. The reduced moment disc coupling is also ideal for gas turbine applications with output shafts (rather than flanges) where reducing the overhung moment is crucial to eliminating potential lateral system problems. Where heat and horsepower loss due to windage are a concern, discs couplings tend to be a better choice because there smaller diameter and therefore less surface speed.

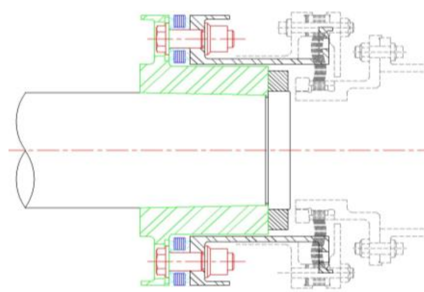


Figure 30

If properly applied, installed, and maintained, the useful life of flexible membrane couplings is often longer than the machines they connect. There are no "hard and fast" rules for selecting one type of coupling over the other. Generally, both disc and diaphragm couplings can be designed for any turbomachinery application. Those that specify one over the other normally base their decision on experience be it good or bad or on recommendation from the OEM. When evaluating cost, the disc coupling tends to be less expensive in the smaller sizes while the diaphragm type, being more power dense in the larger sizes, may be less expensive. In today's competitive world, the question of whether it is necessary to supply the best coupling for the application or one that is "good enough" is becoming a topic of discussion and why end users are getting involved in the coupling decision on turbocompressor trains.

Specification control

When selecting flexible couplings for critical equipment, it is important to understand the difference in specification control of a mechanical drive versus a turbine generator drive (Figure 31). Gen-sets are most often pre-engineered and sold as a package. The turbine manufacturer determines the coupling design. The interface connection and distance between shaft ends is done at the initial design stage and the rotor analysis is typically performed only once. One or two coupling vendors are approved for a particular gen-set package and therefore the coupling choice is limited. Conversely, most mechanical drives are custom designed for a given application based on the customer's requirements. For these machinery trains, the compressor or pump original equipment manufacturer (OEM) is responsible for the coupling selection. The interface connection, distance between shaft ends and application requirements are normally different for every train. The coupling characteristics or mass elastic data is provided to the (driven) OEM for the train rotor analysis. If the result of the computer model determines that there is a lateral or torsional resonant frequency near the operating speed, the coupling is redesigned to "tune" the system. Mechanical drive end users are often involved in the specification of the coupling in efforts to indicate a level of quality and/or vendor preference. Some users leave the decision to the OEM but it is estimated that over half of users in the oil and gas markets have a specification that the OEM is expected to meet. Typically these specifications follow the intent of API-671. In addition, for most off-shore projects, couplings must also meet stringent specifications from insurance companies like Det Norske Veritas (DNV) and American Bureau of Ships (ABS).

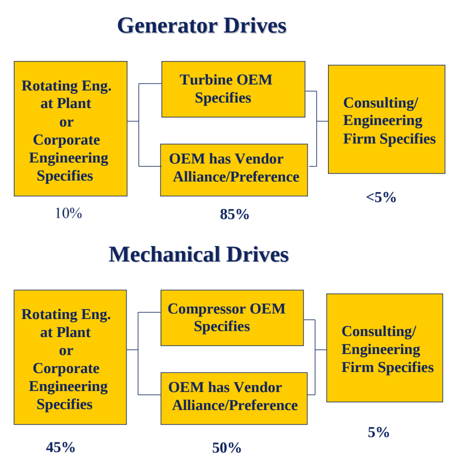


Figure 31

Failure Modes

Metallic element couplings are designed for infinite life provided the combined stresses do not exceed the element's endurance limit. Unfortunately, due to unforeseen events, high performance couplings can and do fail. The most common forms of failure are caused by over-misalignment (over 80%) or over torque. Over- misalignment results from excessive angular or parallel offset misalignment and/or excessive axial displacement. Combination misalignment and torque failures do occur but typically it is caused by one. An angular failure applies an alternating stress on the metallic flexible element or elements. The element(s) bends back and forth each revolution to accommodate the machinery angular or parallel offset misalignments. So the failure mode from these excessive misalignments is bending failure.

As mentioned previously, one of the benefits of disc pack couplings is multiplicity. In the event of a failure due to misalignment where one or even few discs break, the others may still operate, at least for a short period to time, depending on the magnitude of the load. In a disc pack coupling the outer discs fail first as they experience the highest stress from being furthest from the center of bending (Figure 32 & 33). The remaining discs will need to handle a higher torque but less misalignment so failure is often gradual. After enough discs break, there will be enough unbalance to cause higher machine vibrations, so that a decision can be made to shut the connected machines down and investigate the problem.

Figure 32, 33 & 34

The other major cause of failure is torque overload. In the event of failure due to generator short circuit or water ingestion in a gas compressor for examples, the major metal parts of these type coupling will yield. They will then break if the load is large enough or if the coupling is continues to operate in the yielded condition. In Figure 34, the discs are severely distorted from a torque overload. What is not readily seen is that the connecting bolts and bushings are also distorted (yielded). This was made clear when the failed pack was disassembled. Note this photo was taken with the equipment shut down and therefore no load on the coupling.

Under load, with a strobe, you could see a similar condition on the unloaded links in the coupling. Some gaps in the links or "tin canning" are normal for some types of disc couplings (Figure 35).

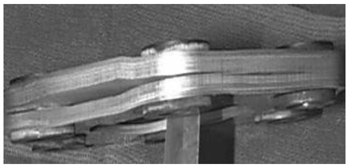


Figure 35

The key to determining an overload is the condition of the disc packs with no load, damaged or bent discs and bolts under a strobe light. This type of inspection is not recommended outside a lab so it is best to contact the coupling supplier for inspection procedures.

The coupling manufacturer can provide various graphs and diagrams that address coupling ratings. These ratings are based on combined stresses. For turbomachinery, torque is considered a steady state stress. Axial deflection due to thermal growth is also a steady state stress. Angular misalignment is an alternating stress. Figure 36 shows a typical Goodman Diagram. Other capabilities can include the relationship between parallel offset and /or angular coupling misalignment and axial misalignment (Figure 37). Turbomachinery manufacturers may also require axial force vs. deflection graphs to insure that the force on the bearings is within "limits".

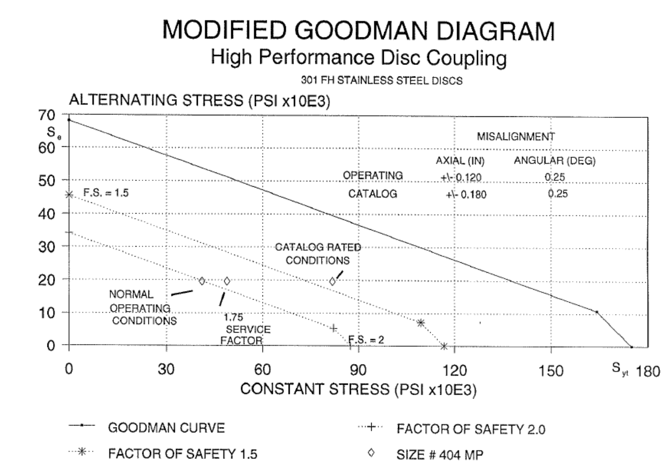


Figure 36

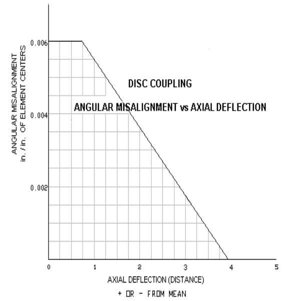
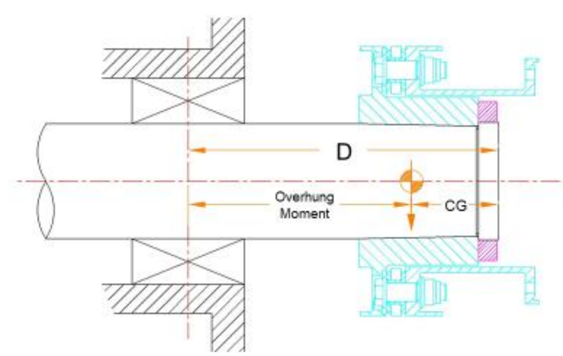


Figure 37

Table I compares the coupling characteristics for the Load and Accessory drive of a popular gas turbine. Two different gear tooth design are compared with dry couplings (disc and diaphragm). Gear couplings are more power dense than the flexible metallic element disc or diaphragm couplings, so they are often smaller in diameter and lighter in weight. For both the accessory and the load coupling, the bending moment for the gear coupling is larger than the dry couplings. This is especially true for the load application.

Note that the axial force (and to a lesser extent the bending moment) of the gear couplings is dependent on the torque and coefficient of friction. Since the accessory couplings were designed for a relatively small continuous load, but a large start-up load (not shown), the axial force from the gear couplings is comparable to the dry couplings. For the load couplings, with much higher continuous torque loading, the axial forces are much lower for the dry coupling.

In Table 2, note the improved centers of gravities of the reduced moment couplings compared to marine styles. The overhung moment of the half coupling is the product of the half weight times the distance from the centerline of the journal bearing to the shaft end (Figure 38).



½ weight x (D-CG) = overhung moment

Figure 38

This value is cubed when modeling the train's lateral critical so it becomes clear that reducing the coupling half weight has much less influence than moving the CG location. The center of gravity is given in terms of how it relates to the shaft end. As the half coupling's CG moves up the shaft toward the bearing it makes sense that since the distance is subtracted it would be in the "minus" direction. However, some coupling suppliers consider this in the "plus" direction so it should be reviewed when comparing coupling offerings.

A similar caution applies to the use of these tables. Coupling designers will often trade off alternating and steady state stresses to optimize a selection to meet specific requirements. For example, by de-rating the angular misalignment, the torque rating is increased. These tables are typical catalog values for a specific design. In another example, the marine style single diaphragm could have been a smaller size if the torque were reduced by 10%.

Coupling Type	Coefficient of friction	OD (in)	Weight (lbs)	Continuous Torque Condition (Lb-In)	Axial Force at continuous conditions (0.25")(lbs)	Continuous Angular Misalignment (Degrees)	Bending Moment (Lb-in/deg)
Accessory Gear Coupling I	U = 0.075	12.75	190	18,000	440	+/- 0.25	1900
Accessory Gear Coupling II	U = 0.25	12.75	190	18,000	1450	+/- 0.25	4800
Accessory Disc Coupling		12.62	220	18,000	1090	+/- 0.25	1000
Accessory Diaphragm Coupling		12.75	250	18,000	500	+/- 0.25	600
Load Gear Coupling I	U = 0.075	16.00	480	534,000	6870	+/- 0.25	56,300
Load Gear Coupling II	U = 0.25	16.00	480	534,000	22880	+/- 0.25	140,800
Disc Load Coupling		18.12	620	534,000	3000	+/- 0.25	16,000
Diaphragm Load		17	580	534,000	4000	+/- 0.25	5,600

Table 1

Coupling Type	OD (in)	Half Weight (lbs)	Center of Gravity (in)	Continuous Torque Condition (Lb-In)	Axial Force at continuous conditions (0.06")(lbs)	Continuous Angular Misalignment (Degrees)	Bending Moment (Lb-in) @ ¼ degree
Reduced Moment Gear	6.06	16.3	1.99	66,800	840	+/- 0.25	3150
Reduced Moment Disc	6.56	18.1	2.31	75,000	166	+/- 0.25	318
Reduced Moment Multiple Diaphragm	6.94	15.8	2.13	76,700	250	+/- 0.25	240
Marine Gear Type	6.06	19.2	0.27	66,800	840	+/- 0.25	3150
Marine Disc Type	7.81	20.3	-0.20	75,000	166	+/- 0.25	318
Marine Single Diaphragm	9.16	25.4	-0.09	118,000	720	+/- 0.25	388

Table 2

For both disc and diaphragm type couplings, it is important to understand how the misalignment requirements intermesh with the torque requirements. The angular misalignment and axial displacement both distort or bend the elements. With each revolution of the coupling the bending from misalignment is reversed or flexed. That bending is the source of the fatigue loading. The coupling manufacturer will help select the coupling so that the effects of the bending are within the coupling capabilities.

Safety factors and Service Factors

Developments in couplings have made the meanings of safety factor and service factor (also application or experience factor) more confusing. Many people use service and safety factors interchangeable. There is an important distinction, however, and understanding the difference is essential to ensure a proper coupling selection and application.

Service factors are used to account for higher torque conditions of the *equipment* to which the coupling is connected. In API 671, a recommended service (or experience) factor of 1.5 times normal torque is applied to take into consideration "off design" conditions. This factor accounts for torque loads which are not normal, but which may be encountered continuously such as low temperature driver output, compressor fouling, or possible vibratory torques. Different service factors are recommended depending on the severity of the application. Is it a smooth running gas turbine driven compressor application or a reciprocating pump application? Also note that service factors should be applied to continuous operating conditions rather than being used to account for starting torques, short circuit conditions, rotor rubs, etc.

Safety factors are used in the *design* of a coupling. Coupling designers use safety factors because there are uncertainties in the design. The designer's method of analysis uses approximations to model the loading and, therefore, the calculated stresses may not be exact. Likewise the material properties such a modulus, ultimate strength and fatigue strength have associated tolerances that must be considered.

Because today's disc and diaphragm couplings have stress loading that is more easily determined, stresses from misalignment, axial displacement and torque are more accurately known than with a gear coupling. Computational tools like finite element analysis (FEA) provide much more accurate results than in the past. In addition, the properties of materials used in high performance products are more controlled and better known. Consequently, couplings designed today versus those designed twenty five years ago can indeed operate reliably with lower calculated safety factors. Generally, torque is the most significant load contributor to the overall stress in gear couplings while the safety fatigue factors of flexible-element couplings are generally not as affected by torque because the failure mode in dry couplings is not very sensitive to torque during continuous operating conditions. It is more often the stresses from misalignment (axial and angular) that affects dry coupling elements. Considering the number of variables that affect gear coupling design such as tooth form, surface finish, materials, temperature and especially lubrication it is difficult to evaluate the "useful" life of these couplings. This is just another reason why gear couplings are rarely used on today's critical equipment

Continuous Operating Conditions and Factors of Safety

The diaphragm, diaphragm pack, or disc pack is the heart of a flexible-element coupling and in general is the most highly stressed component during continuous operation. It must accommodate the constant (steady state, or mean) stresses from axial displacement, torques and centrifugal effects while also withstanding the alternating (cyclic) stresses from angular misalignment and possible alternating torques. Note that normally other components of the coupling such as flanges, tubes and bolts are not subject to the same magnitudes and types of stresses.

To analyze a flexible element and determine it's (and generally the coupling's) application factor of safety at different loading conditions, its endurance limit must be determined. The problem here, though, is what failure criteria should be used to determine this limit. What assumptions are made in combining the stresses? Once criteria are selected, how is the factor of safety determined? What is an appropriate factor of safety for a particular type of coupling? There are many "correct" answers to these questions, and previously the choices were left up to the coupling manufacturers. Now, API 671 has adopted a standard way to calculate factor of Safety.

Let's consider the following load conditions and stresses for a diaphragm coupling in a turbine driven compressor application (Table 3). (Note that the stresses represented below are for illustrative purposes).

CONDITION	AMOUNT	STRESS	STRESS TYPE
Torque	400,000 in-lb	42,000 psi	Constant, shear
Speed	13000 rpm	12,000 psi	Constant, bi-axial
Axial Misalignment	+/- 0.120 in	35,000 psi	Constant, bi-axial
Angular Misalignment	+/- 0.25°	17,000 psi	Alternating, bi-axial

Table 3

To calculate the fatigue factor of safety, there are four basic avenues that must be taken: 1) Determine the basic, *normal stresses* that result from the stated operating conditions with applied service factors. 2) Apply an appropriate *failure theory* to represent the combined state of stress. 3) Apply appropriate *fatigue failure criteria* to establish equivalent mean, and an equivalent cyclic stress from which to compare the material fatigue strength. 4) Calculate the factor of safety by making one of three assumptions regarding the manner in which you are most likely to see a stress increase.

First, the way in which the above stresses in this example were determined is subject to evaluation. Various methods may be employed to determine the normal stresses shown above. These methods include classical solutions, empirical formulas, numerical methods and FEA. The accuracy of each of these methods is largely dependent on the loading assumptions made in the analysis.

Second, after calculating the fluctuating and constant normal stresses, they must be combined to provide an accurate representation of the biaxial state of stress by applying an appropriate *failure theory*. Many theories may be employed. The most accurate choice is generally a function of material characteristics and the type of loading. Among the failure theories that might be employed are: maximum principle stress, maximum shear stress and maximum distortion energy (von Mises).

Third, after an appropriate failure theory has been applied, an equivalent constant, and an equivalent alternating stress, must be determined by applying appropriate *fatigue failure criteria*. The possible choices here include: Soderberg criteria, Goodman criteria, modified Goodman criteria, and constant life fatigue diagrams.

Lastly, a fatigue factor of safety can be determined by comparing the equivalent stress to the fatigue failure strength. To compare the fatigue strength to the equivalent stress, an assumption must be made as to how the stress increase is most likely to occur. Three choices exist: the increase will be constant (torque, speed, axial), cyclic (angle, torsional oscillations), or a combination of constant and cyclic (a proportional increase of all stresses and loads).

API's guideline combines the stresses using the distortion energy *failure theory*, and applied the modified Goodman *fatigue failure criteria* to obtain the combined mean (constant) stress and the cyclic (alternating) stress.

Using these guidelines on a modified Goodman diagram (Figure 36), the constant and alternating stresses are plotted. Where the material used has an endurance strength of 88,000 psi, the yield strength is 165,000 psi and the ultimate material strength is 175,000 psi. To illustrate the effect that the assumed rate of stress increase has on the fatigue factor of safety, we have shown that the factor of safety is found to be 2.59 under cyclic stress increase assumption, 1.61 under constant stress increase assumptions and 1.44 under proportional increase assumptions.

Peak and Maximum Momentary Conditions and Factors of Safety

Like continuous torque ratings, there are different ways to rate a coupling's capability to handle non-continuous peak torques or low frequency high cyclic torques. How much more torque can the coupling withstand above its maximum continuous rating before serious damage occurs? Some couplings have a catalog peak rating in the range of 1.33 to 1.5 times the maximum continuous catalog torque rating, even though the couplings can handle torques of 1.75 to 2.25 times before detrimental damage occurs. Some published ratings are only 10 to 15% away from a yielding limit. Which couplings can handle these peak torques?

The values in the table, adopted by API 671 (Table 4), are recommended as a guide and do not reflect how good a job was done in determining and combining the stresses used to obtain them. A certain level of confidence is required with each coupling manufacturer based on experience with the product and organization.

COUPLING CAPACITY	DESIGN FACTOR OF SAFETY	BASIS
Max. Continuous Rating	1.25 minimum	Endurance
Peak Rating	1.25 minimum	Yield
Max. Momentary Rating	1.00	Yield

Table 4

Rotordynamics

All rotating shafts and rotors deflect during rotation, even in the absence of an external load, due to the fact that a shaft cannot be perfectly balanced, or rotate precisely about its principle inertia axis. There will always be a centrifugal deflecting force. The magnitude of the deflection depends upon the stiffness of the shaft and its supports, the total mass of the shaft and the attached parts, the unbalance of the mass with respect to the axis of rotation, and the amount of damping in the system. The deflection, considered as a function of speed, shows maximum values at so-called critical speeds.

Turbomachinery should be designed to operate well away from these lateral critical speeds, unless the rotors are critically damped. Typically, machines run below the first critical, between the first and second critical, or between the second and third. Both coupling reactive moments and location of the coupling center of mass to the connected rotors affect the connected rotor critical speeds. They lower these speeds. Since machine designers want the machines to run at the highest speed possible just below a critical, it's to their advantage to have the critical speeds be as high as possible. That means, the couplings connecting the equipment must be as light as possible.

Moreover, there are torsional critical speeds that must be considered. These criticals can be particularly dangerous, and must be avoided and/or analyzed for safe operation at them. The reason for the danger is that torsional oscillation monitoring equipment is not normally outfitted on critical machines; only lateral and axial vibration is monitored.

Since the coupling is the torsionally softest component in an equipment train, it has the most dramatic effect on the location of the train torsional critical speed locations (the inverses of the component torsional stiffnesses are used to calculate the overall train stiffness, so the lowest stiffness has the highest effect on the natural frequencies). These resonant frequencies are proportional to the stiffness divided by the inertia. Frequently, couplings are tuned (made stiffer or softer torsionally) to assure an acceptable margin between the machine running speeds and torsional critical speeds.

The portion of the coupling between flexible elements can be considered a power transmission shaft, and will, therefore have its own critical operating speeds. Usually this part of the coupling – called *the floating section* - is tubular, but not completely uniform (Figure 39). Couplings are generally required to run below the first critical speed, therefore, the first critical speed is analyzed and often provided on the drawing.

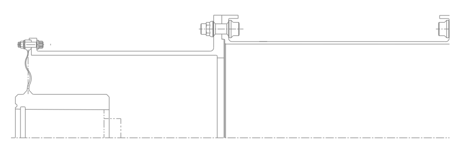


Figure 39

Many gas turbine couplings, especially on mechanical and compressor drives, are connected to the turbine on the exhaust (hot end) of the turbine (Figure 1). So, even more to be considered are high potential temperatures in the coupling tunnel inside the exhaust duct (especially if there's a leak), and, the typically long spacer piece required to clear the duct. Considering all these factors, a modern gas turbine coupling can be quite difficult to design.

Table 5 shows an application where the lateral critical speed of the coupling and the margin of separation from the equipment operating speeds are of concern. The application is a gas turbine driving a centrifugal compressor using a high performance disc coupling (Figure 40). The operating speeds ranged from 7500 rpm to 12200 rpm. Note that depending on the calculation method, the lateral critical speed (LCS) value varied greatly from 15880 to 22200 cpm.

ASSUMED SUPPORT STIFFNESS (LB/IN)	MAXIMUM OPERATING SPEED (RPM)	SST CRITICAL SPEED (CPM)	TOTAL FLOATING WEIGHT CRITICAL (CPM)	RAYLEIGH – RITZ W/ SUPPORT CRITICAL (CPM)	FREQUENCY METHOD CRITICAL (CPM)	FEA ANALYSIS CRITICAL (CPM)
RIGID	12200	22200	16770	-	-	
500,000	12200	-	-	18960	17100	15880
1,000,000	12200	-	-	21500	18870	17280

Table 5

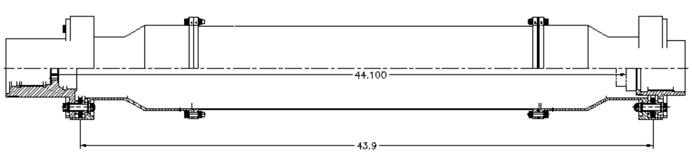


Figure 40

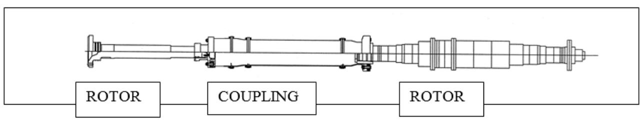


Figure 41

One of the complicating issues is the influence of support stiffness of the floating section of the coupling. In reality, the coupling is considered as part of a rotor system in that it is supported at each end by the equipment shafts (Figure 41). Part of the coupling on each end is *rigidly* attached to the connected equipment shafts, while the floating portion – the part between flexible elements that will have its own critical speed – is *flexibly* attached. The flexing elements must be designed for a low bending stiffness to accommodate machine misalignment while rotating. It will also have a radial (lateral) stiffness. The support stiffness of the coupling's floating section includes the flexing elements of the coupling as well as rotor support bearings, and everything in between. This includes the radial and bending stiffness of the coupling flexing elements, the stiffness of the hub or flange attached to the flexing elements, the equipment shaft back to its support bearing, and the stiffness of the journal bearing, which varies with clearance, oil temperature, oil pressure, and speed.

To most accurately predict the coupling LCS, a complete train lateral analysis (with the driving and driven machine rotors and coupling all connected as one long rotor) is required. However, this is time-consuming and expensive, and requires much coordinated effort between the various equipment suppliers and the coupling supplier. In practice, the critical speed of only the floating portion of the coupling is calculated, and includes certain assumptions for the support stiffness of that section. A margin of safety is then applied to that estimate, usually 1.5 or greater. Depending on the method and assumptions used for the supports, a 1.5 safety factor may not be adequate.

Reviewing Table 5, there are two values each for the Frequency Method, the Finite Element Analysis (FEA), and the Rayleigh-Ritz Method; one value at 0.5 million and one at 1.0 million lb./in support stiffness. The values at 0.5 million lb. /in are significantly lower. In fact, the support stiffness of a typical floating section greatly affects the critical speed (Figure 42). The lower the stiffness, the more is the effect. Suspected low stiffness applications need to be evaluated carefully. Note that for the Total Floating Weight Method, and the Simply Supported Tube (SST) Method, the supports are assumed to be infinitely rigid in practice, so no support stiffness is tabulated.

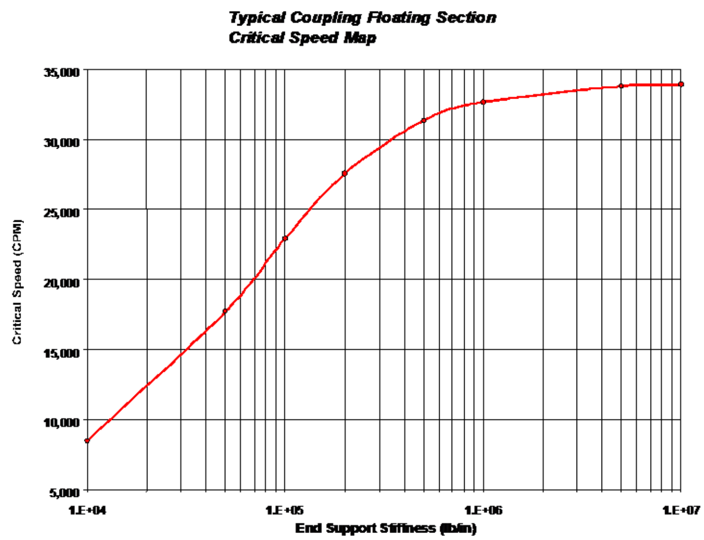


Figure 42

Here are some guidelines to use to insure an adequate margin for coupling LCS to operating speed:

1) *If the supports* (flexible elements, shafts, and machine bearings) *are known to be very stiff* (all with over 1.0 million lb./in stiffness) *the Simply Supported Tube method can be used*, but with a factor (margin) of at least 1.5 times the maximum operating speed

2) *For lower support stiffness, a more accurate calculation is required, or, a higher factor* – 2.0 or greater – *should be used*. In no case should the factor be less than 1.5, unless a full train analysis is done. *The more that is known about the*

support stiffness, and the more accurate the calculation, the lower the factor can be.

3) *If the coupling will run in close proximity to the calculated LCS*, say, less than 1.5 factor, *get the best balance possible, and minimize the shaft and coupling mounting surface runouts*. The exciting force for the LCS will be unbalance due to rotor and coupling runout and inherent coupling unbalance. If the coupling is perfectly balanced and the rotor is turning it exactly about its principal inertia axis, there will be no vibration; but that is not very likely.

4) *Design the coupling with LCS in mind*. Keep large, abrupt changes in diameter to a minimum. Eliminate large overhangs, where the flexing element is well away from the end of the equipment shafts. Keep the weight concentrated near the supports.

These criteria are guidelines since the more that's known about the machines and couplings the closer the margin can be. The geometry and characteristics of connected machines and couplings vary greatly. API 671 4th edition states that the lateral natural frequency of that portion of the coupling between and including the flex elements, assuming infinitely stiff supports, shall be at least two times the highest operating speed for the uniform-tube-equation methodology and at least 1.5 times using more rigorous analysis based on actual geometry (for example FEA). No matter how one calculates the critical speed, increasing the coupling LCS can make the coupling more special - and more expensive - as larger tube diameters, special materials like titanium, etc., may be required to achieve the desired result.

Retrofits

Though dry couplings began to replace gear couplings as far back as the 1970s, there are still high performance gear couplings connecting critical equipment around the world. Over the past two decades most reliability programs included the gear to dry conversion but it is still a topic for discussion.
To best describe the advantages of retrofitting, consider that there are basically two types of flexible couplings; sliding and deforming. Sliding couplings are gear tooth and grid types while disc and diaphragm are examples of those that accommodate misalignment by deforming a flexible element. Sliding couplings wear, require lubrication and therefore maintenance but there are many other advantages to consider when retrofitting. Dry coupling advantages include

- Infinite life - no wearing components
- Lower forces on connected equipment
  20-30% in bending moment, 10% in axial
- Lower repeatable unbalance (no radial clearances)
- Unbalance level does not change over time
- No lock up or spool cracking
- No hub to shaft fretting
- Field interchangeable components
- Less expensive

When properly applied, disc and diaphragm couplings can improve the performance of most trains initially installed with gear type couplings. However, an improperly applied coupling can create many serious problems if consideration is not given to the design differences. Quite often couplings are used to "tune" a multiple bodied train. The rotordynamic response can be greatly influenced by a change in the coupling's mass distribution. To simply retrofit a gear coupling based on price and delivery is a bad idea that can lead to disaster. The question then arises as to who should be involved in the retrofit process. There are two avenues available to the user- the OEM or coupling supplier. The OEM generally offers the greatest advantage but this comes at a cost. If the necessary data is available and a rotor study is not required many users prefer to deal directly with the coupling supplier.

In addition to the basic information required to size a coupling, there are several issues to consider for proper retrofit selection. The most significant coupling characteristics to be study are:

- Weight and center of gravity (Cc) location.
- Torsional stiffness.
- Lateral critical speed.
- Windage/temperature rise.
- Equipment envelope.
- Axial travel capacities.

One of the first considerations should be to match, as closely as possible, the overhung moment of the old coupling (Figure 38). Since membrane couplings are less power dense they tend to be larger in diameter and heavier for the same given torque. Because of this, guidelines should be established to insure successful operation. The following guidelines are sometimes used:

- If the operating speed is less than 3,600 rpm the overhung moment of the new coupling should be within 20 percent of the existing coupling.
- Between 3,600 rpm and 6,000 rpm the weight should fall within 15 percent.
- Over 6,000 rpm the weight should be within 10 percent.

The sensitivity of the machines must be considered when applying these guidelines. If the coupling weight is an appreciable percentage of the rotor weight, very little deviation may be tolerated.
The torsional stiffness should also be as close as possible to the existing coupling. Multiple bodied trains tend to be particularly sensitive to changes in the coupling's torsional stiffness. It is common for end user specifications to require the stiffness to

be within 10%. If either of these criteria cannot be met, rotor dynamic analysis should be performed.

When retrofitting, particularly for reduced moment designs, a basic criteria should include heat generation calculations. Heat generation relates to the coupling's proximity to the equipment casing and guard as a function of speed. Coupling suppliers can estimate the temperature rise in the guard when given housing and guard dimensions. .Bolt windage should also be considered as part of heat generation calculation. There have been cases where dry couplings, due to low pressure created in the housing, have developed a sufficient pressure differential to pull lubricant from the machine's bearings causing it to fail. Another "rule of thumb" is when the back face of the coupling is closer than 40mm from the bearing cage, consideration should be given to a "vacuum breaker" labyrinth. On applications where a 50mm radial clearance exists between the coupling OD and the guard or housing, these calculations are generally not required.

Gear coupling have considerable axial travel while flexible membrane couplings are limited to the fatigue properties of the elements. When retrofitting, it is important to accurately determine the shaft separation, thermal growth of the machines and the direction of that movement. Coupling suppliers often recommend that when the axial thermal growth requirement exceeds 25% of the coupling's capacity, it should be pre-stretched (or compressed). The "pre-stretch" is normally designed into the coupling by changing the spacer length. In most cases dry couplings have sufficient axial travel capacity.

Balance

The subject of coupling balance of turbomachinery and critical pumps is often a topic of discussion because of its influence on machinery vibration.

A coupling can be brought into balance by four basic methods;

A. Tighter manufacturing tolerances
B. Component balancing
C. Assembly balancing
D. Field balancing

Most unbalance comes from machining tolerances and clearance fits of components. The amount of unbalance can be greatly improved by tightening these fits and tolerances. The API standard dictates that couplings operating at speeds greater than 1800 r/min have components centered by means of piloted or rabbeted fits. The fits range from a loose fit of 0.025 mm (0.001 in.) to interference with fits that tighten under centrifugal load preferred. Couplings operating at 1800 rpm or less are only required to have pilots when needed to meet the balance tolerance. Fasteners must be held to dimensional and mass tolerances (typically .1 gram), to allow for interchange within the same set of fasteners or replacement fasteners without affecting the coupling integrity. Fasteners for all piloted flanges must have a diametrical clearance of not greater than 0.13 mm (0.005 in.) in the holes of one flange.

A coupling's balance level is its ability to taken apart and reassembled numerous times without changing the mass eccentricities of the components. To understand API balance methods, it is important to understand the difference between potential and residual unbalance. Residual unbalance is the level of unbalance remaining in a component or assembly after it has been balanced. Potential unbalance is the amount of unbalance that might exist in a complete coupling.

To meet the API-671 Standard, all high performance flexible couplings are required to be balanced. The method of balance is generally determined by the speed of the application and unbalance sensitivity of the connected machines.

Method 1 is to separately balance each major component or factory-assembled sub-assembly. This method is the standard for couplings operating at 1800 rpm or less. (fig.43)

Method 2 is as method 1 but with the addition of a check balance carried out on the completely assembled coupling. This method is the standard for couplings operating above 1800 rpm, with options as specified. (fig.44)

Method 3 is an optional method for couplings operating above 1800 r/min, with options as specified, and is based on the balancing of the completely assembled coupling as an entity. (fig.45)

Method 1- Component Balance

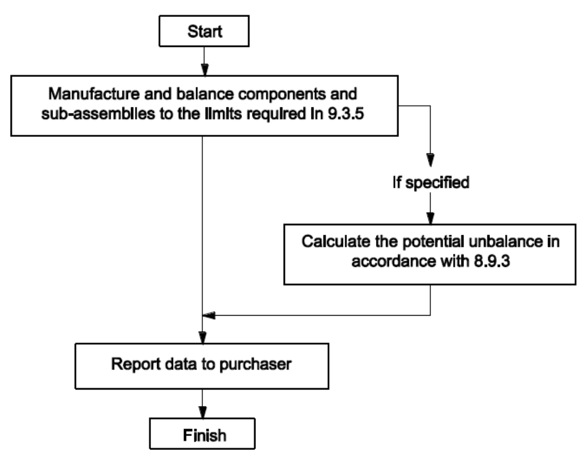


Figure 43

For low speed applications, component balancing is generally the best method. Component balance offers the advantage of interchangeability and can usually produce potential unbalance values equal to those of an assembly balanced coupling. Couplings components are balanced by rotation and each component is balanced individually. All machining of components, except for keyway(s), is to be completed before balancing. Two-plane balancing is required for components or sub-assemblies with a length/diameter ratio greater than or equal to 1.0. Where the length/diameter ratio is less than 1.0, two-plane balancing is preferred but single-plane balancing is acceptable. Each component shall be balanced so that the level of residual unbalance, *U*, expressed in gram-millimeters (ounce-inches), for each balance plane does not exceed the greatest of the values determined by

$$U = \frac{K_2 \cdot m}{N}$$

$$U = K_3 \cdot m$$

$$U = K_4$$

where

K_2 is a constant, equal to 6350 (4);

K_3 is a constant, equal to 1.27 (0.0008);

K_4 is a constant, equal to 7.2 (0.01);

m is the mass, expressed in kilograms (pounds), of the component appc of the balance planes so that the sum of the masses apportioned to total mass of the component;

If specified, the coupling vendor shall perform calculations to verify the potential unbalance of the complete coupling. The potential unbalance limits are determined by the ANSI/AGMA 9000-C90 Standard where the mass of the half coupling is considered to be concentrated and is normally referenced from the equipment shaft end. For couplings operating at speeds of 1800 rpm or less, the potential unbalance cannot exceed the potential mass center displacement of 50 µm (2000) microinches or AGMA Class 9. From 1800 to 5000 rpm, the maximum displacement is 27 µm (1000 microinch) or Class 10 and for couplings operating over 5000 rpm the maximum is 13 µm (500 microinch) or Class 11.

Method 2- Component and Assembly Check Balance

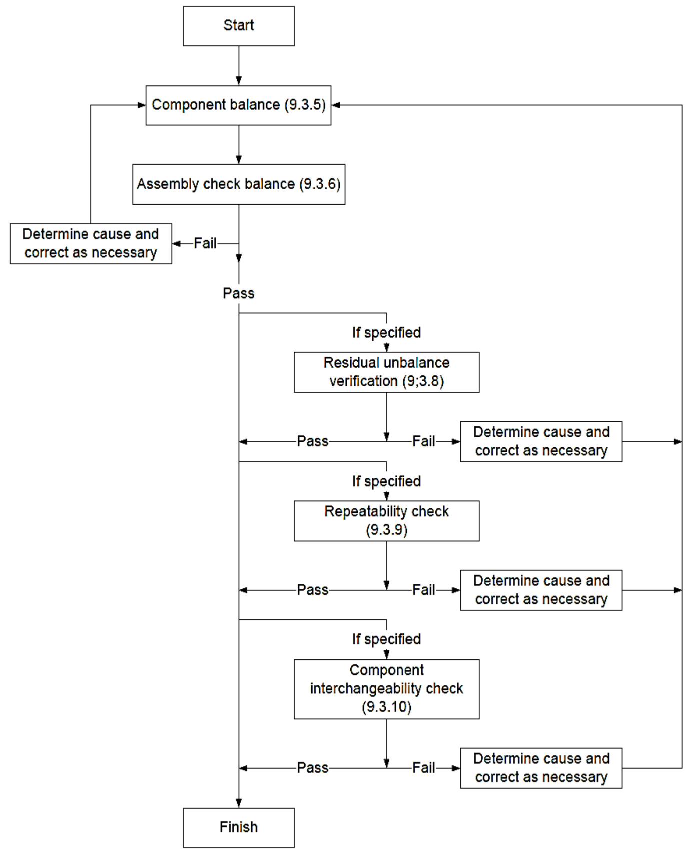


Figure 44

Component balance and assembly check requires that all components be manufactured and component balanced to achieve the required level of potential unbalance. The coupling is then randomly assembled and the balance checked as an assembly within the limits

$$U = \frac{K_5 \cdot m}{N}$$

$$U = K_6 \cdot m$$

$$U = K_7$$

where

K_5 is a constant, equal to 63,500 (40);

K_6 is a constant, equal to 12.7 (0.008);

K_7 is a constant, equal to 72 (0.1);

If the assembled coupling does not meet these tolerances, the components must again be balanced. Note that the maximum unbalance of the assembly is 10 times greater than the limit for components. The stack-up of tolerances, eccentricity of the pilot fits and squareness of the parallel planes dictate these residual limits. Although this method includes matchmarking to insure that the components are in the same orientation in the field, no corrections are made to the assembled coupling. This method also allows for interchangeability of components though meeting these limits is not guaranteed without an interchangability check procedure.

Method 3- Component and Assembly Balance

Component balance with assembly balance offers a lower residual unbalance of the assembled coupling and may provide the best overall coupling balance. For assembly balancing, coupling components or sub-assemblies are balanced to meet component balance limits. The assembled coupling is then match-marked and two-plane balanced with corrections being made only to the component or sub-assembly in each balance plane that was not previously balanced. The final residual unbalance of the assembled coupling in each of the two correction planes shall not exceed the greatest value determined by

$$U = \frac{K_2 \cdot m}{N}$$

$$U = K_3 \cdot m$$

$$U = K_4$$

where

K_2 is a constant, equal to 6350 (4);

K_3 is a constant, equal to 1.27 (0.0008);

K_4 is a constant, equal to 7.2 (0.01);

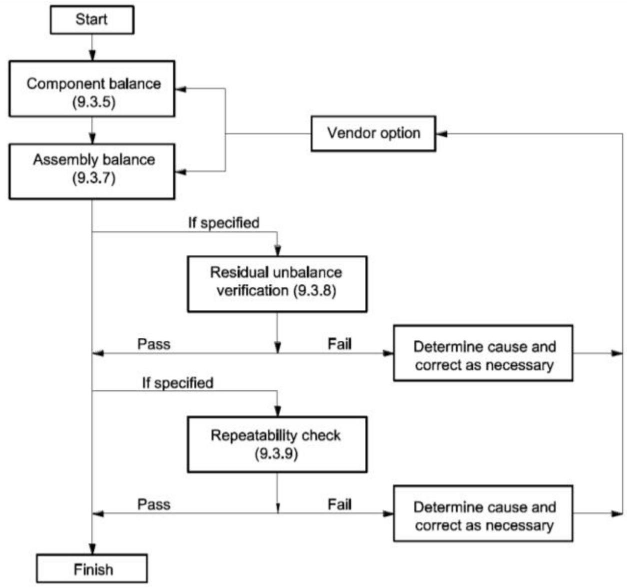


Figure 45

Optional Assembly Check and Assembly Balance Procedures

If specified, a **balance repeatability** check can be performed on assembly check or assembly balanced couplings to insure that the assembly balance can be repeated. The coupling is disassembled to the same extent required for normal field disassembly. The reassembled coupling is then checked on the balancing machine. The residual unbalance measured cannot exceed the assembly check balance limits.

If specified, a **component interchangeability test** can be performed. Where spare components are purchased with the original coupling, following the assembly check balance match-marking procedure, the coupling is disassembled and spare component(s) are used when reassembling. The unbalance of the coupling using field interchangeable components must also meet assembly check balance limits. These spare components are also match-marked to identify proper positioning as applicable.

If specified, a **residual unbalance check** can be performed after assembly balancing is complete and before the assembled coupling is removed from the balancing machine. This check compares the calculation of the residual unbalance with the maximum allowable by using a trial unbalance mass between 1 and 2 times the allowable residual unbalance. Seven tests are performed at 60 degree increments.

Field Balancing

On some high speed and light weight rotor applications it may not be possible to provide a balanced coupling to meet the requirements due to inherent errors introduced when a coupling is on the balance machine.

If specified, threaded holes shall be provided in the coupling for trim balancing. The **trim-balance holes** should be capable of correcting for an unbalance of

$$U = K_6 \cdot m$$

The number, size, depth and location of such holes shall be agreed upon by the purchaser and the vendor. The optimum hole location for keyed hubs is generally on the outboard faces of the hubs, midway between the inside and outside diameters of the hub barrel. The optimum location for keyless (hydraulically fitted) hubs is generally on the coupling flanges, between the bolt holes of the flange.

In addition to the proper selection of flexible couplings, it is important to consider the best balance method and optional procedures to ensure that the influence of coupling unbalance on the equipment is kept to acceptable limits and in many applications the lowest level achievable. Unnecessary and costly procedures are often specified when coupling balance is misunderstood.

Machinery monitoring

Figure 46

Speed pick-ups and key phasor slots can be added to turbine coupling components. Speed pick-ups consist of an external gear ring located on the outside diameter of the hub. The teeth are square in shape and provide a surface for a sensor to read. A key phasor slot is a milled slot in the turbine hub OD. The slot is used along with a sensor mounted to the turbine housing to plot the exact angle of the coupling while rotating. The combination of a key phasor slot and speed pick-up is useful in identifying the location of vibration in the system. Axial probe faces are an additional feature that can be incorporated into the turbine hub. A machined ring on the hub OD allows a proximity probe to measure the axial movement of the coupling/shaft. The axial probe face is machined with an rms 32 finish and help perpendicular to the bore within 0.001 of an inch.

Coupling torquemeter systems provide accurate, real time torque measurement and trending. These systems help to determine a turbocompressor train's performance by measuring the twist or strain in a coupling spacer or other coupling component due to torque. From torque, temperature and speed the machine's power and hence performance can be tracked. Continuous torque monitoring is used to provide machinery train performance for long-term trending, fuel consumption and pollution control monitoring. Torquemeters are used on various test stands application and can be used to identify train torsional critical speeds.

There are two basic types of torquemeters: strain gage and the phase shift. Strain gage torquemeters use strain gages attached to a coupling spacer or other component to measure torque induced strain (Figure 46).

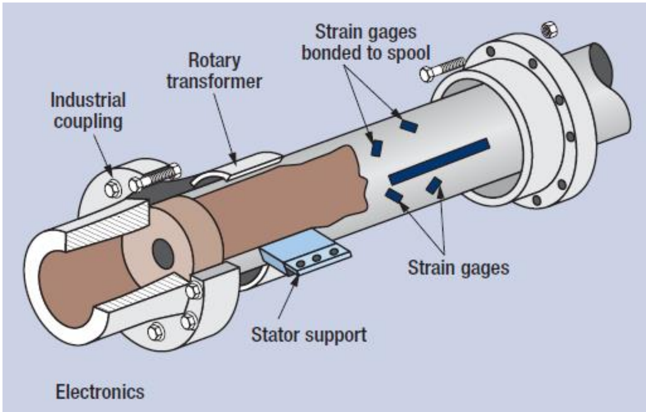


Figure 46

The strain gage requires that power be applied to the gage circuit and an output signal be transmitted to a stationary receiver, typically using FM signals. Electronic components are attached to and rotate with the coupling. A phase shift torquemeter uses stationary probes or pick up coils to measure the torsional deflection of a spacer or component (Figures 47,48 & 49).

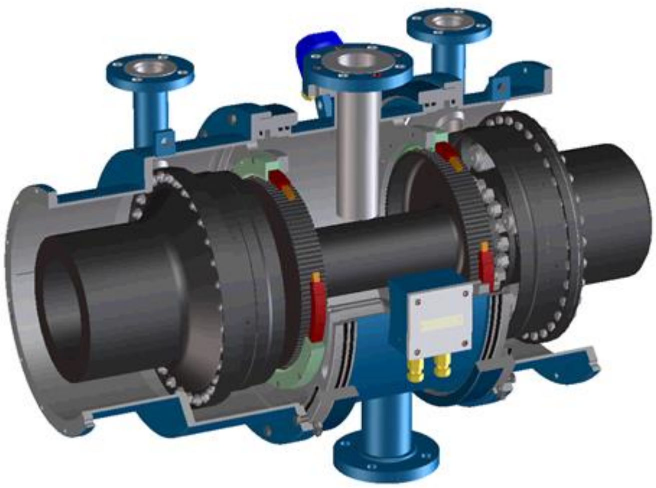


Figure 47

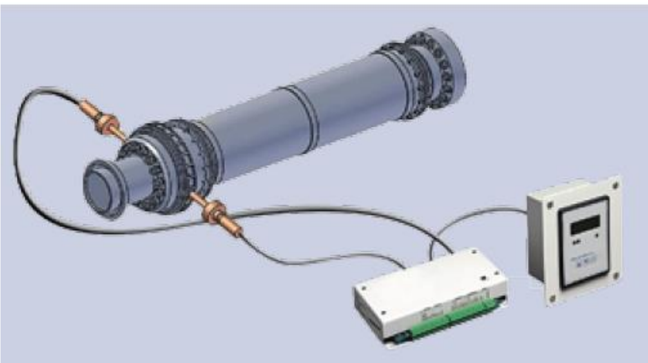
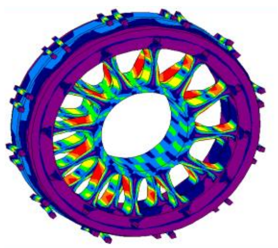


Figure 48 & 49



A pair of signal generating teeth are rigidly attached or machined into a coupling component. The probes or coils measure the relative movement of the teeth and determines deflection directly related to the torque applied. The difference in waveforms between the two generated signals is then processed at a stand-alone display unit or directly to a DCS or PC. The integral temperature measurement in the stator or probes provides temperature compensation of the output for more accurate results.

Both types of torquemeters can achieve less than 1% error when originally installed. The strain gage type is usually less expensive but is often considered more of a lab instrument as it has temperature limitations and often can't stand up to the harsh environments of the application. The phase shift's all metal construction and signal processing away from the equipment make it more reliable and rugged in the long term.

Materials & Coatings

Materials have also advanced. Gas turbine gear couplings were, and are still, made from low alloy steels like AISI 4140 and 4340, and similar carburizing and nitriding grades such as Nitralloy for reducing tooth wear. The diaphragm and disc couplings, however, since so much load is accommodate by metal flexure, required stronger materials without being brittle. These materials have superior grain structure to assure homogeneity and fatigue resistance, and were mainly developed for the aircraft industry. Some are listed below:

- AISI 4340 VAR (Vacuum Arc Remelt)
- Custom 455 Stainless Steel
- 15-5 and 17-4 PH (Precipitation Hardened) Stainless Steel
- AISI 301 Full Hard Stainless Steel
- Inconel 718
- Inconel 625
- Monel
- Titanium 6AL-4V

In addition, to prevent corrosion of non-stainless steel at high temperatures, these materials are sometimes coated with an aluminum ceramic or other sacrificial coating (sometimes used on turbine blades) and advanced epoxies. Moreover, to further enhance fatigue resistance, some diaphragm elements are shot peened with very fine hard steel shot, applied at precise pressures. This puts the surface of the metal in compression, thereby preventing cracks from developing.

Analysis

Analysis methods have also kept up. From simple hand calculations and rudimentary basic computer programs, the coupling designer now has many sophisticated tools available to accurately predict stresses and rotordynamic behavior and design a safe and reliable coupling.

Finite Element Analysis (FEA) and similar numerical computer software is used to accurately predict stresses from torque and misalignment on diaphragm and disc profiles. FEA for rotordynamics has allowed for better lateral critical speed analysis of long gas turbine couplings in the exhaust end of the turbine.

But FEA (Figure 47) is still not an end all. Any FEA model must be checked with classical methods and laboratory testing (strain gaging, etc.) to prove the model. Many engineers believe that an FEA calculation answer is automatically correct. This cannot always be, because of the load and boundary assumptions used to build the FEA models.

Conclusion

So, what's next? As gas turbine technology evolves, so must the associated couplings. Couplings are now looked at as part of the total system of the power transmission train consisting of the drive (gas turbine) couplings and driven equipment rather than an add-on commodity. This trend will continue with higher powered system modeling software. Flexible couplings are now required on many large gas turbine applications. Turbine outputs of over 200MW now challenge the coupling designers and manufacturers.

Advanced materials are being looked at meet these new challenges. Composites are now being used in high speed aerospace shafting and flexible coupling elements. Composites are lightweight (always a requirement at high speeds) and can be wound to get the maximum strength in the load direction, but they add another level of complexity in design.

Electronics and wireless technology are becoming familiar to coupling manufacturers. "Smart couplings" can be designed today to measure torque, speed, temperature and critical speeds, but what if wireless gages built into the coupling could give information about changes in alignment and a coupling's useful life? The years ahead will be exciting for the coupling designer.

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