ADVANCES IN GAS TURBINE COUPLINGS

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ABSTRACT

Over the last 50 years industrial gas turbines have advanced significantly. Horsepower has increased 200 percent (and even higher) within the same envelope. Improvements in efficiencies and reliability (more operating time before overhauls) are being made more often. What took 15 years to change 30 years ago, now, in the last 15 years, happens in five years or less. There are constant demands for lighter weight, lower maintenance designs. Moreover, the operation engineers demand to be able to run six or eight years, and more, before an overhaul, not just three years.

As advancements in the industrial gas turbine have occurred, so have advancements in the couplings that connect them to the driven equipment—a gear, compressor, pump, or generator. Most of the early industrial gas turbines that required flexible couplings pushed the state-of-the-art in gear couplings in the late 60s and early 70s. But because of high vibration at higher speeds, and unexpected and

maintenance downtime of the equipment due to gear coupling technology, there started the industrial use of and rapid advancements in diaphragm and disc couplings for gas turbine use.

In this tutorial is a comparison of the characteristics of the various types of gas turbine couplings: gear, diaphragm, and disc. (Rigid and quill shaft couplings are not covered.) The analysis of characteristics for these aforementioned different types of couplings will show that they do not react in the same way when they see the same torque and misalignment. Discussed will be the reliability of the various types, what kinds of safety margins are required, and what experience factors should be considered for various applications. Also discussed will, of course, be the conversion of gear couplings to disc or diaphragm types. In addition, the level of unbalance that can be achieved, the reactive forces that each type can produce on the equipment, and advances in manufacturing, materials, and design tools are covered.

INTRODUCTION

Gas turbine technology has steadily advanced since its inception and continues to evolve; research is active in producing ever smaller gas turbines for the same power output. Computer design, specifically computational fluid dynamics (CFD) and finite element analysis (FEA) along with material advances, have allowed higher compression ratios and temperatures, more efficient combustion, better cooling of engine parts, and reduced emissions. There have been major advances in seals (dry gas) and bearings (magnetic and foil). State-of-the-art electronics allow for continuous monitoring and operation of these machines for periods lasting years with minimal downtime.

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

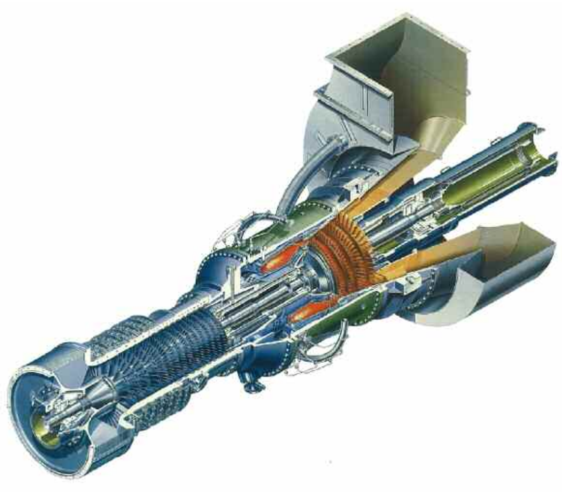

Figure 1. Typical Gas Turbine.

Most gas turbines are special purpose machines that require special purpose couplings. In general, special purpose machines can be identified as expensive, high-powered, and high speed. The horsepower is usually in excess of 1000, and speeds are over 3000 rpm. Usually, for the reason of expense, they are not spared. Another point is that although these machines are high powered, they are also sensitive to certain loads. That is, forces or moments that would seem insignificant to robust steel mill machinery become life threatening to these critical machines. As a result of that sensitivity—and the speed and the power—coupling criteria for the machines are very demanding.

GEAR COUPLINGS

One of the first choices, from a historical perspective, to couple critical equipment was the gear coupling (Figure 2). Some existing critical equipment will still use gear couplings and certainly there are many of them in service right now on critical applications. The gear coupling was chosen for its high power density. It provides more horsepower capability per pound of weight and cubic inch of space than any other coupling. The gear coupling is also very rugged, which means it can take the type of beating that might come from torque spikes or starts and stops. If kept well lubricated, these couplings can be very reliable for many years.

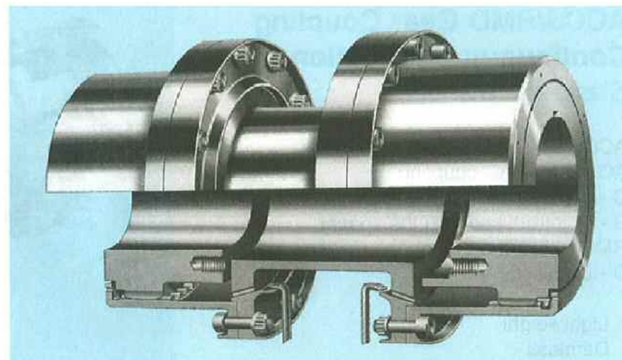

Figure 2. High Performance Gear Coupling.

However, the gear coupling transmits torque and accepts misalignment by the meshing and movement of gear teeth (Figure 3). While that is great for power intensity, the compromise is wear on the mating surfaces. With wear comes the question of when is the part worn out? Can it be predicted? Extended? Known? The measure of reliability is understanding when an event will happen or knowing that it will never happen. With that in mind the gear coupling had to evolve into a very special purpose device when it came to critical applications, but there are still, as will be discussed below, limitations to its use in these critical applications.

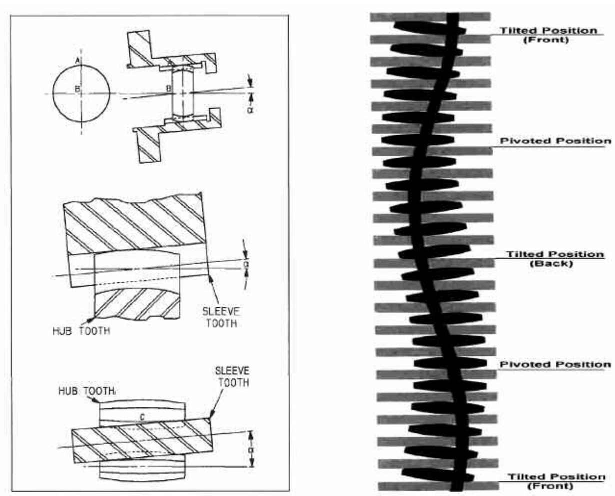

Figure 3. Meshing of Gear Teeth.

The gear coupling as we know it consists of two sets of meshing gear teeth. Each set becomes a flexible point or pivot point. It is known as a double engagement type, and double engagement is a requirement of API 671 (1998) and ISO 10441 (2007), the American and international standards for special purpose couplings on critical applications. The farther apart the two flexible points, the more shaft parallel offset alignment can be accommodated in the device.

In effect each set of gear teeth provides an angular pivot. The amount of permissible angle is a function of backlash and tooth form. The power is transmitted from tooth to tooth at the pitch line. Larger pitch diameters are necessary for higher power transmission. The most common mode of failure for a gear coupling is wear. Due to the number of variables that can affect its successful operation it can be difficult to design and evaluate. Some of the variables affecting its design and characteristics are:

- Tooth design.
    - Straight or the type and amount of crown
    - Pressure angle of tooth
    - Amount of backlash
    - Accuracy of tooth spacing
- Material.
    - Type of material(s)
    - Type of core heat treatment
    - Surface hardening
- Lubrication.
    - Oil
    - Grease
    - Sealed lubrication
    - Continuous lubrication

A gear coupling has its most significant effect not only on itself but on the system components from the forces and moments generated when it slides and/or misaligns to accommodate machine misalignment. When a gear coupling accommodates the shaft float from thermal growth, foundation deflection, etc., axial forces react back onto the thrust bearings and other equipment. When misaligned a gear coupling will produce a bending moment that will load equipment shafts, bearings, and other system components. Both the axial forces and bending moment are significantly affected by the lubrication and the coefficient of friction between the mating members.

Types, Styles, and Applications—
Marine and Reduced Moment

The gear coupling comes in several configurations. These couplings are available in sealed lubrication and continuous lubrication types. There are two basic styles.

A marine style coupling has the flexing elements attached to both ends of a spacer or spool piece (Figure 4). This one-piece center assembly is then mounted to a machine flange or to a shaft mounted rigid coupling hub flange. The idea behind this design is speed and ease of change-out if the flexible elements fail or need to be replaced. The spool piece can be quickly unbolted from the flanges at either end, and a new one installed, and is especially applicable for shipboard (marine) applications where the drive train cannot be down for any significant length of time. The removed center spool can then be repaired at a more convenient time, once the train is up and running.

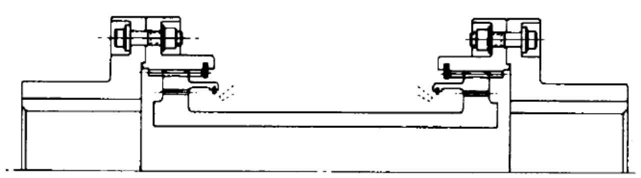

Figure 4. Marine Style Gear Coupling.

A reduced moment style coupling has the flexing elements—gear teeth—mounted on the shaft hubs, with the flexing elements located as close to the equipment bearing as is practical (Figure 5). In this configuration, the effective center of gravity of the coupling is moved closer to the bearing, resulting in a decreased overhung moment. The amount of this moment directly affects the lateral critical speed of the equipment rotor, and therefore affects the sensitivity of the machine to unbalance. The larger the overhung moment, the lower is the rotor critical speed. Since many high performance compressors operate between the first and second critical, too large an overhung moment could place the rotor second critical near running speed, and without adequate damping, vibration problems could result.

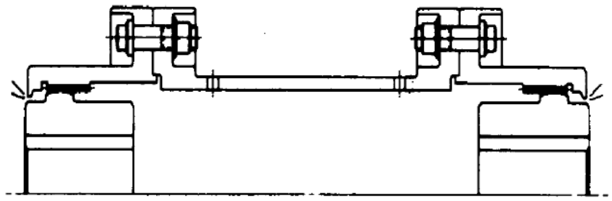

Figure 5. Reduced Moment Gear Coupling.

An example of a high performance gear coupling application is a popular gas turbine. A gear coupling is typically used between this turbine and whatever it drives: gear box, compressor, generator, or pump.

This gas turbine has a 30 year history. During that time, the horsepower of the turbine has almost doubled. The gear coupling for this application has been changed to keep up, through design improvements, without changing its size (Figure 6). Higher strength materials are one change. The geometry of the crowned tooth was improved, for another. The tooth spacing and profile were also improved by lapping the sets together.

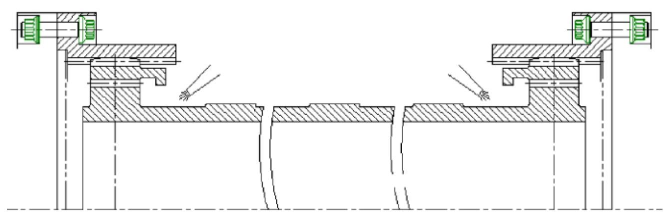

Figure 6. Gas Turbine Gear Coupling.

These gas turbines also use gear couplings for starting and driving the accessories. In that application the couplings are required to take high loads at start up and must accommodate large axial movements (½ inch to 1 inch) while having low axial reactionary loads.

It is important to note that gear couplings can be configured to meet many specification requirements beyond the usual torque and misalignment. The spacer piece can be modified to achieve various torsional stiffness requirements, by changing the diameter and length. The coupling can be built using lightweight strong materials like titanium alloys to reduce weight. When built with the external gear teeth on the spacer piece, a marine style, the replacement of wear pieces is easier. It is not unusual for the coupling to be built with external gear teeth on the outer sleeve to accommodate a slow speed turning device. Gear couplings have been made with attachments such as torque measuring devices.

Design Considerations

Criterion number one under high speed applications and high power considerations is to improve the life of the coupling torque transmission surfaces. For gear couplings that is the gear teeth. Tooth shape and involute angle have evolved into the optimal 20 degree tooth angle. The geometry of the crowned tooth has improved, as has the tooth spacing and profile by lapping the sets together.

Also required are the best, most wear-resistant materials. That choice means strength and hardness. Hard smooth surfaces also reduce friction, but lubrication is important too. Alloy steel that can be hardened to Rockwell C 45 minimum (required by API 671, 1998) or even higher is sometimes required. This is usually accomplished by nitriding the tooth surface. The harder the surface is made, the longer the life of the tooth. Under that hardness it is necessary to have a durable material (high tensile strength).

After the hardening, important is the lubrication of the mating or sliding parts. Low friction means less heat buildup at the surface and therefore less chance for the surface to breakdown or weld together. API 671 (1998) shows a preference for continuous lubrication (Figures 2 and 7). That is accomplished by drawing the lubricant from the bearing lubrication system. The circulating oil lubricant is filtered and cooled. Even with the filtered circulating oil, sludge can still build up in the coupling. At the teeth, where the lubricant is needed, the centrifugal forces are large enough to cause solids in the oil to separate from liquids. Those solids can build up in the cavities and eventually block the oil passage or the movement of the teeth within the mesh, the same movement needed for the coupling to accommodate misalignment.

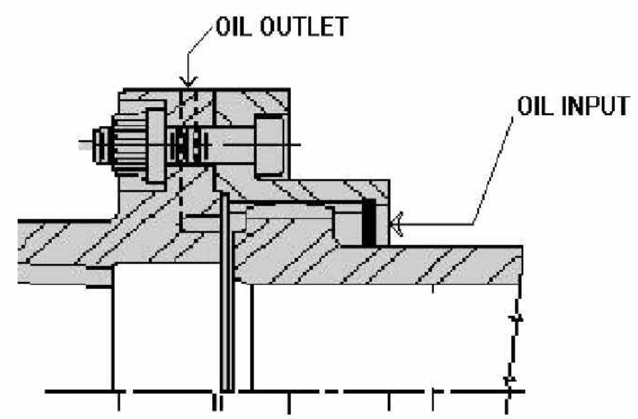

Figure 7. Continuous Oil Lubrication.

Balancing of the gear coupling involves some special considerations. In order for the gear coupling to work as it was intended there must be some looseness to the assembly. That looseness is the backlash in the teeth and the major diameter fit allowance. That looseness means the coupling assembly cannot be as well balanced as a dry type, which typically has no clearances. The gear coupling has a major diameter fit on the teeth, to locate the center section or sleeve and spacer, but the coupling centers on the pitch line when operational. In effect the sleeve, or internal teeth, are changing position. That would change the center of gravity versus the center of rotation, thereby causing unbalance.

As with all balanced couplings, the process starts with the making of the components to very tight tolerances on concentricity and squareness. That is not always easy when part of the component is a clearance gear tooth. Component balance is common, but to achieve assembly balance or do an assembly check balance the gear hub is made special so that the fit at the major diameter is tight rather than loose for the balance operation. After balance, the coupling is disassembled and the tips are relieved (reduced) to allow for flexure and assembly. Size to size fit gear coupling teeth will limit the misalignment capability, if allowed to remain.

The most common gear coupling failures involve a lubrication problem. As mentioned before, foreign materials such as dust and metal particles can mix with the intended lubricant, or worse, separate from the lubricant and centrifuge out to the tooth area, and form sludge (Figure 8). In the extreme case this sludge can lockup the coupling and prevent the movement necessary to accommodate misalignment.

Figure 8. Sludge in Gear Teeth.

Another lubrication problem is an inadequate supply. This could lead to heavy pitting and spalling (Figure 9) and/or excessive wear (Figure 10). Excessive misalignment will also lead to heavy spalling, also called worm-tracking.

Figure 9. Heavy Pitting and Spalling.

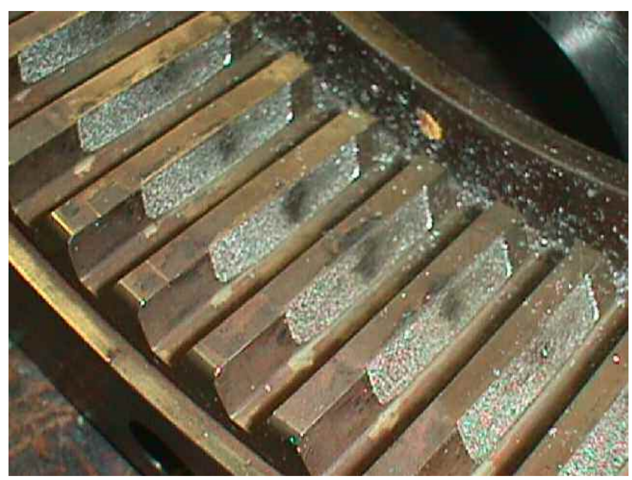

Figure 10. Excessive Wear.

Under extremely high misalignment, tremendous forces are transmitted to the connected shafts and bearings through the couplings. This is especially true for a gear coupling, which has up to 35 times the bending moment under misaligned conditions compared to a metallic flexible element coupling. Serious damage can result if the situation is not rectified (Figure 11).

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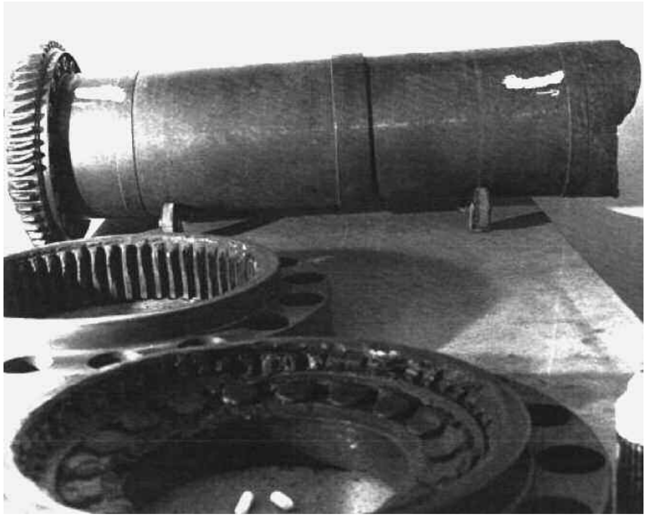

Figure 11. Broken Gear Coupling.

The work done to improve gear couplings was done to keep using a device that was rugged, inexpensive, and space saving. It is not an infinite life device, and it is not a maintenance-free device as it does require lubrication. In the final consideration even though the life has been extended to reasonable values and the lubrication is already available on the machinery, the device is being replaced by a more expensive metallic flexible element (disc and diaphragm) coupling that have infinite life and therefore more reliability. The cost tradeoff eventually favored the higher first cost flexible element device that can be installed and forgotten.

Change to, and Reasons for, Gear to Dry

Prior to the 1970s gear couplings had been used on most gas turbine applications, but as gas turbine development pushed for higher horsepower, speeds, and operating temperatures, problems with gear coupling soon were exposed as they were unable to accommodate the high angular misalignments without excessive reaction loads, and the more stringent balance requirements. These new applications required lower moments, forces, and noise production. The coupling industry responded with the development of metallic membrane couplings that could accommodate these increased requirements.

Now because of the need, 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 technically and/or commercially 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

Diaphragm couplings accommodate flexure from the metal between the coupling's outside diameter (OD) and inner diameter (ID). 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 12). A contoured diaphragm coupling typically uses a single diaphragm "plate" for the flexible member; the plate has a contoured or a wavy profile, which usually has a variable thickness from OD to ID to provide an optimum stress condition.

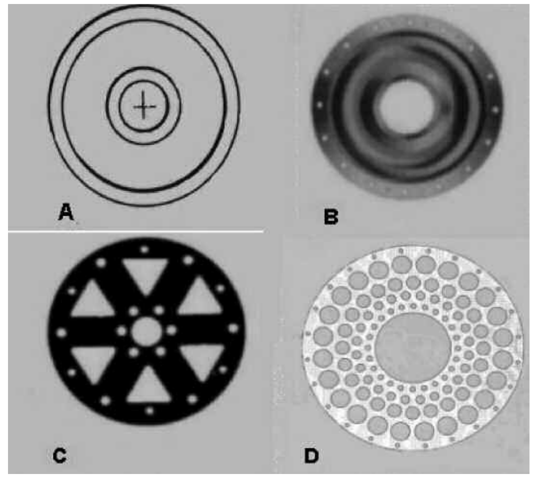

Figure 12. Diaphragm Profiles.

A convoluted and flat-profile diaphragm coupling typically uses multiple diaphragm "plates" that have a "wavy" profile or other modified profile. All types of diaphragm couplings attach the flexible member to other components with bolts, splines, or welds, and both transmit torque in the same manner, in that the torque transmission path through the diaphragm members is in the radial direction; from the outer diameter to the inner diameter, or vice-versa. Load from operating torque is seen as a shear stress on the diaphragm member(s).

Each diaphragm can be deformed much like an automobile axle rubber boot. This deflection of the outer diameter relative to the inner diameter is what occurs when the diaphragm is subject to angular and axial misalignment. Angular misalignment twists the outer diameter, relative to the inner diameter, and produces a complex shape on the diaphragm where it must stretch one way at one point and then stretch the other way at 180 degrees. In between these points, the diaphragm is subject to a combination of stretching and twisting. Axial displacement attempts to stretch the diaphragm, which results in a combination of elongation and bending of the diaphragm profile.

Goodrich's (the former Bendix Fluid Power Division) first patent of the single element contoured diaphragm coupling (Figure 13) was in 1949 and after years of research and development the first diaphragm coupling was delivered for an aircraft application in 1955. This aerospace proven technology yielded a most reliable and lightweight approach to transferring torque and misalignment. In 1967 Goodrich supplied the first contoured diaphragm coupling for use in the industrial petrochemical market and has supplied well over 120,000 contoured diaphragm couplings over the past four decades.

Figure 13. Single Element Contoured Diaphragm Coupling.

The contoured diaphragm coupling has as its flexible element a thin profiled diaphragm machined from a solid disc of heat-treated alloy. This 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 14). The purpose of contouring the profile is to keep the diaphragm as thin as possible consistent with the transmitted torque. This keeps the misalignment bending and axial bending stresses as low as possible for a given torque capacity.

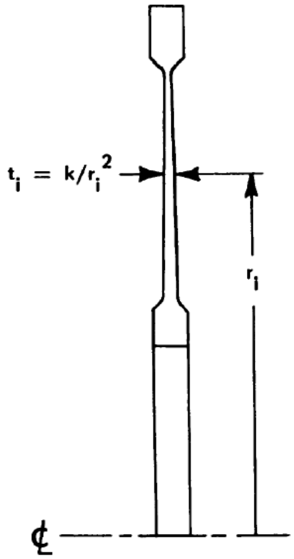

Figure 14. Contoured Profile.

The thickness of a diaphragm can be changed to permit a trade off between torque capacity and flexibility. A thicker diaphragm has greater torque capacity, but is not as flexible, and vice versa. Smooth fillet junctions are provided between the flexing portion and the rigid integral rims and hubs, which connect to the rest of the coupling, to reduce stress concentration.

In 1971, Zurn's Mechanical Drives division developed a multiple diaphragm design, with a number of thin plates in parallel, rather than a single thick one. This type of design provides improved flexibility and lower stresses, as the stresses are proportional to the cube of the material thickness. Additionally, the use of multiple separated diaphragms provides the built in fail-safe feature of redundancy.

For diaphragm couplings, the bending moments and axial forces are more predictable than the previous gear coupling designs since they can be accurately determined based on the stiffness of the flexible elements, as opposed to gear couplings that are dependent on the coefficient of friction that varies with tooth design, type, and quantity of lube, types of material, and tooth finish. As stated before, gear couplings can impose up to 35 times higher bending moments and four to five times axial forces than the equivalent diaphragm coupling. Based on these improved characteristics, the successful introduction of diaphragm technology was accepted within the industry for gas turbine applications.

Building on the experience with the multiple convoluted diaphragm couplings in high performance gas turbine applications, Zurn (now Ameridrives Couplings) developed a new design in the early 1990s, the integral filler. In the original design, the flexible members, the diaphragms, are separated to help prevent fatigue failure by fretting. The diaphragms were of uniform thickness, separated by filler rings on the inside diameter and segmented rings on the outside diameter. After assembly on a fixture, the completed pack was riveted and clamped, or electron beam (EB) welded at the inside diameter and bolted together at the outside diameter (Figure 15).

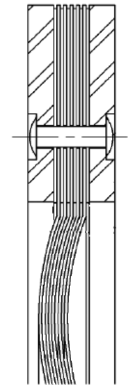

Figure 15. Multiple Convoluted Diaphragms.

The newer integral filler design (Figure 16) optimizes the shape, thickness, and number of diaphragms. Machining a thicker diaphragm material, through the flex area only, eliminates the separate filler rings and segments. This reduces the number of components and manufacturing process. The customization of the diaphragm material thickness, convolution depths, and number of diaphragms used in the assembled flexible element allows for varying torque capacity, bending moment, axial stiffness. This allows for the optimization and tuning of the coupling design when working in partnership with the gas turbine original equipment manufacturer (OEM).

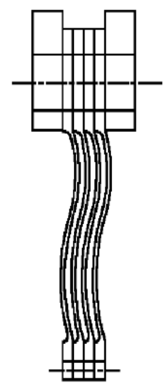

Figure 16. Integral Filler Design.

Kop-Flex introduced the single element flanged design in 1990 (Figure 17). Rather than having the diaphragm welded to a spacer tube or other component (Figure 18), the diaphragm is bolted to a connected component. Whereas one design minimizes the number of mechanical connections in the coupling, the other allows for flex element replacement.

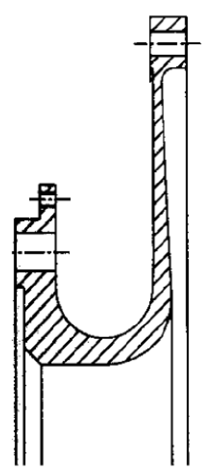

Figure 17. Single Element Flanged Diaphragm.

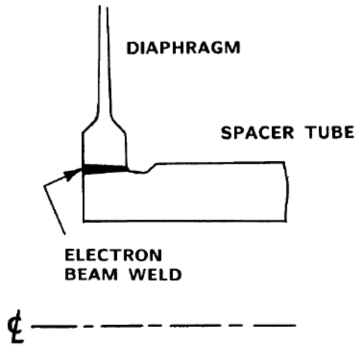

Figure 18. Welded Diaphragm.

An interesting development in the last 15 years with many of the diaphragm types is that the profiles of the diaphragms are being optimized to fit (accommodate) the misalignment and torque requirements of a single application. One application may need more torque capacity, while another needs to accommodate high axial growth and/or angular misalignment. This optimization is possible because the stresses on these diaphragms are so well known now, and the tools are available to do it (FEA type software and modern machine tools).

Diaphragms are made of high strength materials. Some are corrosion resistance (15-5/17-4 PH), others use high quality 4300 steel or other alloys and coat the diaphragms for corrosion protection. Some diaphragm couplings are shot-peened to reduce the residual stresses that are imposed during the manufacturing process, and to prevent the development of surface crack initiation points.

The most often used diaphragm style is the Marine style coupling configuration, which allows the mounting hub bore to vary considerably without affecting the diaphragm diameter (Figures 19, 20, and 21). Thus, the size of the coupling is being chosen to fit the torque and misalignment requirements rather than be dictated by the connected machine shaft size.

Figure 19. Marine Style Diaphragm Single Element Flanged (Photo).

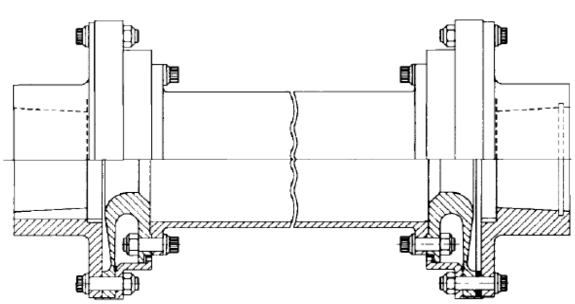

Figure 20. Marine Style Diaphragm Single Element Flanged (Drawing).

Figure 21. Marine Style Diaphragm Multiple Convoluted.

Most diaphragm couplings have a guard to protect the diaphragm or diaphragm pack from scratches and nicks that would act as stress risers on the diaphragms. These guards also act as antiflail devices, to keep the center section contained in the unlikely event of a diaphragm failure.

Furthermore, the guards (Figure 22) can act as pilots to locate the center section to meet API 671 (1998) and enhance the balance repeatability of the coupling. This makes using the technique of only balancing the coupling components (for component interchangeability) more practical.

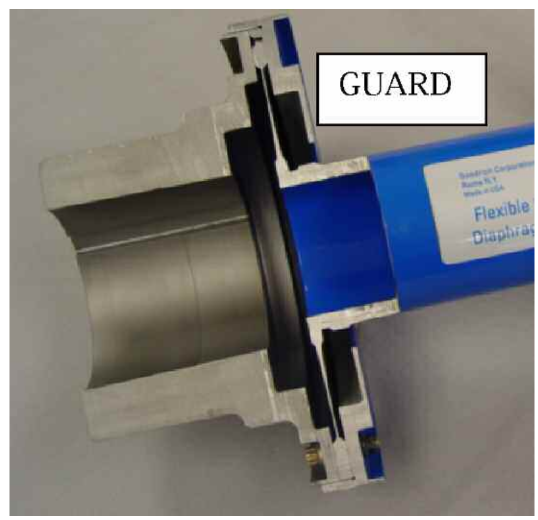

Figure 22. Diaphragm Guard.

For those special applications requiring a reduced moment coupling, the contoured diaphragm coupling is made with the diaphragms machined from forgings with integral hubs (Figure 23), or the diaphragm is reduced in ratio and inverted so it fits over the hub (Figure 24). This configuration shifts the flexible center closer to the machinery bearings to reduce the overhung moment from the weight by moving the coupling center of gravity (CG) toward the machine bearings.

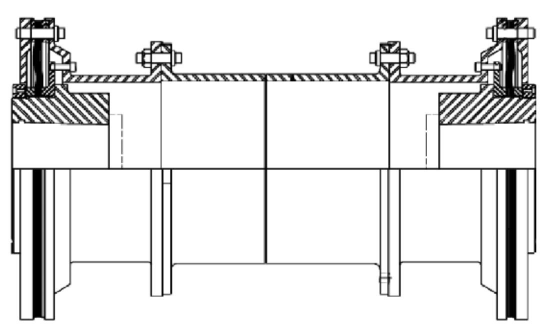

Figure 23. Reduced Moment Diaphragm (Drawing).

Figure 24. Reduced Moment Diaphragm (Photo).

Like the disc metallic flexible element coupling, the diaphragm type will generally fail from either overmisalignment or overtorque. 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 25). Finally, torque overload will cause one or more ripples in the diaphragm (Figure 26).

Figure 25. Failure From Angular Misalignment.

Figure 26. Failure From Torque Overload.

Large gas and steam turbines 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 very large amounts of axial travel. Some single diaphragms can accommodate ± 1 inch of misalignment.

Other variations include a double diaphragm, used for high axial misalignment along with high torque loads, and "J" diaphragms, which have two different profiles per end (Figure 27). There are also, of course, the multiple wavy and multiple flat configurations, which also work well in gas turbine applications.

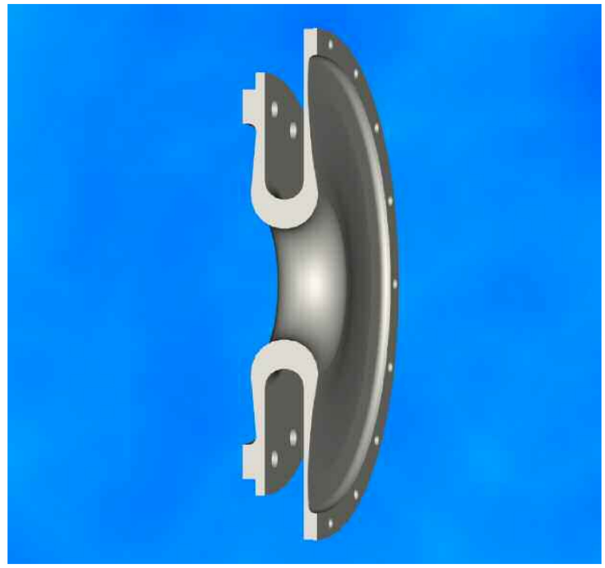

Figure 27. J Diaphragm.

DISC COUPLINGS

In most applications, a well-designed high performance coupling will do the job no matter which type or style it is. As long as it meets the torque and misalignment requirements, and the weight and any other mass elastic characteristic limitations, it will operate well, as long as it is not operated outside of its stated limits. Cost and delivery then become significant factors. However, there are some cases where one type of coupling or the other is typically well suited.

Disc couplings accommodate misalignment by flexure from the metal between adjacent bolts—the flex elements—that are attached to opposite flanges (Figure 28). Optimizing the flex elements produces drastically different capacities and characteristics in diaphragm and disc couplings of the same OD.

Figure 28. Disc Pack Connection.

The multidisc reduced moment design is ideal for gas turbine applications with output shafts where moment reduction is crucial to eliminate potential lateral system problems (Figure 29). Since the torque is transmitted circumferentially from bolt to bolt through the discs, the discs can slide over the hub and shaft end to where the disc pack is closer to the equipment bearings. It 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, because it will travel at less surface speed, and therefore heat up and/or shear less air in the coupling guard.

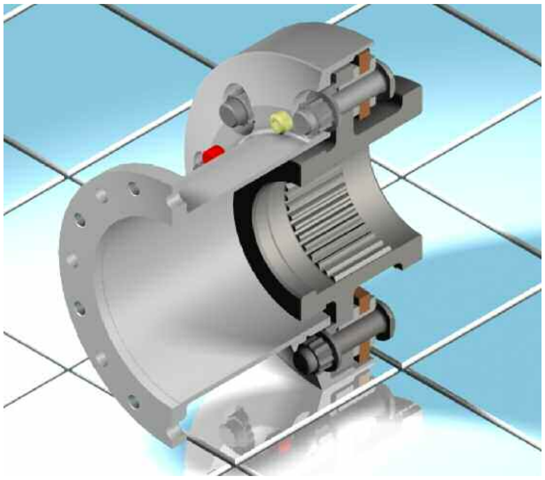

Figure 29. Reduced Moment Gas Turbine Disc Coupling.

The element 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 (Figure 30), one of the benefits of the design is multiplicity. If one or more discs fail, the rest can still carry the load until the equipment is shut down.

Figure 30. Disc Pack.

The thin element has both tensile stresses and cyclic bending stresses imposed on the element so that a fatigue life analysis is required. Once the fatigue life is determined, an infinite life coupling can be designed to keep loading below that fatigue limit. It took some time to develop high fatigue strength materials. Also, advances in material load analysis, such as finite element analysis, were necessary to speed the development of reliable couplings.

The modern high performance disc coupling is nonlubricated and designed for infinite life. The amount of misalignment available is a function of the bending that can be accommodated while under tension. The unit will allow axial movement such as seen with thermal growth. However, that is not unlimited. Also, axial capability and angular capability are interrelated.

An important feature of disc couplings, and couplings that work in a similar manner, is the low reactionary load that is transmitted to the machinery that it couples, compared to the relatively high load of gear couplings. Disc couplings can be more expensive in first cost compared to gear couplings.

There are circular, straight sided, and scalloped profiles of the discs (Figure 31). 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. The other shapes keep the force lines within the boundaries, so the element has mainly tensile stresses. Since the fatigue problem occurs at the bending location the wider cross section is next to the bolts where it does the most good.

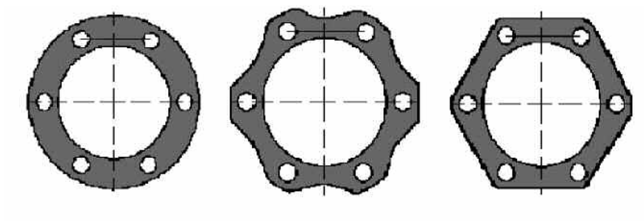

Figure 31. Disc Shapes.

Just like gear and diaphragm couplings, there are three basic configurations for a disc coupling: close-coupled, marine style, and reduced moment style (Figures 32 and 33).

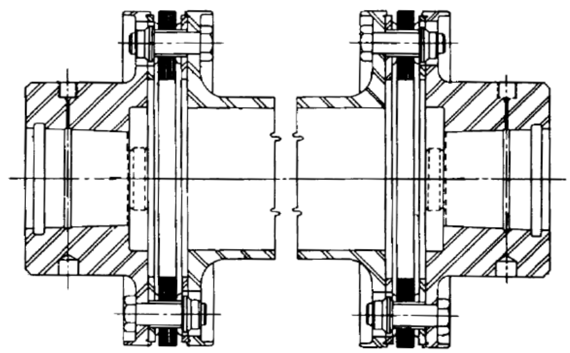

Figure 32. Marine Style Disc Coupling.

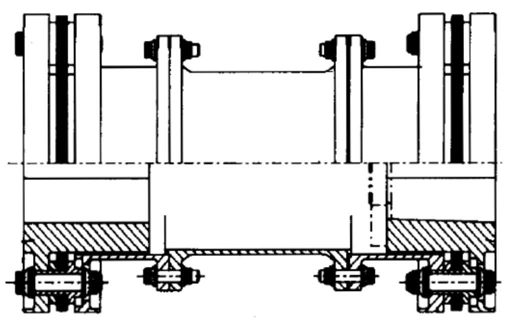

Figure 33. Reduced Moment Style Disc Coupling.

As is the case for other types of couplings, additional considerations are needed for the special purpose application. In order to save space and reduce cost the coupling designer will select disc packs to closely match the application. In some cases a disc will be designed specifically for the equipment. That design will involve the bolt circle diameter, number of bolts, size of bolts, and the number of discs needed. Once the disc pack unit is designed and built at the factory, the pieces should not be disassembled. Piloted disc packs or factory assembled disc packs help to ensure against fatigue failure.

Designing for strength is a function of the disc pack materials and the shape of the disc at critical points such as the bolt attachments. The high performance discs are made from typically cold rolled stainless steel (generally 300 series). Special discs are made of Monel®, Inconel®, PH stainless, and other special materials. Sometimes discs are coated to minimize or even eliminate the effects of fretting at high angles. Corrosion, if it is a factor, is controlled by material selection. Bending, which comes from the misalignment, is controlled by geometry, individual disc thickness, overall disc pack stiffness, the number of bolts, and the fatigue strength of the design.

Flexing metallic element couplings generally fail in either of two basic causes: overmisalignment (more than 80 percent) or overtorque. Overmisalignment generally means excessive angular or parallel offset misalignment, with or without excessive axial misalignment. There are, of course, combination failures, misalignment and torque, but there is usually only one that is primary.

An angular misalignment 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 fatigue.

As mentioned before, one of the benefits of multiple disc pack couplings is multiplicity. If one or a few discs break, the others can still carry the load, at least for a short period of time, depending on the magnitude of the load. In a disc pack coupling, the outer discs, the ones farthest from the center of the pack, experience the highest stress from angular misalignment, as they are the farthest from the center of bending.

So, if an outer disc breaks, the load is redistributed to the inner discs, which then might have a higher torque load, but a lesser misalignment load. 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.

Note in Figures 34 and 35 that the outer discs have failed, from excessive misalignment, but the inner discs are still intact. The connected machines were still operating, though with higher vibration levels, and were safely shut down.

Figure 34. Excessive Angular Misalignment Failure (A).

Figure 35. Excessive Angular Misalignment Failure (B).

The other major cause of failure is torque overload. In the case of a torque overload—for example a compressor ingesting a liquid slug, or a generator short circuit, etc.—the major metal parts of these type couplings will yield. They will then break (albeit at a much higher torque level) if the load is large enough, or if the coupling is allowed to continue operating in the yielded condition.

In Figure 36, 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), made more clear when the failed pack was disassembled. Note that this photo was taken with the equipment shut down, and no load on the coupling.

Figure 36. Torque Overload.

Under load, with a strobe, one could see a similar condition on the unloaded links in the coupling. Some gaps in the links are entirely normal for many types of disc couplings. The key to determine an overload is the condition of the disc packs with no load, or, obvious damaged or bent discs and bolts under a strobe light. It goes without saying that extreme caution should be used when investigating a rotating coupling under a strobe light. It is not recommended without adequate personnel protection. Please consult your coupling vendor about what to look for.

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.

The coupling manufacturer can also provide various charts to show the coupling capabilities. Those capabilities can include the relationship between parallel offset and/or angular coupling misalignment and axial misalignment (Figure 37). Other capabilities and restrictions would include the axial thrust versus axial displacement. Each of these items is needed to be sure the right size and type of coupling are selected and to be sure the designers and operators of the equipment train are aware of the coupling capabilities and limits.

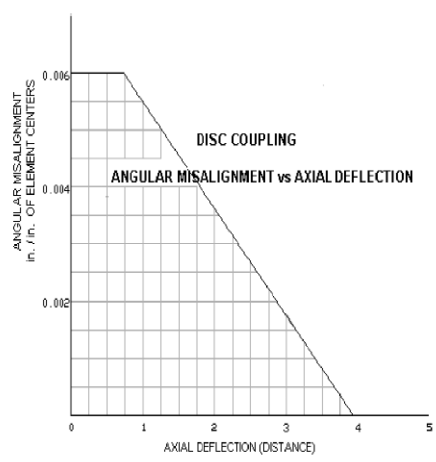

Figure 37. Axial Versus Angular Capacity.

In Table 1 is a comparison of the coupling characteristics for the load and accessory drive of a popular gas turbine. Two different gear tooth designs are compared with dry couplings (disc and diaphragm). As explained before, the diameters of the gear couplings are less than the flexible metallic element disc or diaphragm couplings, as are the corresponding weights. For both the accessory and the load couplings, the bending moment for the gear couplings is larger than the dry couplings. This is especially true for the load application.

Table 1. Load and Accessory Coupling Characteristics.

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

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 startup 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 couplings.

In Table 2, note the improved centers of gravities of the reduced moment couplings compared to marine styles. The centers of gravity are from the end of the shaft toward the equipment bearing. Some vendors call this minus, others plus, so be careful when comparing coupling offerings. Also note the decreased reaction forces of the dry couplings.

Table 2. Reduced Moment Versus Marine Style Characteristics.

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

A similar caution applies to the use of these tables; a good coupling designer can design couplings of lower than catalog weight with higher torque, different center of gravity, etc. In these tables are typical catalog values for a specific design. If the torque requirement was slightly less or more, very different selections could have been made. For example if the torque requirement was 10 percent less in this application, the marine single diaphragm could have been a smaller size.

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 interchangeably. There is an important distinction, however, and understanding the difference is essential to ensure a proper coupling selection for a particular application.

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.

Today, with the use of such computational tools as finite element analysis, stress analysis is generally capable of more accurate results than in the past. In addition, the properties of the materials used in high performance products are more controlled and better known. Therefore, couplings designed today versus those designed 25 years ago can indeed operate safely with lower calculated safety factors. Also, the design factor for flexible-element couplings can be lower than gear couplings simply because the "safeness" is more accurately predicted.

That is because the flexible-element (dry) couplings of today have stress loading that is more easily determined, so the stresses from misalignment, axial displacement, and torque are generally more accurately know than with a gear coupling. Because of the number of variables that affect their design (such as tooth form, surface finish, materials, temperature, and especially lubrication), "life" and "safeness" are difficult to evaluate for gear type couplings.

Generally, torque is the most significant load contributor to the overall stress picture in gear couplings. However, 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 the stresses from misalignment (axial and angular) that affects dry coupling elements more.

Service factors, on the other hand, are used to account for the higher operating torque conditions of the *equipment* to which the coupling is connected. In API 671 (1998), a service (or experience) factor is applied to the normal operating torque of, for instance, a turbine or compressor. This factor accounts for torque loads that are not normal, but that may be encountered continuously such as low temperature driver output, compressor fouling, or possible vibratory torques. Also, service factors are sometimes used to account for the real operating conditions, which may be 5 to 20 percent above the equipment rating.

Different service factors are used or 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.

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 its (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 (1998) has adopted a standard way to calculate factor of safety.

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.)

Table 3. Stresses on a Diaphragm Coupling.

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

To calculate the fatigue factor of safety, there are four basic avenues that must be taken:

- Determine the basic, *normal stresses* that result from the stated operating conditions with applied service factors.
- Apply an appropriate *failure theory* to represent the combined state of stress.
- Apply appropriate *fatigue failure criteria* to establish equivalent mean, and an equivalent cyclic stress from which to compare the material fatigue strength.
- Calculate the factor of safety by making one of three assumptions regarding the manner in which one is 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.

Last, 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 is to combine 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 38), 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, the authors 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.

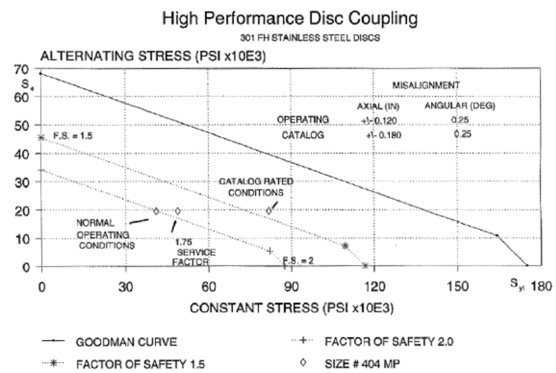

Figure 38. Modified Goodman Diagram.

PEAK AND MAXIMUM MOMENTARY CONDITIONS AND FACTORS OF SAFETY

Just like continuous torque ratings, there are different ways to rate a coupling's capability to handle noncontinuous peak torques or low frequency high cyclic torques. In a gas turbine application, these can be caused by such things as starter motor startup, short circuit conditions, driven compressor surges, or other transient conditions, or even unaccounted for high cyclic torque oscillations.

As for the coupling, how much capacity above the maximum continuous rating is there before serious damage to the coupling 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 percent away from a yielding limit. Which couplings can handle these peak torques?

Finally, the values in the table, adopted by API 671 (1998) (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.

Table 4. Peak Load Factors.

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

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. As a result 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 is 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 is 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 are 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.

Plus, 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 almost always run below the first critical speed, so that is the one analyzed.

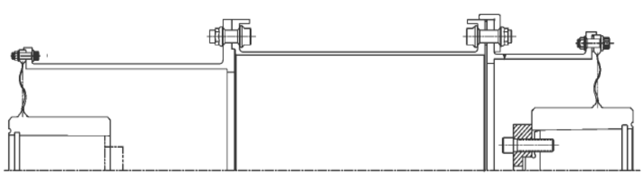

Figure 39. Floating Section Between Flex Elements.

So, a coupling designed for a high speed gas turbine application must be lightweight, laterally stiff, and have the proper torsional stiffness, in addition to being able to transmit the peak and continuous torque loads and accommodate misalignment, sometimes at ¾ degree angular, and ± 1 inch or more.

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 is 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 is for an application where the critical speed of the coupling, and the margin of separation from the equipment operating speeds, became the critical issue. The application involves a gas turbine connected to a centrifugal compressor using a high performance disc coupling (Figure 40). The operating speeds ranged from 7500 rpm to 12,200 rpm. Note that depending on the calculation method, the lateral critical speed (LCS) value varied from 15,880 to 22,200 cpm.

Table 5. Critical Speed Issue Data.

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

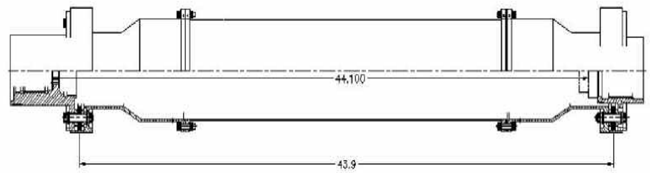

Figure 40. Gas Turbine Coupling with Critical Speed Issue.

The lowest value is 30 percent away from maximum operating speed (1.30 safety factor), and the highest value (22,200 cpm) is over 80 percent away (1.82 factor). These results vary greatly.

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.

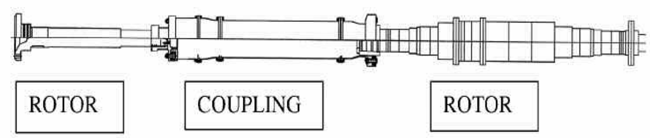

Figure 41. Rotor System.

So, the support stiffness of the floating section of the coupling 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, 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 the effect. Suspected low stiffness applications need to be evaluated carefully.

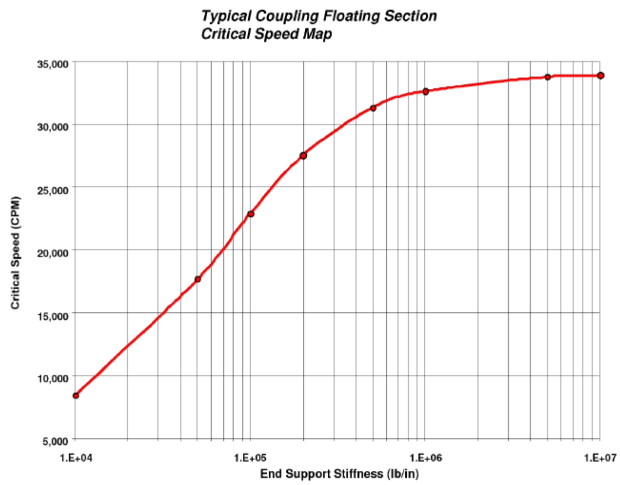

Figure 42. Graph of Support Stiffness Versus Critical Speed.

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.

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

- *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.
- *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*.
- *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.
- *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. The geometry and characteristics of connected machines and couplings vary greatly. The more one knows about a machine or coupling, the lower the LCS margin can be. Current industry margins are not always adequate. API 671 Third Edition (1998), for example, only specifies an LCS ratio of 1.5, without defining the method. This factor should be at least 2.0 if the supports are not considered, or, better, the stiffness of the supports should be taken into account.

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., will be required to achieve the higher number. Whatever the design, the important point is that one should be careful when evaluating a coupling's lateral critical speed.

BALANCE TOLERANCES AND METHODS

Balance is directly related to the machining of the coupling. The concentricity, pilot fits and squareness of parallel planes will determine in a large degree the balance and repeatability of the balance. API 671 (1998) devotes several paragraphs to machining and tolerance specifics. This applies to both gear and dry flexible element couplings.

Several types of balance requirements can be specified for couplings. Component balance requires that each individual part or factory assembled component be balanced to a specified tolerance. Once each component is balanced the balance requirement is complete. This type of balance lends itself to the interchangeability of parts but does not address the balance of the complete assembled coupling.

Component balance and assembly check are the next level of balance. Each individual component is balanced to a specified tolerance. Then the coupling is completely assembled and the balance of the assembly is checked to a specified tolerance. If the assembly does not meet the tolerance, the coupling must be disassembled and the components rebalanced. Balance corrections are not allowed while the coupling is assembled. When the assembly check is complete, mating parts are match marked to ensure the balance is repeated when installed. This balance still allows for the interchangeability of parts.

Component balance and assembly balance provides a much lower residual unbalance of the complete coupling assembly. After all the individual components are balanced the assembled coupling is balanced as an assembly with corrections made to the assembled coupling. This prevents the interchangeability of parts and the coupling must be assembled according to the match marks. Dry flexible couplings, disc and diaphragm, have no clearances and can be balanced to much lower tolerances.

In critical gas turbine high speed applications, balance tolerances are required to be much lower then API 671 (1998). Critical applications can include high speed operation, long shaft separations, and sensitive bearings. Special balance procedures and tolerances have evolved to run in these sensitive applications. For these applications, parts are balanced down to tooling levels (< 0.28 g/in), hardware is balanced to match sets of 0.05 gram, and parts are machined to minimal runout tolerances. Hubs and sleeves for dry couplings are then assembled into flexible halves and skimmed so that mating flanges and outer diameters runout to 0.0003 inch TIR or less. These halves are then balanced.

When it is required, the coupling is completely assembled, and then an assembly balance is done (Figure 43). The coupling is then taken apart at all customer connections and reassembled for an assembly check balance. This practice ensures that the coupling balance can be repeated. API 671 (1998) calls for the assembly check balance to 40 W/N where W = coupling weight and N = the coupling operating speed (rpm). With tighter machine tolerances and better component balances, many times needed for gas turbine applications, balance repeatability of 8 to 10 W/N can be achieved.

Figure 43. Assembly Balance.

GAS TURBINE FEATURES

Speed pickups and key phasor slots can be added to turbine coupling components. Speed pickups 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 pickup 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 held perpendicular to the bore within 0.001 of an inch. Turning gear teeth on one of the coupling components are another of the many options available for gas turbine couplings.

Torquemeters help determine the performance of connected turbomachinery by measuring the twist or strain in a coupling spacer or other coupling component due to torque. From torque and speed, the machine's horsepower output (or input) and hence performance can be tracked. The output of torquemeters is used for long-term trending, gas consumption, and pollution control monitoring. Torquemeters are also used on test stands and in commissioning of new installations.

There are two main types of torquemeters: the strain gauge and the phase shift. The strain gauge torquemeter uses strain gauges attached to a coupling tube or other component to measure torque induced strain (Figure 44). The strain gauge requires that power be applied to the gauge circuit and an output signal be transmitted to a stationary receiver, typically using FM signals. Electroniccomponents are attached to, and rotate with the coupling.

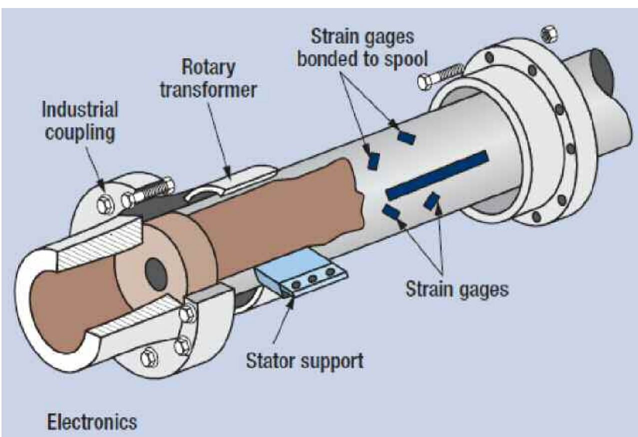

Figure 44. Strain Gauge Torquemeter.

A phase shift torquemeter uses stationary probes to measure the deflection of a tube or coupling part (Figures 45 and 46). A pair of signal tooth wheels is rigidly attached or machined into a coupling component. The probe picks up the relative movement of tooth wheels and determines deflections, which are directly related to the torque applied.

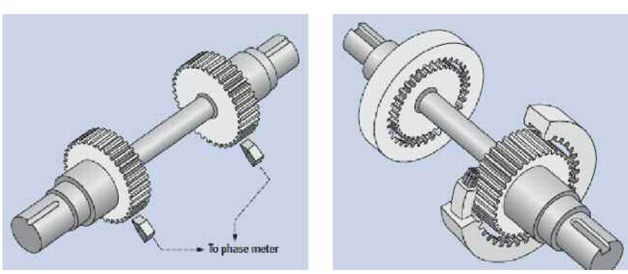

Figure 45. Phase Shift Torquemeters with Separated Gear Rings.

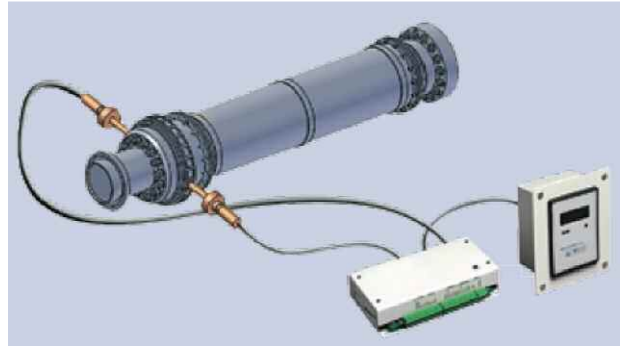

Figure 46. Phase Shift Torquemeter Compact Design.

Both types of torquemeters can achieve less than 1 percent error when originally installed, and the strain gauge type is usually less expensive. But the gauge electronics may not hold up in long-term continuous service, especially in high temperatures.

In the phase shift type, no electronic components are attached to the coupling. The electrical signal is processed away from the coupling. This makes it more reliable in the long term. The drawback is higher initial expense, as more complicated modifications or additions must be made to the coupling.

MATERIALS AND 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 accommodated 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 nonstainless 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 and similar numerical computer software are 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 gauging, etc.) to prove the model. Many engineers believe that an

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FEA calculation answer is automatically correct. This cannot always be, because of the load and boundary assumptions used to build the FEA models.

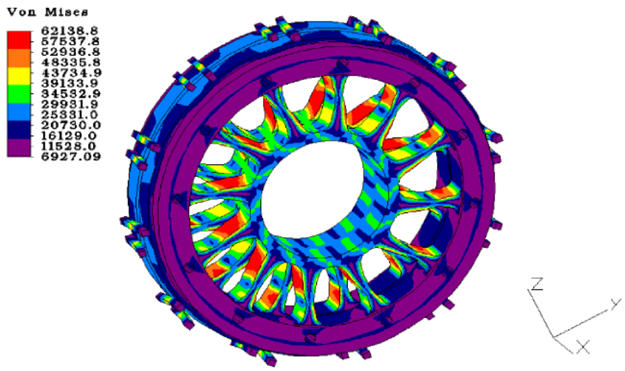

Figure 47. FEA Example.

CONCLUSION

So, what is 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 driver (gas turbine) couplings and driven equipment, rather than an add-on commodity. This trend will continue with higher powered system modeling software.

Moreover newer materials will come on the scene. 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. There already are torquemeters built into the couplings, but what if wireless gauges built into the coupling could give information about alignment, about coupling life. The years ahead will be exciting for the coupling designer.

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