COUPLING CREDIBLE FAILURE MODES AND OWNER OPTIONS TO INTERVENE

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ABSTRACT

While all machine component failures can impact reliability, cost and production, some like coupling failures may have safety risks for personnel. Intervention requires us to build appropriate steps into our machine and system designs as well as our coupling design, operation and maintenance practices. Vibration instrumentation alone can leave us vulnerable for some failure modes.

In this paper, a user's and an OEM's views are presented of the most common coupling failure modes with examples showing details in the final stages of deterioration and identifies options to enable owners to intervene in sufficient time to prevent catastrophic failure.

INTRODUCTION

Most of our effort as machinery specialists has historically been to "do it right" and avoid ever experiencing a coupling failure at all, driven strongly by our need for reliability. Yet, to adequately manage the safety risks of a coupling failure, we must also entertain what failure modes are credible, the details of how each failure mode takes place and finally, what our options are to intervene in time.

It can be very easy for machinery specialists to overinvest in the value of instrumentation in avoiding catastrophic coupling failures. But, just as machinery has failure modes, instrumentation also has failure modes plus we can fail to prevent a failure mode by applying unrealistic vibration measurement strategies.

Some failure modes can only be prevented with sufficient system design considerations, such as operating too near a torsional critical speed or encountering large torque shocks such as driven load impacts or an out of phase breaker closure. Other coupling failure modes such as shaft fatigue failures due to improperly fitted hubs and other human errors are not likely to be detected with instrumentation and can only be reliably prevented by proper mounting of coupling hubs, in other words, proper installation.

If a coupling begins to degrade in service, the coupling design itself needs to be robust enough to restrain the spacer element long enough for vibration interlocks to act, sometimes called anti-flail design. Further, machines such as sleeve bearing motors may have large amount of axial shaft travel which needs to be considered in the machine, system and coupling design to adequately restrain a spool piece.

Finally, despite all of our efforts to avoid coupling failure, our installations and our personnel need be able to adequately respond to failure modes which experience shows are credible. For example, after a disc pack or diaphragm fractures or after severe gear tooth wear, our vibration interlock set points and voting logic strategies must enable the protective system to shut down in time to contain the coupling spool. Voting logic strategies and the selected trip points, especially on slower machines should account for the actual rotor response to insure that an interlock trip will be timely.

COUPLING TYPES AND FAILURE MODES

Gear, disc, and diaphragm couplings are the most common types of couplings in petrochemical and power plants likely to be able to cause major damage and/or downtime in a catastrophic failure. These couplings can be classified two ways, either by how they function or how they are used. The two functional distinctions of couplings are mechanical sliding element (e.g. Gear) and flexible metallic element (e. g. Disc, Diaphragm) and their usage can be separated into either general purpose or special purpose couplings.

No matter the purpose, there are three primary functions of flexible couplings, the first being to connect two pieces of rotating machinery. A coupling must also transmit torque from the driving equipment to the driven equipment, and finally, a flexible coupling must be capable of accommodating all types of misalignment - angular, offset, combination angular and offset, and axial. In addition to these primary functions, flexible couplings can also be required to dampen vibration, reduce peak or shock loads, protect the equipment from overload, or measure the output torque from the driven equipment, in addition to a number of other functions.

General Purpose

General purpose couplings are used in lower speed applications, usually up to 3600 rpm, such as in motor to process pump arrangements. They are considered less critical to plant performance and if shut down will not affect critically other plant processes. These couplings are generally less sophisticated in design, less expensive and the flexible element can be easily inspected and replaced. Gear, grid, elastomeric and disc couplings can all be used in general purpose applications.

Special Purpose

A coupling moves from the general purpose to the special purpose category once it is applied to very critical equipment within the production or process system. The American Petroleum Institute defines "High Performance" in API 671 / ISO 10441 as having a "minimum service life of 5 years…without interruption…" A 27,000 horsepower un-spared boiler feed pump in a base station, or a 12,000 RPM compressor is certainly special purpose and critical to trouble free operation.

Special purpose couplings are used on this equipment in critical applications. Due to the expense of these high power, high speed machines, often times they are not spared yet remain critical to the continued operation and therefore production of the plant. Although these machines are high powered, they are also increasingly sensitive to their environment. Forces or moments that would seem insignificant to high-powered mill machinery become extreme to special purpose machines. Due to the sensitivity, speed, and power, coupling criteria for these applications take on an entirely different perspective.

Years ago the first choice to couple critical equipment was the gear coupling. Although many existing critical equipment applications will still use gear couplings today, newer applications will almost always use disc or diaphragm couplings (Figure 1).

Figure 1. Special Purpose Diaphragm Coupling

This is in large part is due to the fact that a gear coupling experiences wear throughout its service life and requires much more maintenance in making sure they are properly lubricated with oil or grease to minimize the wear. Wear at the tooth tip causes pilot clearances to increase, while on the flanks increases the backlash between the hub and the sleeve; which can increase vibration and cause the automatic shutdown of the train if excessive. Additionally, it is difficult to evaluate and predict the wear rate of a gear coupling without periodic and time consuming inspections. With advances to turbomachinery technology, increased horsepower and speeds led to problems with gear couplings. No longer could venders manufacture couplings to meet balance requirements, accommodate high angular misalignments and ensure extended service life without interruption. For these reasons, a transition has been made to the non-lubricated couplings that are primarily used on special purpose equipment today.

For information on how Gear, Disc, and Diaphragm couplings transmit torque and accommodate angular and offset misalignment and axial movement, see Corcoran, et al. (2007).

Causes of Coupling Failures

Except for material flaws, which are rare, all couplings that fail do so primarily, and overwhelmingly, because they are subjected to loads, and therefore stresses, beyond the capacity for which they were designed. These overstresses will come from poor maintenance, incorrect operation, and incorrect installation, which leads to excessive misalignment, excessive torque (either steady state, transient, momentary spike, or oscillating), lack of proper lubrication (gear couplings), and other damaging conditions. Table 1 lists these general causes and examples of each, while Table 2 lists specific causes for coupling failures. These excessive loads can affect differently how each of the coupling types covered react and fail.

Table 1

GENERAL CAUSE OF FAILURE	EXAMPLES
Poor Maintenance	Improper Lubrication, Lack of Lubrication (Gear Couplings), Failure to Inspect (Disc and Diaphragm), Poor Maintenance of connected equipment leading to excessive loads on the couplings
Incorrect Operation	Running Incorrect Gas, Liquids in Gas, Too Frequent Starts and Stops, Start up before previous Shut Down is complete, Improper Vibration Monitoring, Electrical faults
Incorrect Installation	Poor alignment, Over tight or Loose Fasteners, Incorrect Hub/Shaft fits on Keyless, Keyed, and Splined Connections, Improper Lubrication or Poor Installation allowing lack of lubrication (Gear Couplings)

Table 2

SPECIFIC CAUSE OF FAILURE	EXAMPLES
Excessive Misalignment	Poor initial Alignment, Unaccounted for Thermal Movements, Foundation Settling, Pipe Strain, Improper Equipment Mounting Design or Loose Equipment Mounting Bolts
Excessive Torque	Electrical fault (Momentary Spike), Liquids in Gas (Momentary Spike or Excessive Steady State), Unaccounted for Motor Induced Torque Oscillations (Synchronous Motor – Transient Start Up), (VFD – Steady State)
Improper Lubrication (Gear Couplings)	Wrong Direction Oil Nozzle, Inadequate Oil Flow, Sludge Build Up, Wrong Grease, Not Enough Frequency of Greasing, Not Cleaning out old Grease, Allowing Grease to Leak Out (Damaged or Missing Flange Gasket), Grease Displacement by water or dust
Other damage	Bolt Failure from Too Loose or Too Tight Installation Torque

Gear Coupling Failure Modes

The typical failure mode of gear couplings is that the teeth wear away from improper lubrication and/or excessive misalignment until there is excessive backlash and tooth tip clearance leading to high vibration, or they become so thin they can no longer support the torque and roll over, an example which is a hub spinning inside a sleeve (Figure 2).

Figure 2. Excessively Worn Gear Teeth

While failures caused by a combination of factors occur, a primary cause can usually be identified, and the most common gear coupling failures involves improper lubrication (approximately 75% of known failures).

In one scenario, the additives and impurities present in the lubricant are separated and retained in the coupling due to centrifugal forces, creating sludge. As the quantity of sludge increases over time, it can impair axial float, corrode the teeth thereby accelerating their wear rate, or reduce the circulation of the lubricant. In extreme cases, sludge can even lock-up the coupling and prevent the movement necessary to accommodate misalignment in (Figure 3) per Calistrat and Webb (1972).

Figure 3. Sludge Lock up In Gear Coupling. (Calistrat)

Another common scenario is caused by inadequate supply of lubrication. For example, in a continuously oil lubricated special purpose gear coupling, if the oil flow rate is too low, or the oil flow nozzle is pointed in the wrong direction, increased friction will accelerate tooth wear. As the wear increases the cross sectional area of the tooth will decrease, and can lead to premature failure.

In addition to lubrication, excessive misalignment causes failures due to an increased bending stress towards the end of the gear teeth, or end loading (Figure 4).

Figure 4. Gear Teeth Worn From Excessive Misalignment

Under extremely high misalignment, tremendous forces are transmitted through the couplings to the connected shafts and bearings. As the bearings wear and the misalignment increases, bending stress at the end of the gear teeth is increased and wear accelerated. Misalignment failures account for about 20% of known gear coupling failures.

It's important to note that tooth wear is not the only result of inadequate lubrication and excessive misalignment. If a gear coupling locks up from excess sludge or just from high misalignment the bending loads on the coupling are severe and other coupling components and shaft and flanges can fatigue and fail catastrophically.

Disc Coupling Failure Modes

The most common failures for disc couplings are caused by either excess misalignment or a torque overload. Excessive angular misalignment applies an alternating stress to the disc packs and which can eventually lead to a failure due to bending fatigue. One of the benefits of multiple disc pack couplings is multiplicity since the outer discs experience the highest stress from angular misalignment and they are the first to break (Figure 5).

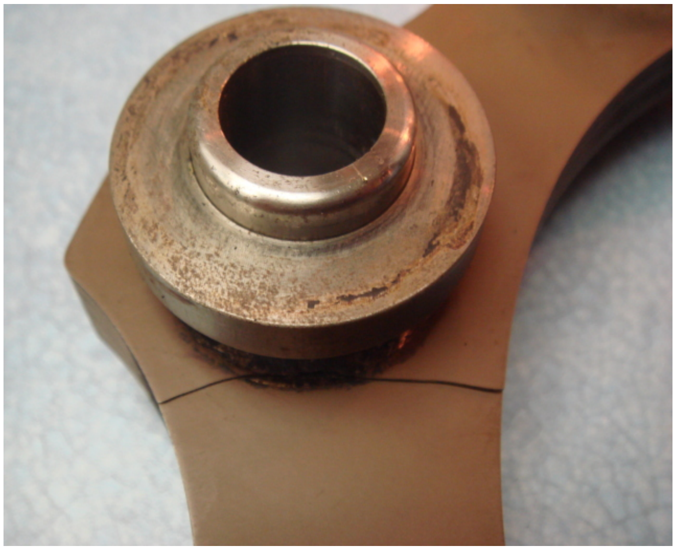

Figure 5. Disc Excessive Angular Misalignment Crack

If one or a few outer discs break the load redistributed among the inner discs, each disc will subsequently carry a higher torque load but a lesser stress due to misalignment. As the disc pack unravels from the outer discs inward, increased unbalance will cause higher machine vibrations, and the machinery can be safely shut down. Excessive axial with angular misalignment produces a tensile stress in both the compression and tension links of the disc pack, and bending fractures are likely to occur on both sides of the bolt hole (Figure 6).

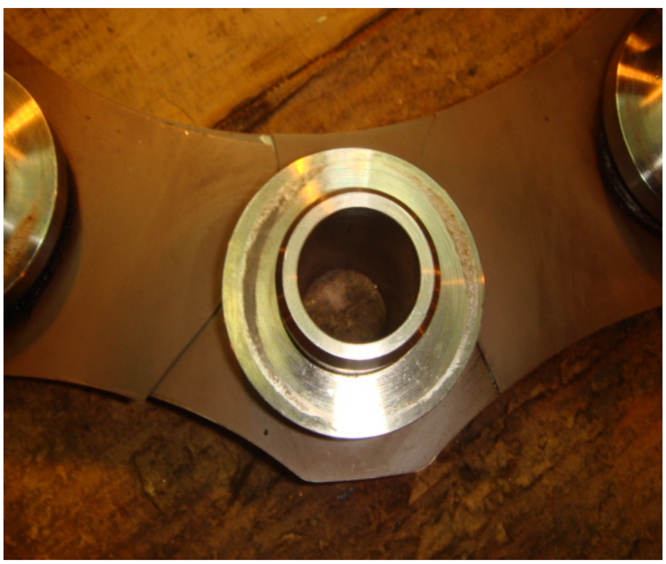

Figure 6. Excessive Angular Misalignment And Axial Movement Cracks

The other major cause of disc coupling failure is due to a torque overload. In the event of a torque overload – caused by a compressor ingesting a liquid slug, or a generator fault, etc., – the disc pack will yield. Distortion of the bushings, bent bolts and contact between the bushing and the disc pack flange may also be evident upon further inspection (Figure 7).

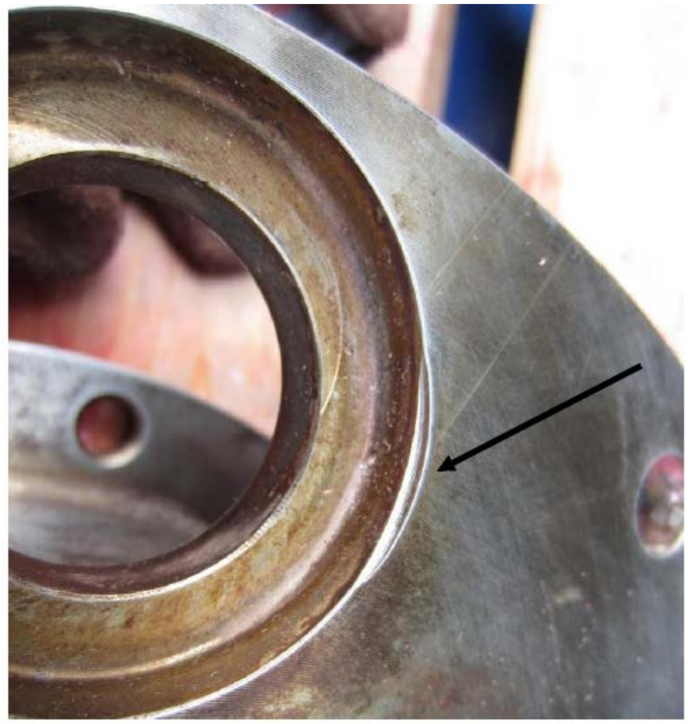

Figure 7. Torque Overload Bushing Indentation in Disc Pack Flange

If the load is large enough, fractures in the disc pack can occur at the links or at the bolt holes. However, though yielded, the disc pack coupling will stay together, and typically cause high vibration from unbalance, unless the load is so high and monstrous that the whole coupling fails at various places catastrophically. More commonly, again, parts will yield but not break from potentially high over torques (Figure 8).

Figure 8. "Twisted Sister" Yielded (Buckled) Spacer From Over Torque

Torsional oscillations can also cause significant damage to couplings. Seen on trains driven by synchronous motors or variable frequency drives, failures due to severe torsional oscillations can also occur with applications with reciprocating engines or pumps that have torque pulses from an engine firing cylinders or reciprocating pump action. Moderate oscillations may cause fatigue failures (either high or low cycle) at either the bolt holes accompanied by considerable fretting, or in adjacent tension and compression links. High magnitude torque fluctuations can result in bidirectional elongation of the disc pack at the bolt hole as well as serious fatigue failure to other coupling components if they are not designed for the torsional load (Figure 9), Corcoran, et al. (2010).

Figure 9. Excessive Torsional Oscillation Fatigue Failure in Spacer

Diaphragm Coupling Failure Modes

Just like Disc Couplings, the most common failure modes for diaphragm couplings are due to either excess misalignment or a torque overload. Failure due to misalignment generally means excessive angular or parallel offset misalignment, with or without axial misalignment.

Angular misalignment causes an alternating stress in the web of the diaphragm as it bends back and forth with each revolution. Failures occur due to bending fatigue and start with a crack in the flexible element. Axial misalignment stretches the diaphragm, resulting in an additional continuous stress (Figure 10).

Figure 10. Excessive Angular Misalignment and Axial Movement Diaphragm Failure

Torque overloads cause distortion of the diaphragm and can cause a rippling effect in the web (Figure 11).

Figure 11. Diaphragm Torque Overload

In addition, excessive torsional oscillations can cause a fatigue crack at a 45 degree angle to rotation. (Figure 12)

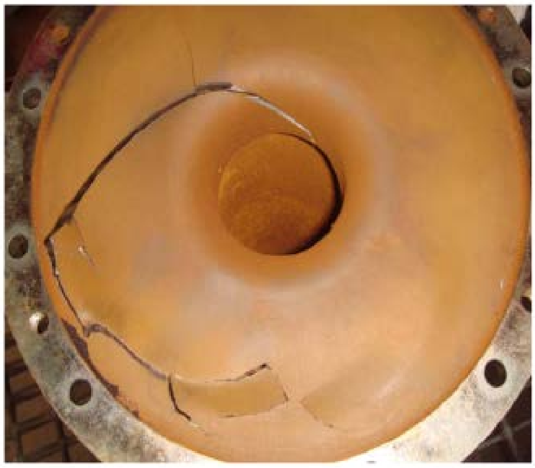

Figure 12. Excessive Torsional Oscillation Failure

CASE HISTORIES

The case histories presented below have been chosen to show the different possible scenarios of a coupling failure, and how or whether or not the controls in place mitigated the failure. They are outlined in Tables 3 and 4. Of course, it must be noted that in 99%+ applications catastrophic ejections do not and will not occur. The purpose here is to make the User aware of the potential, and also the possible ways to even further prevent a nasty coupling failure.

TABLE 3

Case Number and Title							
Coupling Type	**Equipment**	**HP @ RPM**	**Correct Coupling Design**	**Vibration Probes Mitigated**	**Anti-Flail Worked**	**Fail-Safe Worked**	**Coupling Ejection**
Case 1: Disc Coupling & Parts Ejection - Insufficient Service Factor & Misalignment Cause Turbine Over-Speed Burst							
General Purpose Disc	**Steam Turbine/Cent. Pump**	**600 @ 1800**	**NO**	**NO**	**NO**	**n/a**	**YES**
Case 2a: Gear Coupling Ejection – Loss of Lube & Misalignment from Bad Foundation							
General + Purpose Gear	**Motor/GB/Cent. Compressor**	**2500 @ 6820**	**YES**	**NO**	**NO**	**n/a**	**YES**
Case 2b: Mitigation of Gear Coupling Ejection by Vibration Interlock							
General + Purpose Gear	**Motor/GB/Cent. Compressor**	**2500 @ 6820**	**YES**	**YES**	**n/a**	**n/a**	**NO**
Case 3: Disc Coupling Impact Failure Mitigated by Interlocking Flange - Risky Run on High Vibration							
Special Purpose Disc	**Steam Turbine/Centrifugal Compressor**	**15000 @ 10500**	**YES**	**NO at FIRST**	**YES**	**YES**	**NO**
Case 4: Disc Coupling Fatigue Mitigated by Vibration - Misalignment from Soft Turbine Mounting Design							
Special Purpose Disc	**Gas Turbine / Centrifugal Compressor**	**5890 @ 12133**	**YES**	**YES**	**n/a**	**n/a**	**NO**

TABLE 4

Case Number and Title							
Coupling Type	**Equipment**	**HP @ RPM**	**Correct Coupling Design**	**Vibration Probes Mitigated**	**Anti-Flail Worked**	**Fail-Safe Worked**	**Coupling Ejection**
Case 5: Diaphragm Coupling Failure Mitigated by Vibration & Shroud – Fabrication Weld Defect							
Special Purpose Diaphragm	Motor/GB/Centrifugal Compressor	12,000 @ 1800	YES	YES	YES	n/a	NO
Case 6: Diaphragm Fatigue on Test Stand Mitigated by Vibration & Anti-Flail – Excessive Misalignment							
Special Purpose Diaphragm	Motor/GB/Centrifugal Compressor	7719 @ 9596 Fail @ no load & 3800	YES	NO	YES	n/a	NO
Case 7: Disc Coupling Ejection – Severely Overheated Gear Box Shaft Removed Centering Integrity							
Special Purpose Disc	Gas Turbine GB/Centrifugal Comp	15000 @ 9496	YES	NO	NO	NO	YES
Case 8: Diaphragm Coupling Ejection – Extraordinary Torque Shock from Generator Phasing Error							
Special Purpose Diaphragm	Steam Turbine / Generator	37221 @ 3600	YES	NO	NO	n/a	YES
Case 9: Gear Coupling Rapid Teeth Wear – Failure Mitigated by People - Caused by Lube Loss & Human Errors							
General + Purpose Gear	Motor/GB/Centrifugal Compressor	1500 @ 6200	YES	NO	n/a	n/a	NO
Case 10: Disc Pack Ejection – Large Misalignment and Incomplete Instruments - Machine Risk vs. Process Release							
General Purpose Disc	Turbine/Compressor	900 @ 3000	NO	NO	NO	NO	YES
Case 11: Disc Pack Ejection – Slip Fit Hub, No Anti-flail, No Low Speed Vibration Mitigation							
General Purpose Disc	Motor/Vertical Compressor	100 @ 3590	NO	NO	NO	n/a	YES
Case 12: Disc Pack Failure Mitigated by People – Large Misalignment							
General Purpose Disc	Turbine/Refrigeration Compressor	1000 @ 4900	YES	NO	NO	n/a	NO
Case 13: Gear Coupling Rapid Tooth Wear - Chronic Water Contamination Cause Turbine Over Speed Burst							
General Purpose Gear	Turbine/Boiler Feed Pump	400 @ 3590	NO	NO	n/a	n/a	YES
Case 14: Disc Pack Fracture Mitigated by Vibration & Interlocking Bolts - Impact fracture from Process Solids							
General Purpose Disc	Motor/GB/Centrifugal Compressor	900 @ 8300	YES	YES	YES	n/a	NO

Case 1: Disc Coupling & Parts Ejection - Insufficient Service Factor & Misalignment Cause Turbine Over-Speed Burst

A disc coupling failed on a 600 HP, 1800 RPM steam turbine-driven centrifugal pump. The failure resulted in ejection of the coupling spacer in spite of it being an anti-flail design. When the coupling failed the turbine speed increased from 1800 to 3100 due to a failure of the governor and trip systems. When the coupling speed increased, the coupling spacer was ejected from the machine. No personnel were injured, but the turbine was damaged beyond repair (Figures 13, 14, 15, and 16).

Figure 13.Damaged Turbine

Figure 14. Turbine Minus Bearing Bracket and Seal Housing

The bolts in the pump end of the coupling disc pack spacer failed at the adapter plate interface. Note that this is not the usual disc pack fatigue failure due to misalignment. The failed bolts are the bolts that hold the adapter plate to the disc packs. These bolts are not usually removed or otherwise disturbed when a coupling spacer is removed or installed.

Figure 15.Section of Pedestal Support

Why did the bolts fail? Even in the over speed condition, the turbine speed did not exceed the rated maximum speed of the coupling. Indeed, many of this style coupling were used in this plant at speeds in excess of this particular over speed condition. The coupling was undersized for the application; it had less than a 1.0 service factor. Additionally, it was very difficult to maintain shaft alignment between the turbine and pump because of excessive pipe strain on the pump from piping design to compensate for pipe movement from thermal growth. The piping system was heavy wall alloy pipe with very limited space around the pump and pipe rack. Although the thermal alignment problem had been identified in the past, modifying the piping system was not considered cost effective because there had been very few bearing or seal failures and no coupling failures in the maintenance records.

Figure 16. Section of Turbine Shaft

The coupling bolts failed because of incorrect application (under sizing) and misalignment. Once the coupling bolts failed at the pump end of the spacer, the spacer assembly became eccentric, causing severe vibration and friction contact between the spacer which was still rotating and the pump end coupling hub which was stationary due to the broken bolts. At the same time, the turbine speed increased due to failure of the governor and over speed trip system, which increased the vibration due to the spacer eccentricity. Within seconds, the coupling spacer was ejected because the anti-flail extension on the spacer assembly was overheated and deformed from friction due to contact with the non-rotating pump end coupling hub. The disc pack at the turbine end of the spacer failed from low cycle fatigue, as the eccentric spacer stressed the discs far beyond the design limits. The disc pack at the pump end of the spacer failed from impact with stationary objects while rotating during the failure (Figure 17).

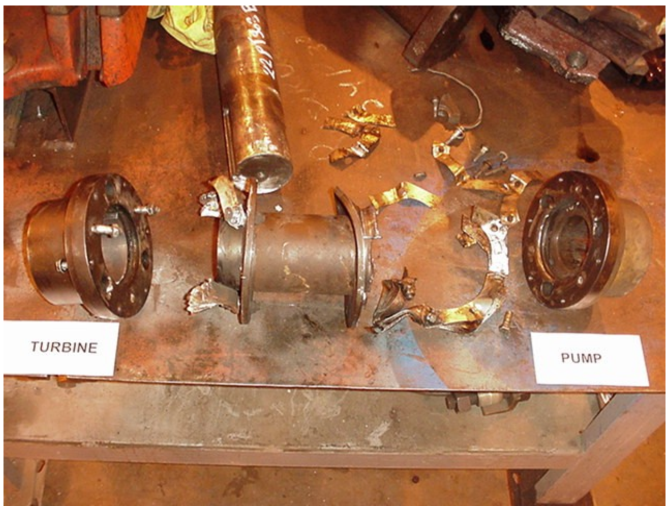

Figure 17.Failed Coupling Components

The turbine shaft was pulled through the turbine wheels (Figure 18) approximately 8-10 inches and the turbine shaft broke at the axial location of the second stage turbine wheel (which was the turbine wheel location after the shaft had pulled through the wheels). The turbine shaft was also bent approximately 30 degrees, with the bend starting approximately inboard of the outboard radial bearing. (Figures 19 and 20) The bend and break of the turbine shaft were a result of the large eccentric mass and unbalance caused by the eccentric coupling spacer. It is unusual to generate enough force to pull a turbine shaft through the substantial interference fit of the turbine wheels to the shaft, but perhaps the wheel fits were released during the over speed of the turbine rotor.

Figure 18.Turbine Disc with Damage at Bore

Figure 19.Bent Turbine Shaft Segment

Figure 20.Opposite End of Failed Turbine Shaft

The turbine inboard bearing housing and pedestal support essentially disintegrated during the failure. This housing and pedestal assembly is made of cast iron. The damage to the bearing housing was most likely caused by forces from the bent shaft and possible impact from the rotating shaft and flying parts. Although the pieces of the housing were so badly damaged as to make reconstruction of damage progression impossible, there is some indication that the bearing saddle broke off inside the housing from overload. The broken inboard turbine bearing housing caused the turbine to fall towards the pump, which caused the governor and trip valve assembly to break away from the turbine casing.

Case 2a: Gear Coupling Ejection – Loss of Lube & Misalignment from Bad Foundation

This legacy refrigeration compressor shown in Figure 21 had a size 2-1/2 gear coupling was driven at 6820 RPM by a gearbox and a 1775 rpm, 2500 HP motor and initially had no vibration instrumentation.

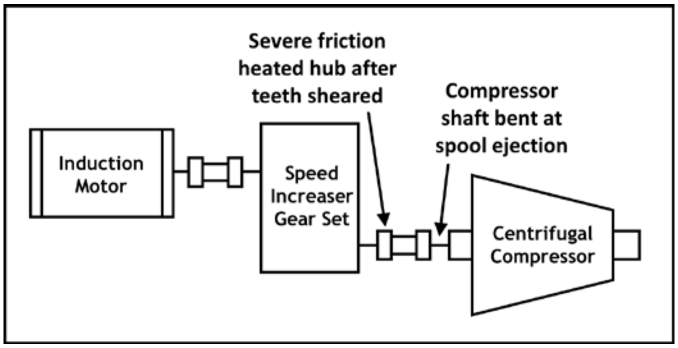

Figure 21. Legacy refrigeration compressor initially had no on line vibration measurement for alarm and interlock protection.

Operators had taken readings and inspected the compressor just 90 minutes prior to the failure and did not notice anything unusual, which underscores how quickly the failure occurred. The first indication of trouble was when the control room got out of limits process condition alarms.

In this first failure, the coupling teeth had worn thin enough to strip. Then, and without the benefit of high vibration interlock protection, the motor continued to run as friction severely overheated the hub and spool teeth as shown in Figures 22 and 23.

Figure 22. Friction heated coupling hub, teeth are gone!

Once the teeth stripped, the compressor rotor turned very slowly, driven only by friction. The friction was exacerbated by unbalance forces of the spool piece which could no longer be accurately centered by the damaged hub. The rate of heat generation is thus the frictional force times the difference in velocity between the spool pinion driven by the motor versus a very slow rotor hub. We estimated the hub heated to red hot in about five minutes.

Figure 16 shows the spool ring gear not only stripped the teeth, the combination of rapid heating and unbalance forces permanently swelled the diameter where the teeth had been. Forces were high enough to bend the compressor shaft slightly and damage the compressor bearings.

Figure 23. Friction heated coupling spool.

As the spool continued to heat and weaken, it ran progressively farther off center producing ever higher unbalance forces. Soon after the teeth stripped, the unbalance forces favored one side and heated a localized area much faster, became very plastic and weak enough to punch through the hub wall, which then ejected the spool from the machine and hit a block wall and ricocheted back as shown in Figure 24. Fortunately, no one was near and no one got hurt.

The ejected spacer weighed 28 pounds and on the side of the ring gear was beginning to yield rapidly, that end was then off center by at least 0.25 inch which produced an unbalance force in excess of 9200 pounds to bend the compressor shaft.

The gear box sole plate grouting was broken loose and was pre-existing damage. Once the mounting integrity of either casing is lost, it is no longer possible to control the pinion to compressor shaft alignment.

Lack of vibration interlock protection and loss of control of shaft alignment left the plant vulnerable with no way to effectively mitigate loss of lubricant to the coupling teeth.


Figure 24. Ejected spool impact and final location.

Case 2b: Mitigation of Gear Coupling Ejection by Vibration Interlock

A spare compressor was then installed with a shaft vibration monitoring system. Additional deficiencies which were found and corrected included a loose gearbox soleplate due to damaged grout and excessive nozzle loads from the evaporator.

After a full year of operation, a plant engineer had made a habit of inspecting the machine for grease leaks or other signs of distress. Just a few hours after his inspection, the coupling failed again, but this time vibration interlocks safely shut the machine down prior to stripping the coupling teeth (Figure 25).

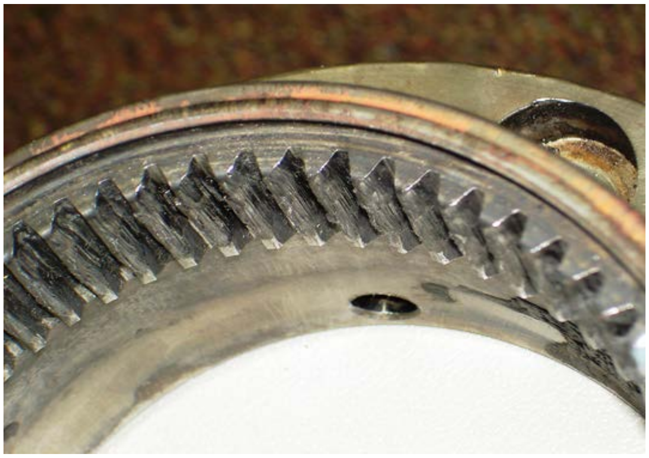
Figure 25. Heavy tooth scoring from loss of lubrication.

Vibration appeared normal until shortly before the failure, but the time interval between alarm and interlock was too short for operators to intervene. Automatic shutdown mitigated the risk by getting the compressor out of service safely before a repeat of the first failure.

Case 3: Disc Coupling Impact Failure Mitigated by Interlocking Flange - Risky Run on High Vibration

At an Air Separation plant where oxygen and nitrogen from the air is separated and liquefied a Special Purpose Disc Coupling on a Main Steam Turbine Compressor train running at 15,000 hp and 10,500 rpm failed, but the failure was not noticed until extremely high vibration – 5 mils – did not recede. Apparently the turbine bearings were damaged after a condensate slug 2 months previously, but in the process of replacing the bearings, the coupling was not inspected!

When it finally was, cracks were noticed in the discs. When a detailed inspection at the factory took place, it was noticed that discs in the pack had failed progressively from outer pack to inner, indicating an excessive misalignment condition. As found, all the discs on the turbine end had failed (Figure 26), as had many on the compressor end.


Figure 26. Failed Turbine End Discs

Moreover, the coupling was running on the interlock feature (Figures 27 and 28) on the turbine end! The train had run at 3.5 – 4.5 mils for 2 months and 2 restarts after bearings had been replaced, but no one suspected a coupling failure.


Figure 27. Interlock Engagement Wear on Hub

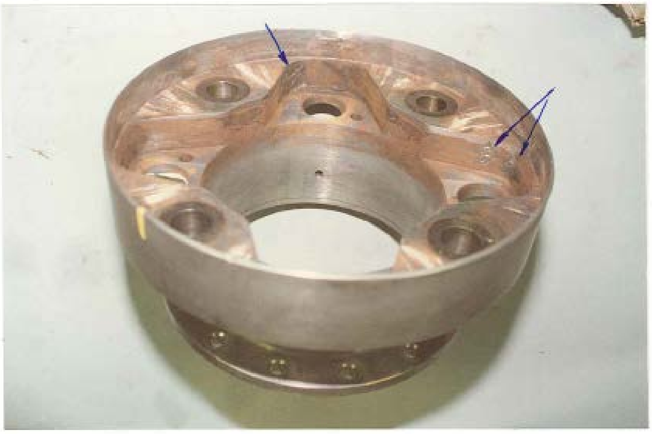

Figure 28. Interlock Engagement Wear on Sleeve

Since the interlock feature (full depiction in Figure 29) worked, serious damage to the equipment train was avoided while the exact cause of the problem – condensate slug, leading to bearing damage, leading to misalignment then coupling failure – was determined.

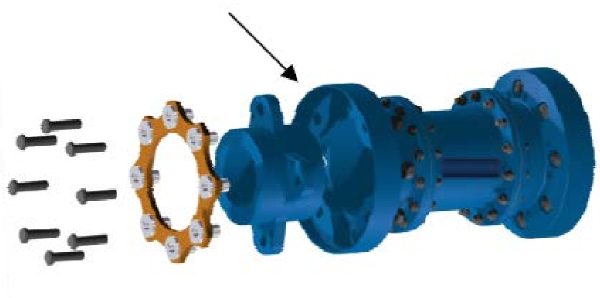

Figure 29. Interlocking Flange Feature

Case 4: Disc Coupling Fatigue Mitigated by Vibration - Misalignment from Soft Turbine Mounting Design

After two identical gas turbine upgrade procedures on an offshore platform, where the turbines were driving centrifugal compressors for gas injection service, the couplings on both trains failed from excessive angular misalignment (Figure 30).

Figure 30. Disc Angular Misalignment Failure

However, since these disc couplings failed gradually and the vibration from unbalance increased gradually, catastrophic failures were averted as, in the 1st failure the high vibration tripped a shutdown, and in the 2nd, the operators shut down the machines for investigation after the vibrations exceeded the Alarm Level (Figure 31).

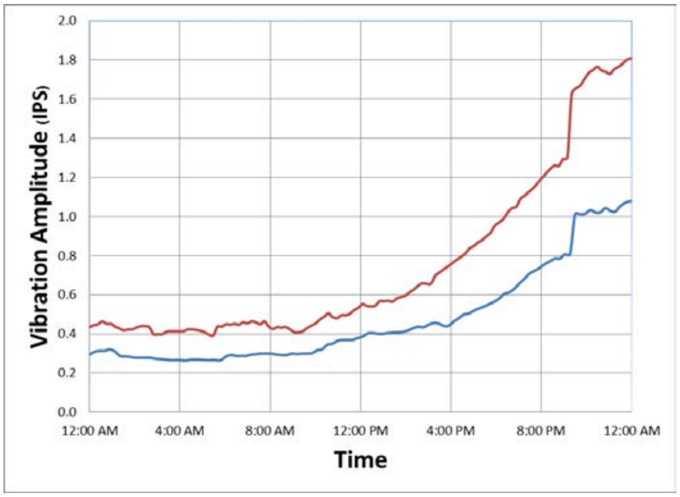

Figure 31. Vibration Probes Mitigated the Failure

The original couplings were repaired after it was determined that the new turbine mounting structure had undersize bolting to keep the Turbine aligned properly during full load conditions, which led to the excessive misalignment to fail the couplings. This undersize bolting was changed.

However, these repaired couplings failed also, so it was determined that the turbine mounting needed to be even more rigid, accomplished with more robust supports and welding (difficult to do on an offshore platform). After the corrections, the trains are now operating well.

Because the vibration probes were in good working order, and the operators followed closely the rises in vibration levels and forced shutdowns for investigations, a catastrophic coupling failure was averted.

Case 5: Diaphragm Coupling Failure Mitigated by Vibration & Shroud – Fabrication Weld Defect

The motor to gearbox diaphragm coupling on a 12,000 HP compressor train failed from fatigue due to incomplete fusion of a friction weld, which acted as a stress raiser. Very high nominal stress was present on the piece, as shown by the small fatigue area and the large final-fracture region. This friction weld was used in the coupling between the low alloy carbon steel spacer tube and the precipitation-hardened martensitic stainless steel diaphragm on the gearbox end.

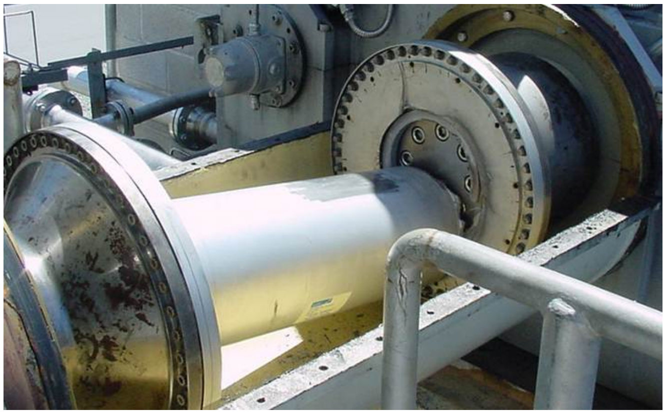

Figure 32. Coupling Assembly after Failure

The motor to gear box coupling failed while the compressor was coming up to speed (Figure 32). The vibration monitoring system shut the compressor down automatically during the failure. No previous maintenance was performed on the coupling when the equipment was down. The failed coupling was four years old at the time of the failure.

The inner shrouds on the coupling provided anti-flail protection preventing the spool piece from being ejected from the coupling assembly and guard. The eccentricity of the spool within the shrouds resulted in mass unbalance sufficient to increase the vibration amplitudes such that the trip system was energized and because the trip levels were set at realistic limits.

The coupling spacer tube is friction welded to the diaphragms as part of the normal manufacturing process. Both ends of the spacer tube had broken from both diaphragms and both diaphragms had shattered (Figure 33).

Figure 33. Components from Coupling Spacer

Visual examination of the failed ends showed the suspected initiation point was on the spacer tube (Figure 34). This face retained more failure features (less smeared) than the matching areas on the diaphragm and was used for further evaluation. Visual examination showed that the majority of the fracture, about 90%, was the final-fracture region, indicating a highly stressed weld joint.

Figure 34. End of Spacer Tube; Failure Initiation Site is at 10 O'clock Position on the ID

SEM analysis showed that while fusion had occurred on most of the spacer tube to diaphragm weld joint, the failure initiation site was an area that had not fused during welding in manufacturing.

Most coupling failures are some kind of degradation in service due to wear or fatigue of components. Metallurgical and fabrication flaws are the exception, but they do occur. This particular manufacturing problem could have been discovered by detailed inspection, but would have requiring dismantling and inspection by the manufacturer. This type of problem would have been best if detected when the coupling was first manufactured.

Case 6: Diaphragm Fatigue on Test Stand Mitigated by Vibration & Anti-Flail – Excessive Misalignment

In a test stand application, Motor – Hydraulic Coupling- (Diaphragm Coupling) – Gearbox - (Diaphragm Coupling) – Low Pressure (LP) Compressor – (Diaphragm Coupling) – High Pressure (HP) Compressor, the Diaphragm coupling between the HP and LP Compressors failed on test from excessive angular misalignment. The normal running condition for this coupling was 7719 HP @ 9569 rpm, but it failed catastrophically at no load at 3800 rpm.

Vibration monitors tripped at 75 microns, when apparently the coupling diaphragms were ripped completely around circumferentially and the now free center section was completely loose from the rest of the coupling. (Figure 35)

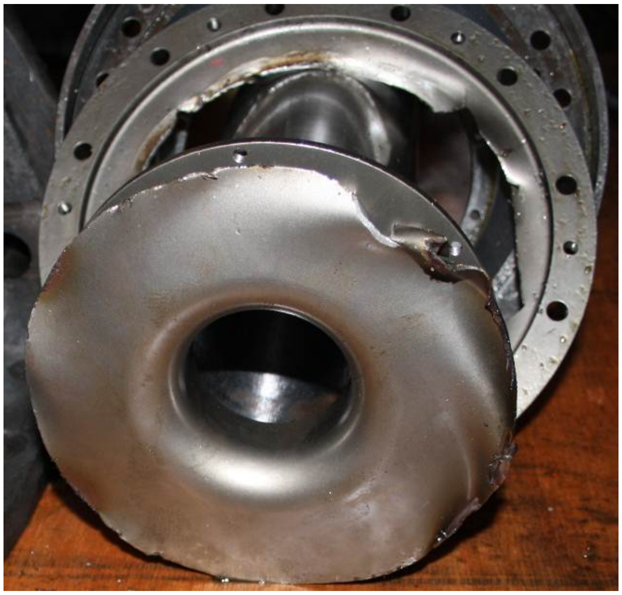

Figure 35. Angular Misalignment Failure

However, due to the anti-flail feature built in to the coupling, the free rotating center section was contained inside the coupling envelope and did not have enough force to be ejected from the coupling guard (Figure 36).

Figure 36. Anti-Flail Worked

Case 7: Disc Coupling Ejection – Severely Overheated Pump Shaft Removed Centering Integrity

In a Gas Turbine / Gearbox / Compressor application the coupling between the turbine and gearbox failed catastrophically and parts were ejected through the coupling guard and through a wall to outside the building. (Figure 37).

Figure 37. Hub with Disc Pack Ejection

Part of the guard went through a window (Figure 38). This was classified as a Near Miss occurrence and gained much attention from all vendor and operator parties involved.

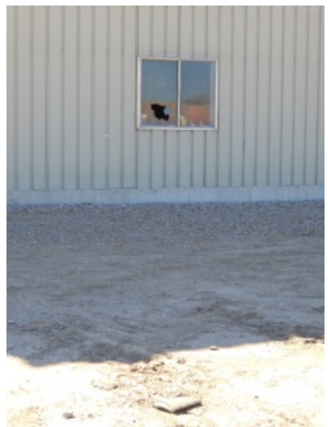

Figure 38. Ejected Part of Guard

Apparently, after much investigation time and expense, it was discovered the gearbox bearings had failed due to a lubrication problem. The resulting high bearing temperatures and vibrations were not picked up by the existing older monitoring system.

The temperature of the bearing got so hot (1600 degrees F) that the shaft on the gear box end grew soft (Figure 39), flailed off center from centrifugal forces, taking the coupling with it, causing a catastrophic breakage of the coupling where parts were then ejected. (Figure 40).

Figure 39. Bent Shaft from Heat

Figure 40. Disc Packs Ripped from Tremendous Forces

This reduced moment disc coupling, inherently an anti-flail design, could not resist the tremendous forces of being off center so much from the center of rotation. Just after the disc pack fractured, the initial unbalance force from the off center spool mass was in excess of 6900 lb. As the overheated pinion journal yielded, the unbalance force got progressively much higher which further accelerated friction heating, and added energy to the spool when it ejected.

Whenever the centering ability of either of the connected shafts is severely compromised, coupling failure is imminent.

Case 8: Diaphragm Coupling Ejection – Extraordinary Torque Shock from Generator Phasing Error

In a Steam Turbine / Generator application, 37221 hp @ 3600 rpm, a 15 inch diameter Diaphragm coupling failed catastrophically on the 1st start up and parts blew right through the coupling anti-flail feature and the coupling guard (Figure 41).

Figure 41. Ejected Diaphragm Spacer

Apparently, the generator was wired incorrectly, and when the operators attempted to synchronize the Generator output with the line, instead of electricity flowing out, it came in and shorted (faulted) the Generator which tried to stop. The coupling took the brunt of the extremely high torque spike load and failed catastrophically (Figure 42).

Figure 42. Ejected Parts when Anti-Flail did Not Work

The Diaphragms were completely ripped apart, flange bolts were sheared and the center spacer broke completely through the anti-flail guards (Figure 43).

Figure 43. Broken Anti-Flail Guard

This is an example of where a typical anti-flail feature could not have prevented the extreme failure of this coupling from the sudden unexpected seriously high torque spike. Proper electrical wiring checks were required and are now in place.

Case 9: Gear Coupling Rapid Teeth Wear – Failure Mitigated by People - Caused by Lube Loss & Human Errors

Coupling manually removed from service just prior to failure

This legacy compressor with a size 2-1/2 gear coupling was driven at 6200 RPM by a gearbox and a 1790 rpm, 1500 HP motor. The coupling was new and had only run for two weeks. The maintenance crew knew shortly after startup that the coupling was leaking grease and hoped to reach a scheduled outage (Figure 44).

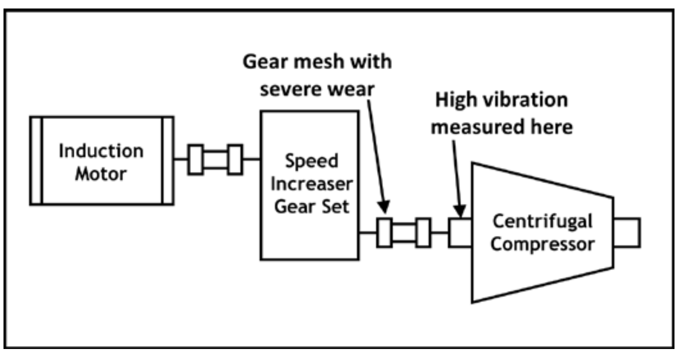

Figure 44. High vibration measured at opposite end of coupling from the severe wear.

Unfortunately, one side of the coupling ran out of grease and then the teeth wore very rapidly during the last 12 hours as shown here in Figure 45.

Figure 45.Coupling half, sectioned for easier viewing.

After Loss of Lube, the teeth wore very deeply in just 12 hours

The teeth deeply embedded which restricted and weakened the hub side teeth enough to start breaking chunks off of the teeth as can be seen in the offset tooth wear in Figure 46. The amount of shaft misalignment was within acceptable limits.

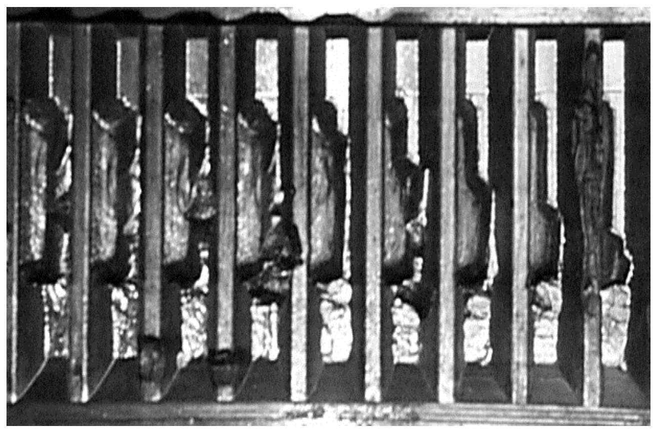

Figure 46. Tooth Damage Close Up.

Also evident are teeth which are beginning to shear emphasizing the teeth would have soon failed catastrophically as the case 2a gear coupling in this paper. The velocity probe interlock should have tripped the compressor off line at 0.6 inches/sec, but it did not! Prompt intervention by a knowledgeable vibration technician got the production crew to manually shut down the compressor which prevented far more severe damage and risk for personnel.

Later in the investigation, an unrealistic voting strategy was being used which circumvented the interlock protection and kept it from shutting down. The voting strategy was set to require high radial velocity and excessive shaft thrust travel, which would never catch this failure mode.

Coupling tooth wear rate when dry was very rapid onset

The most important key learning demonstrated by this "near miss" coupling failure is the very rapid onset of coupling tooth wear. Vibration trend history for this machine showed constant vibration levels which were flat lines for nearly all of the two week run until the last twelve hours of operation when the vibration levels became very erratic and ramped up quickly as shown here in figure 47.

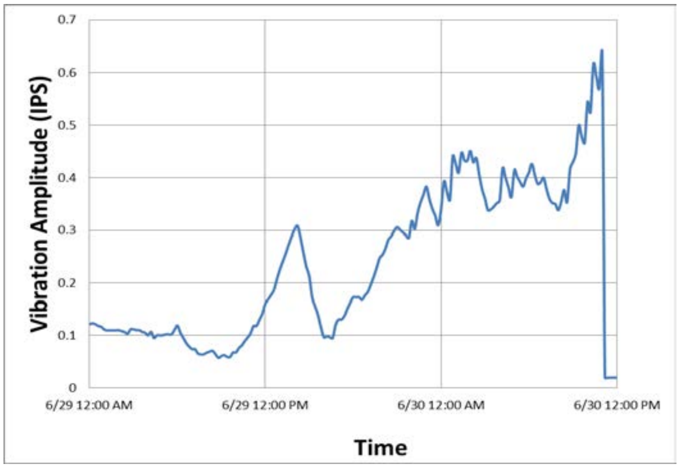

Figure 47.Vibration ramp rate shows very rapid onset

The rapid vibration ramp and an active grease leak just hours before the shutdown shows most of the tooth wear occurred during the mere twelve hour interval after the gear teeth transitioned from boundary lubrication to dry wear.

Like machines, vibration systems may not function properly!

The interlock strategy for machine shut down needs to be able to account all credible failure modes. Unfortunately, in an effort to protect plant production, the instrument engineer presumed any compressor failure would exhibit both high vibration and excessive thrust movement. But, this real failure demonstrated a severely out of balance, failing gear coupling had no affect rotor thrust position.

Why did this coupling leak grease?

This coupling leaked upon start up due to use of a hard faced hammer by a new mechanic during alignment. The resulting dimples shown in Figure 48 were sufficient to prevent the thin OEM gasket from sealing the flanges.

Figure 48. Dimple prevented sealing, allowed grease to leak

Unlike pipe flanges with much softer gaskets, coupling hub gaskets must be thin and stiff enough to make the hubs a rigid joint to stop movement which would fatigue the fasteners.

Case 10: Disc Pack Ejection – Large Misalignment and Incomplete Instruments - Machine Risk vs. Process Release

Compressor upgrade on an existing process gas installation

A legacy 3000 RPM, 900 HP turbine-driven compressor installation was upgraded with a new generation compressor casing and a 6.38 inch diameter disc pack coupling, but the vibration probes and temperature sensors on the new compressor were not yet wired to the control room. Since the old compressor never had any bearing protection, the plant crew did not believe that running for a few months without the instruments active was a major risk, but they were wrong.

However, excessive piping strain which eluded detection by the alignment technician during the final alignment pulled the shafts far enough out of alignment that one disc pack failed in fatigue three months after startup of the new compressor.

Trouble escalated quickly

The first indication that the compressor was in distress by the production crew in the nearby control room was a loud, single bang followed by smoke coming from the compressor bearing housing. The production crew immediately ceased production, and then cut the turbine to 1500 RPM, half of normal operating speed. Half speed operation was to purge the process adequately to avoid any risk of environmental release.

The crew wisely roped off the area to keep personal away from the compressor, not knowing the extent of the damage. When the process purge was nearly complete two hours later, another loud bang was the coupling spool being ejected from the compressor which actually landed outside the roped off area. Fortunately, no personnel were near.

Oblong bolt-hole wear on the compressor hub in Figures 49 and 50 show that after the disc pack failed torque was transmitted through the interlocking bolts until the hub wore too thin and failed into fragments in Figure 51.

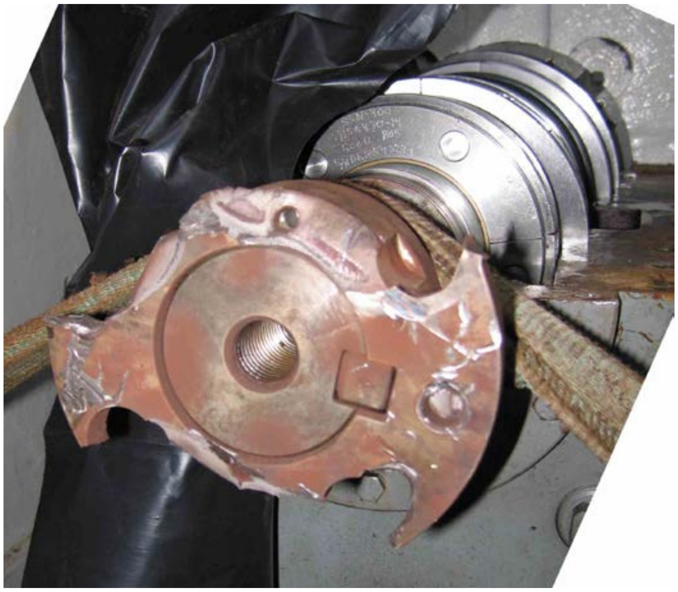

Figure 49.Compressor hub, fractured with worn bolt holes.

Figure 50.Oblong bolt holes in spool hub on compressor side.

Figure 51.Remnants of compressor side disc pack & hardware

Compressor side coupling hub and ejected spool

After failure of the compressor hub, the spool was no longer restrained and the unbalance forces caused a sudden overload fracture of the disc pack shown in Figure 52 which ejected the spool piece as shown in Figure 53.

Figure 52.Turbine hub disc pack sudden fracture.

Figure 53.Coupling spool piece which ejected from the machine.

What held the spool piece in place for two hours?

The compressor and turbine each had tilting pad thrust bearings with substantial load capacity. This prevented the shafts from spreading apart during the coupling failure and helped retain the spool piece for two hours, although not nearly as effective as a modern anti-flail design.

By operating at half speed for the process purge, the compressor power was one eighth that normal load. Using the PV wear rate method, full speed operation would have worn the bolts much faster and would likely have ejected the spool in just fifteen minutes with much more energy.

Even at the comparatively slow 1500 RPM, the spool piece and coupling guard both landed about fifty feet from the compressor, outside the roped off area. While unbalance forces were sufficient to fracture the turbine side disc pack, the turbine shaft was not bent at half rated RPM.

Case 11: Disc Pack Ejection – Slip Fit Hub, No Anti-flail, No Low Speed Vibration Mitigation

A 100 HP 3600 RPM, vertical motor driven compressor (Figure 54) ejected the coupling spool piece (Figure 55) immediately during start up following maintenance work. The operator was still standing at the start switch, and was very close to the trajectory, but fortunately was not harmed. The compressor had vibration probes on the high speed output shaft, but not on the motor or gearbox input shafts.

Mechanics had struggled with the alignment methods on this machine, and the shafts were in fact misaligned. Without appreciating the risks or consequences, someone had modified the hub to be a slip fit to simplify installation, as the protrusion of the compressor shaft beyond the hub shows in Figure 56.

After one of the disc packs failed during startup acceleration, the loose hub on compressor side and lack of any anti-flail features allowed the bolt contact on the hubs to force the hub down the compressor shaft and eject the spool piece.

Figure 54. Vertical compressor with 3590 RPM motor.

Figure 55. Disc pack spool piece was ejected on start up.

Figure 56. Hub was incorrectly modified to be a slip fit.

Although good alignment can provide unlimited life for a disc pack coupling, fatigue is likely to occur whenever misalignment is excessive or if the machine is subjected to a torque shock. So, we must be able to shut machines down safely with a failed disc pack. That in turn requires coupling anti-flail features, properly mounted coupling hubs and thrust bearings in good condition to limit axial shaft end movement.

Case 12: Disc Pack Failure Mitigated by People – Large Misalignment

A legacy 1000 HP, 4800 RPM turbine driven refrigeration compressor (Figure 57) with a disc pack coupling but without any installed vibration protection was found to have unusual vibration by a vibration technician, though the amplitude was not excessive. Disassembly and inspection showed the disc packs were failing in fatigue due to excessive offset shaft misalignment and incorrect axial spacing to accommodate the shaft thermal expansion of thrust bearings at the far ends of each rotor (Figure 58).

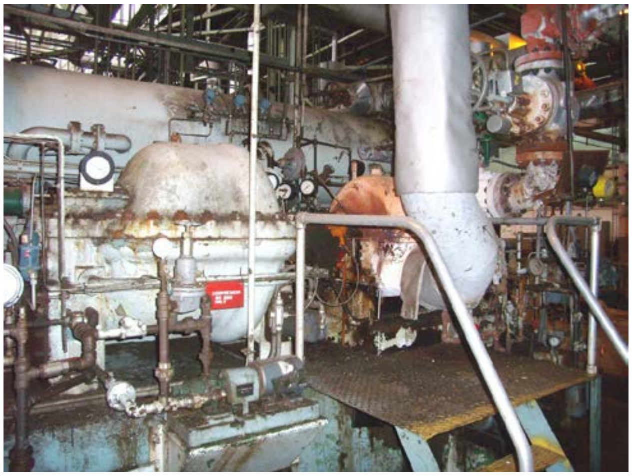

Figure 57. Turbine direct drive refrigeration machine

Figure 58. Failing disc pack with missing shards found by an alert operating crew!

Corrections were made, but a hot check showed that misalignment was still too high. While this is clearly not a reliable method, the crew was coached to monitor carefully for evidence of shards from the disc pack underneath the coupling guard. The worried crew in fact did find shards, missing from the disc pack in figure 19 and managed to get the coupling out of service before a complete failure of the disc pack, despite the lack of an on line vibration monitoring system.

A second realignment was successful in resolving the turbine thermal growth which stopped disc pack fatigue.

Case 13: Gear Coupling Rapid Tooth Wear - Chronic Water Contamination Cause Turbine Over Speed Burst

A 3590 RPM, 400 HP boiler feed pump gear coupling failed, which was the final failure in a cascade sequence of failures that resulted in a runaway of drive turbine and then the turbine blades burst through the cast iron casing as shown in Figure 59.

The turbine had been running load limited with RPM floating on the pump demand. Investigation showed that the maintenance crew was overdue on maintenance tests, so neither the governor nor over speed trip was functioning.

Figure 59. Turbine ran away after gear coupling failed

The most important cause was inadequate maintenance on the governor and trip systems. But, when a coupling fails, the loss of the pumping load and pump rotor inertia imposes the most extreme demand on turbine protection.

Examination of the gear coupling showed that it had lost all of the grease and was instead full of water which created rapid tooth wear as shown in Figure 60.

Figure 60. Heavy wear due to grease being displaced by water.

This boiler feed pump operates in a building with routine steam plumes as the source of water in the coupling. Since water is heavier than grease, any water droplet that gets past the coupling seal ring will centrifuge out and displace the lighter grease, which leaks out on the coupling guard in Figure 61.

Loss of grease should not be considered normal as we can never know how much grease volume remains. For this failure, the loss of grease was also a good indicator that the coupling could not be expected to have normal grease life and is not currently the best choice. A disc pack coupling avoids the weaknesses of limited lube intervals and contamination.

Figure 61. Grease displaced by water collects on guard made visible by an expanded metal guard. Solid guards hide leakage.

Case 14: Disc Pack Fracture Mitigated by Vibration & Interlocking Bolts – Impact fracture from Process Solids

This 900 HP single stage process gas centrifugal compressor was driven by a 1785 RPM motor through a gear speed increaser at 8300 RPM and a 5.5 inch diameter disc pack coupling (Figure 62 and 63).

Impact overload of a coupling can come from multiple sources. In this case the source was a major process upset, resulting in the compressor ingesting a large amount of process solids. The resulting shock load fractured the disc pack of the high speed coupling.

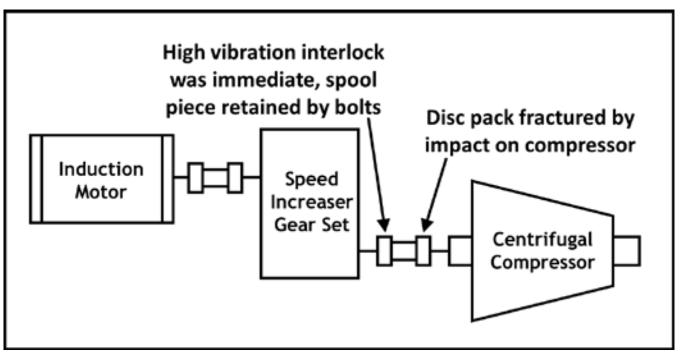

Figure 62. Compressor configuration

The unbalance force of the off-center spool piece running at 8300 RPM was high enough to immediately shut down the compressor on high vibration. While this older style disc pack coupling did not have the best modern anti-flail features (Figure 64), the compressor and gear box thrust bearings provided enough axial load capability to enable the bolts to retain the spool piece long enough for vibration interlocks to shut the compressor down safely.

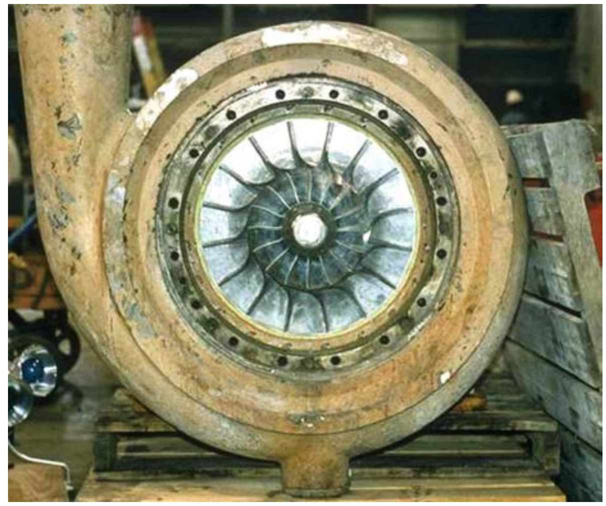

Figure 63. Single stage overhung compressor.

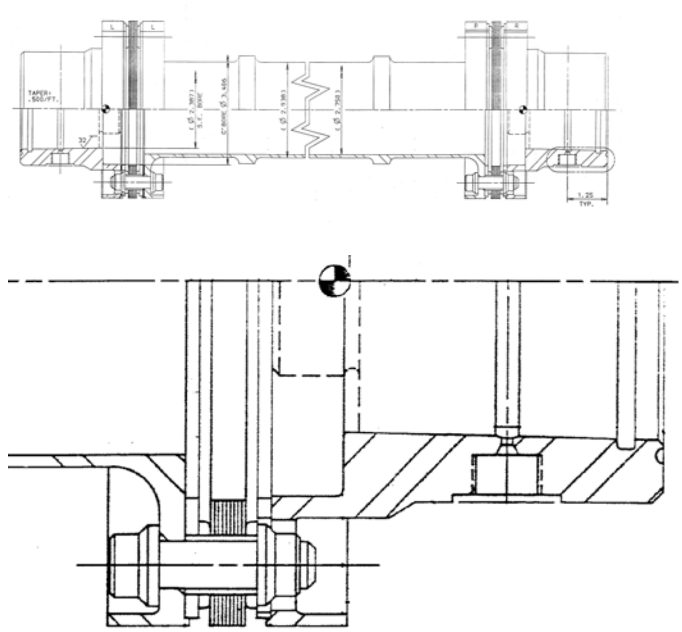

Figure 64. Older style disc pack interlocking bolts only.

AVOIDANCE & PREVENTIVE MEASURES

Application Design Importance

The most important information for the designers of power transmission couplings, especially for Special Purpose Couplings, in terms of preventing coupling failures due to excessive loads, are all the possible loads the coupling will or can experience in the application.

These loads include:

- Normal Steady State Torque (Horsepower and Speed)
- Maximum Continuous Torque

- Potential and Maximum Momentary (Fault) Torques
- Transient Torques (like Start Up Torques) and Maximum Frequency of Same
- Maximum and Normal Angular Misalignment
- Maximum and Normal Parallel Offset Misalignment
- Maximum and Normal Axial Movement
- Potential Oscillatory Torques – Frequency and Magnitude
- Maximum and Trip Speed (Centrifugal Forces)
- High Temperature Potential

Potentially related to the magnitude of these loads, the coupling designer needs information on:

- Rotor Sensitivity to Unbalance
- Maximum Overhung Moment at each end of the coupling
- Torsional Stiffness Requirements
- Shaft Separation
- Maximum Thrust Bearing Loads
- Atmospheric Conditions (Like High H2S, Nitric Acid, etc.)

With all this information the best practical coupling can be designed with the highest level of confidence to avoid a catastrophic event.

Shear Devices – Low and High Speed

There are other options to protect equipment, especially in generator or motor potential fault situations. One is a shear or fault device built in to or attached to the coupling that shears or fails at a predetermined high torque, allowing the machine to coast down safely after the shear event. These devices are recommended for applications with a significant potential for a high torque shock load where a catastrophic failure would be extremely costly.

There are hydraulic shear devices, where a hub or other coupling component uses hydraulic fluid pressure to maintain enough friction between an adjacent component until the torque exceeds a pre-specified amount. Then, an accurately machined and material verified "plug" shears, allowing the hydraulic pressure to release and the components to rotate relative to each other allowing the equipment to coast down safely without damage after the high torque event. This and other shear features such as shear pin couplings (Figure 65) are good for low speed applications only, as with higher speeds and centrifugal forces, parts could rub together and catastrophically fail, leading to potential ejections of parts against or through a coupling guard.

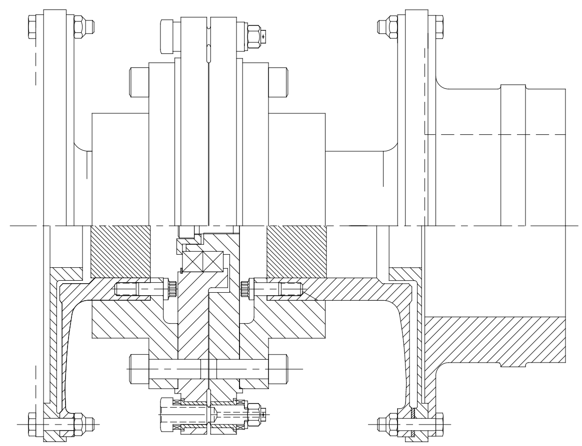

Figure 65. Low Speed Shear Pin Device

The shear cartridge shown in Figure 66 and 67 is a device, which safely works for high speed applications. It has special high speed bearings and other features allowing it to shear evenly when a pre-determined high torque spike is encountered allowing the connected equipment to coast down safely.

Figure 66. Shear Cartridge

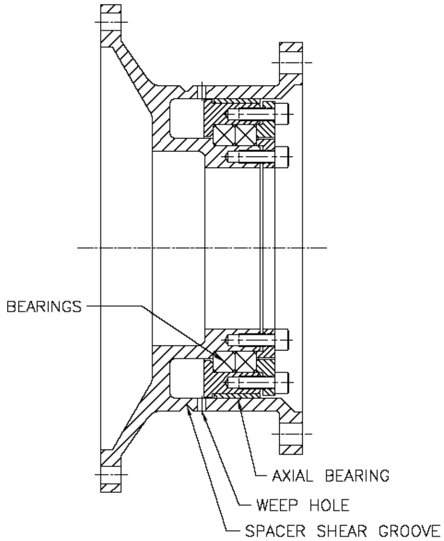

Figure 67. Diagram of Shear Cartridge

Installation

Proper Installation and maintenance of couplings is critical. An improperly fit key can cause a hub failure (Figure 68). Furthermore, an improper hub fit can lead to fretting, hub slippage, yielded hubs, and even shaft failure. See Figures 69 and 70. For information on the proper fitting of keys and information on proper installation of keyless hubs, see ANSI/AGMA 9002-B04 (2005) and ANSI/AGMA 9003-B08 (2008).

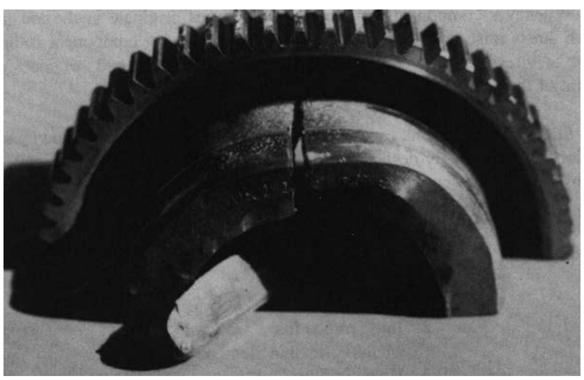

Figure 68. Hub Failure Due to Improper Key Height Calistrat (Calistrat 1994, Flexible Couplings – Their Design Selection and Use*)*

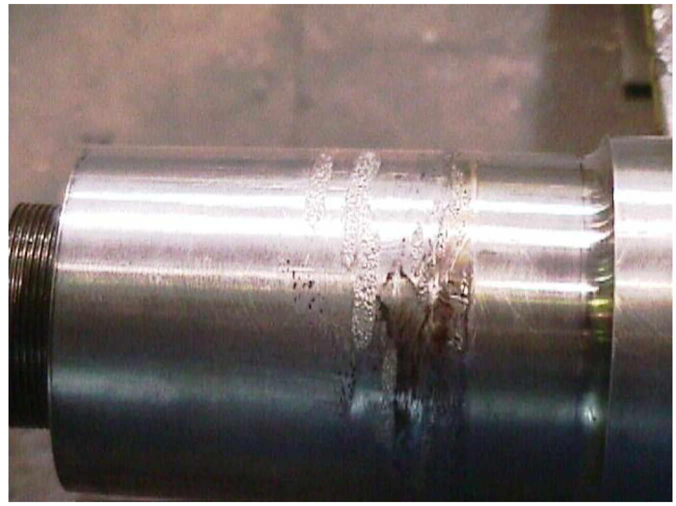

Figure 69. Shaft fretting from improper hub fit

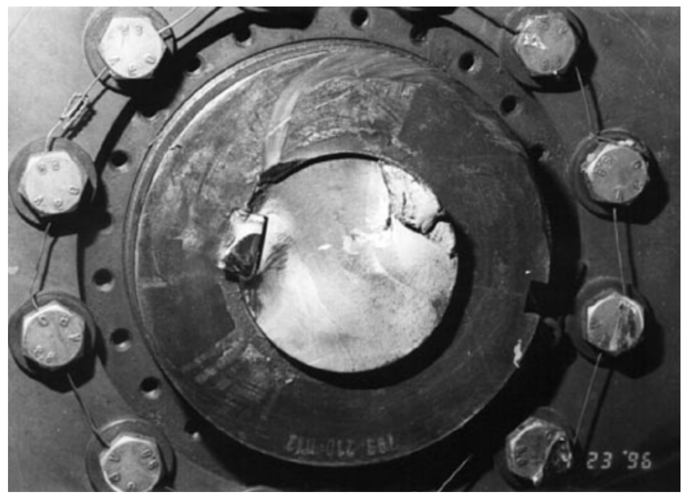

Figure 70. Shaft failure from improper hub fit

Bolting

Bolts are used in couplings to connect flanges and to connect disc or diaphragm packs to hubs and spacers. Some coupling designs use close clearances on the bolt diameters to maintain concentricity between parts. Coupling bolts are subjected to bending, shear, and torsion. When bolts are tightened incorrectly, they are more likely to fail in fatigue.

Torque is transmitted between coupling flanges by means of friction between flange surfaces and shear of the bolts. For lubricated couplings where a gasket is used between flanges, almost all of the torque is transmitted by the bolts, since the coefficient of friction between the two flanges is very low. For non-lubricated couplings, the coupling is designed so that most of the torque is transmitted by the friction between flanges. This means that proper pre-tensioning of the bolts to maintain the clamping force is extremely important for coupling reliability.

Couplings are supplied with high grade fasteners, commonly with yield strengths greater than 100,000 psi. Bolts for high performance, high speed couplings are usually supplied in weight matched sets, with the weight of each bolt and nut assembly held to within 0.1 gram tolerance. The diametrical clearance around coupling bolts is usually tighter than standard clearances. These tighter tolerances help maintain balance quality after repeated disassembly and reassembly.

Coupling bolts are usually sized so that the initial tightening torque stresses the bolt to 80% of the yield strength. The high initial pre-tension of the bolts means that the clamping force keeps the flanges together and makes the joint less sensitive to fatigue failure from alternating stress. High performance couplings usually have bolts and nuts that have integral washer heads to minimize indentation (and loss of pre-tension) of the bolts into the flanges.

Disc and diaphragm packs are usually supplied pre-assembled with the bolts pre-tensioned. Unless the coupling design requires that the disc pack bolts be removed for spacer installation, disc packs should not be disassembled in the field. The bolts in disc packs are subject to shear from torque and bending from misalignment (flexing of the disc packs). The maximum combined stress occurs at the flange surface, where most disc coupling bolt failures are from bending.

Distance between Shaft Ends

Coupling assemblies are designed based upon a given distance between shaft end (DBSE). It is rare that machines are set to the exact DBSE as specified on the coupling assembly drawing. Likewise, hub position on the shafts is almost never exact due to mounting and manufacturing tolerances. Gear couplings have a certain amount of DBSE tolerance built in to the coupling design because the gear shrouds have longer teeth than the hubs. Flexible element couplings, like disc or diaphragm designs usually are supplied with spacer shims to adjust the spacer length.

DBSE should be measured and compared to the coupling assembly drawing to verify that the dimension is within tolerance for the coupling design and size. If the DBSE is out of tolerance, one of the machines may need to be moved. The coupling assembly drawing should also be checked to see if any axial offset is required to account for axial thermal expansion in the cold condition.

Spacer Installation

Most coupling assemblies are match marked, and the match marks need to be lined up during spacer installation.

Installation of spacers for grease-lubricated couplings requires the gear teeth be packed with the correct coupling grease before assembly. For couplings with hollow spacers, the spacer bore needs to be isolated from the grease packed area. Coupling manufacturers usually supply isolation plates (with gasket or O-ring) that fit in counter bores to keep the grease in the proper location. Grease packed coupling assemblies require gaskets made either of grease-proof material or O-rings at the flanges to keep the grease from leaking out between the flanges. Flange surfaces, pilots, and counter bores should be checked for nicks or burrs that could keep the flanges from making up correctly when tightened.

Continuous oil lubricated gear couplings should have the gear teeth coated with a light, oil soluble lubricant during installation. It is very important to verify the positions of the oil supply nozzles in the coupling enclosure before the enclosure is sealed.

Spacers for disc and diaphragm couplings are often piloted in counter bores located at the face of the hub or flex element pack. Because of this, the spacer is somewhat longer than the distance between hubs or flex-element packs. Most coupling manufacturers use jacking bolts to compress the flex elements axially to allow spacer installation. Care should be taken so that the jacking bolts do not bear on the flex elements, as this will damage the flex elements. The amount that the flex elements are compressed should be measured so that the allowable axial compression (listed on the coupling drawing) is not exceeded.

Maintenance

Gear Coupling Maintenance

To mitigate any gear coupling problem frequent inspection is required until a pattern of good running and maintenance is established. The most accurate way to inspect gear couplings in the field is by performing a span check on the external hub teeth and a pin check on the internal sleeve teeth (Figure 71and 72).

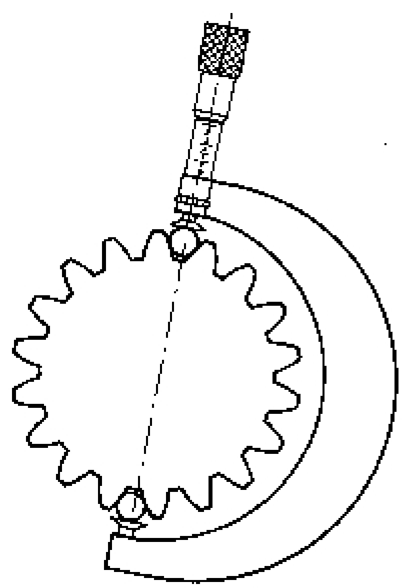

Figure 71. Pin Check

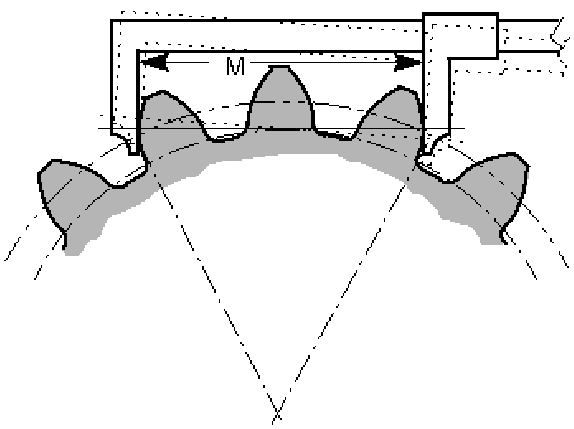

Figure 72. Span Check

By taking tooth measurements during consecutive maintenance intervals, a wear rate can be established and a replacement interval determined. By rotating the sleeve relative to the hub and using an indicator or other instrument, the total movement or backlash can be measured. This can then be compared to the recommended limits of the coupling manufacturer.

If a dimensional inspection is not practical, the teeth should be visually inspected for any unusual wear, such as: pitting, spalling or the development of a worn step on the tooth flank. Additionally, an analysis of the lubricant can be performed to determine the quantity and characteristics of wear particles.

Finally a lift check can be done, which gives an indication of how far out of balance the coupling is or could be while running, as with tooth tip clearance, part of the coupling can run off the center of rotation. The tooth tip clearance can be determined with a dial indicator on the top of a sleeve with the coupling at rest. Then the sleeve is "lifted" at the bottom, with a lever or by other means, and the indicator shows the amount of tooth clearance. This value should be checked against the manufacturer's allowable figures to determine if excessive wear has occurred.

Disc Coupling Maintenance

It can be difficult to determine the condition of a disc coupling while in service. Using a strobe adjusted to the disc coupling's revolution, if the discs are exposed, it's possible to examine the disc packs for signs of buckling, spreading, or broken discs (Figure 73 and 74).

Figure 73. Spread Discs

Figure 74. Disc Breakage from Angular and Parallel Misalignment

Extreme caution should be used when investigating a rotating coupling under a strobe light and is not recommended without adequate personnel protection. Cracks and or severe spreading in two consecutive links is a sign that the disc pack should be replaced. If a strobe cannot be used, vibration monitoring and careful inspection at every shutdown should be used to understand the condition of the coupling.

Diaphragm Coupling Maintenance

Diaphragms should be inspected periodically to prevent unscheduled downtime and premature failure. Nearly all diaphragms will have a guard which acts to protect the diaphragm from scratches and also act as an anti-flail feature, to keep the connected center section from “flailing” in the event of a severe diaphragm failure, and eventually breaking through a coupling guard. To inspect the diaphragm, the guard must be removed. Then look for any cracks, severe scratches, or distortion. Checking the runout at the rim of the OD can verify that the diaphragm is free of distortion.

MITIGATION TO LIMIT EFFECTS

Detection of Failing or Degrading Couplings

We would like to presume that vibration interlocks can always provide adequate shut down turbomachinery prior to a catastrophic coupling failure, but this is not always true.

Torque amplifications and shocks

Torsional critical speeds are rarely possible to detect with conventional instrumentation after a machine is put in service, so these risks should be resolved at the design and test phase, Corcoran, et al. (2010).

Two cases in this paper discuss impact torque shocks from process liquids and solids that fractured disc packs. In each case, the speeds were high enough for the off-center coupling spool piece for unbalance shaking forces to mitigate the risk by automatic shutdown with vibration interlocks. But lower speeds require higher spool run out for vibration to trip.

Torque from an out of phase generator breaker closure as in Case 8 is so extraordinarily high, that the only possible way to prevent failure is to avoid the initiating event. Though rare, out-of-phase reclosing into residual field of a coasting motor such as Mulukutla and Gulachenski (1992) can also produce torque levels at 12X the design torque, high enough to shear shafts as well as couplings. Most modern breaker controls prevent this risk, but legacy equipment may be vulnerable.

Off-center spool unbalance shaking forces

If the flexing element of a coupling fails, the floating spool piece will move off center by an amount determined by the coupling design. At high RPMs most couplings will produce enough unbalance force for an immediate vibration interlock. But with low RPMs or light couplings on heavily loaded journals, the response may not trip vibration interlocks until more damage has been done.

The method below is to compare various ratios of floating spool mass to the rotor journal loads, then calculate the spool run out needed to produce a shaking force of 10% of the journal load (Figure75). Obviously many rotors are more sensitive to unbalance, especially at the coupling hub, so this method attempts only to provide a means to quickly identify coupling and rotor combinations which might not produce enough unbalance forces to quickly shut down on high vibration. Several cases in this paper demonstrate that couplings running above 6000 RPM were successfully shut down with high vibration interlocks and retained the spools.

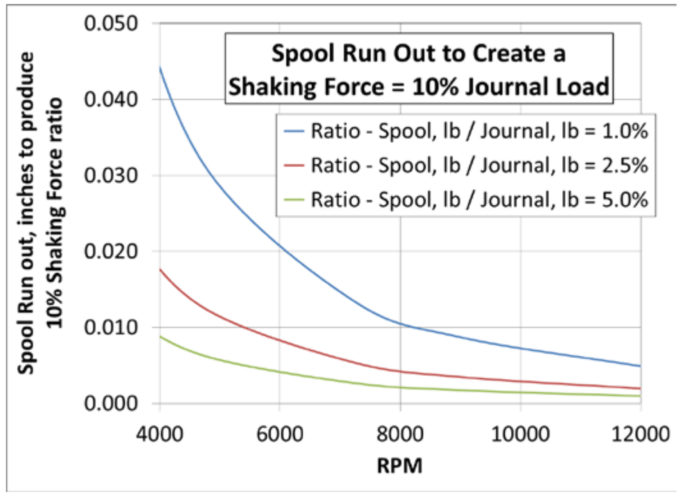

Figure 75. Coupling floating spool run out to produce 10% shaking force on a rotor at high RPMs.

Lightly loaded coupling end rotor journals such as an overhung compressor in Case 14 at 8300 RPM were very easily disturbed by a failing coupling and shut down. A beam type compressor coupling end bearing has more load, but the Case 9 beam compressor was manually shut down safely with velocity probes prior to catastrophic failure (Figure 76).

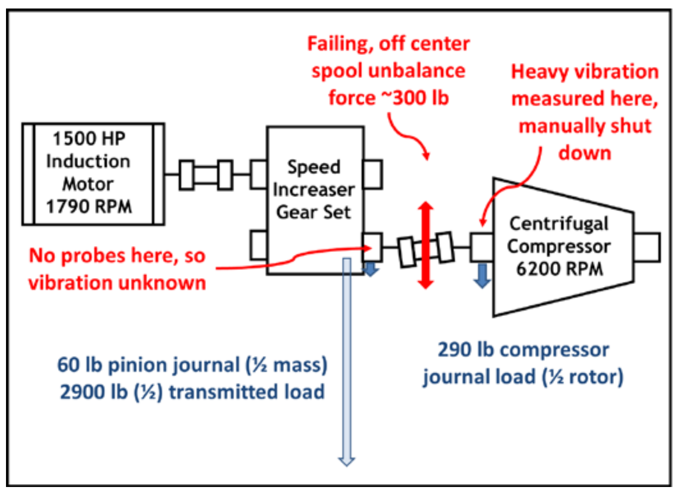

Figure 76. Case 9 compressor gear coupling damaged mesh on pinion side, picked up on compressor vibration

The example described in Case 9 had a failing gear mesh adjacent to the pinion bearing, but no installed vibration probe. The compressor bearing had a velocity probe, which picked up the failing gear coupling from the opposite side where the gear mesh was still in good condition.

Machines with relatively light coupling spools or slower speed machines could be at greater risk because a larger coupling spool run out is required to generate the 10% shaking force (Figure 77). At least two compressors at DuPont have been identified where a failing coupling operating at a lower speed did not generate enough unbalance for an automatic trip at normally guide-lined vibration levels.

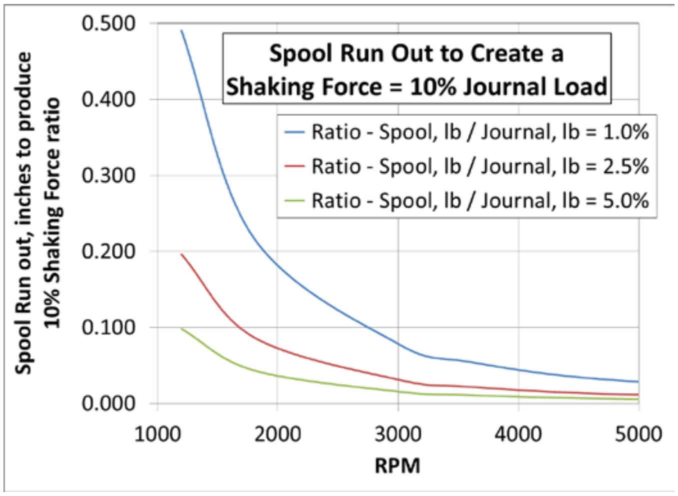

Figure 77: Coupling floating spool run out to produce 10% shaking force on a rotor at low RPMs.

Coupling Guards

Coupling guards are designed to prevent someone from entangling their clothing or hands, etc., in rotating equipment. They are NOT designed, in most cases to prevent major failed coupling pieces from being ejected through the guard. The industry depends on coupling design, etc., to prevent catastrophic incidents.

Through the use of the empirical THOR equations in combination with laboratory testing (Figure 78) it has been shown that coupling guards can be designed to contain a coupling nut and bolt fragment. The analysis method uses the fragment kinetic energy upon release and compares the result to the energy absorbing capability of the guard material to determine if the fragment will be contained by the guard or if it will penetrate the guard. The method also estimates the kinetic energy of the fragment after penetration.

Figure 79 shows the kinetic energy of a 90 gram nut and bolt fragment versus shaft speed and the energy absorbing limits of two different steel plate guard thickness.

The analysis method has been used for both solid metal and perforated metal guard materials (Figure 80).

Having the ability to contain a coupling nut and bolt fragment with an appropriately designed coupling guard is beneficial as a means to mitigate a bolt failure because there is no other line of defense for this failure mode. A vibration monitoring system will only sense the resulting unbalance after the failure, and then it is too late.

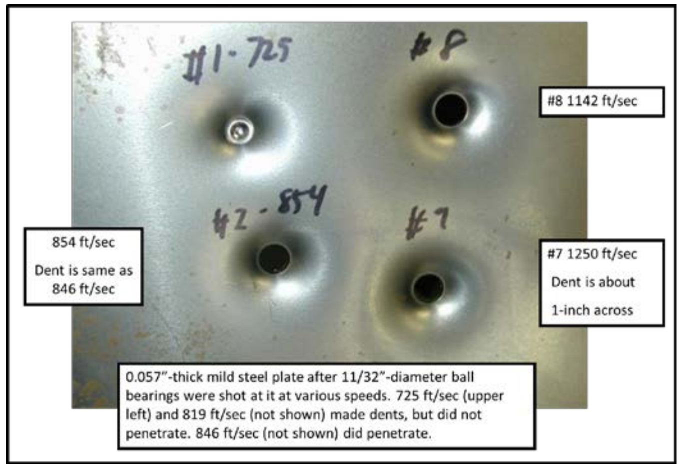

Figure 78. Target plate fragment penetration testing

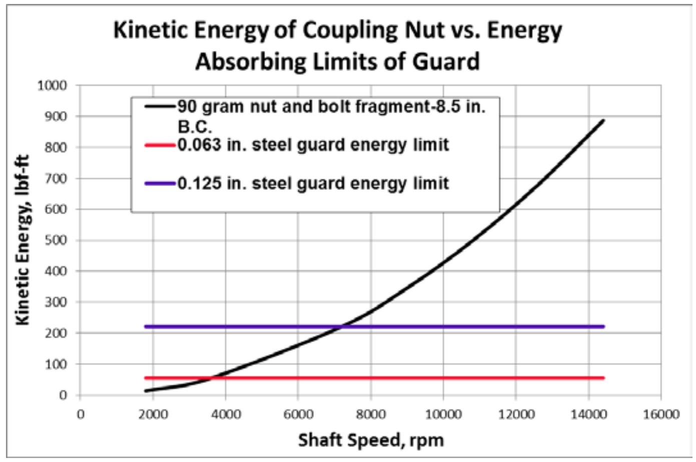

Figure 79. Coupling Guard versus a coupling nut

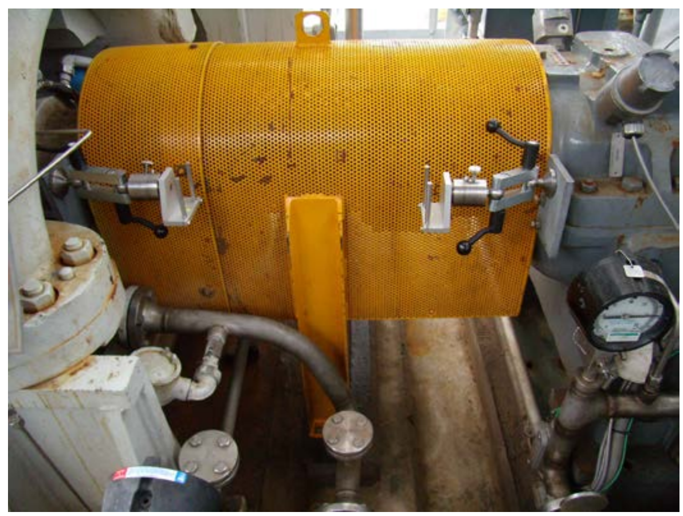

Figure 80. Perforated metal coupling guard

However, containment of a failed coupling component any larger than a nut and bolt fragment is not practical because the kinetic energy of the failed component is too great. Figures 81 and 82 show the results of a coupling separation. Not only did the guard not contain the coupling but he base did not retain the motor.

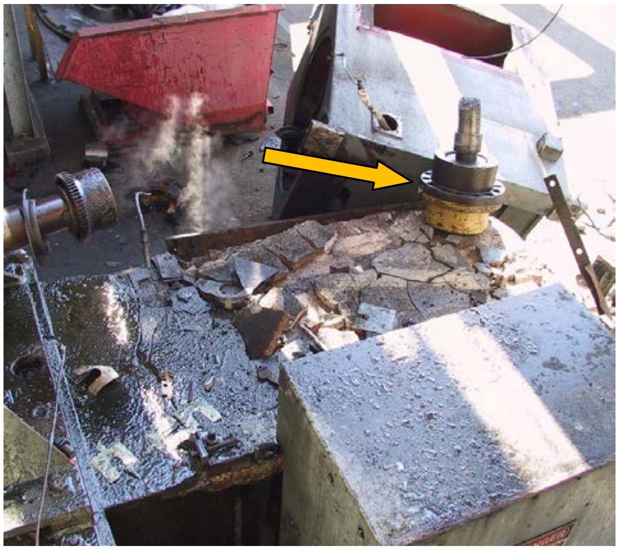

Figure 81. Coupling Guard versus a flailing coupling

Figure 82. Close up of fractured motor shaft

There are two general categories of coupling guards: open guards and closed guards. Open guards are designed to keep personnel from harm by contact with rotating elements. Perforated guards such as the one shown in Figure DD can be designed to retain a failed nut and bolt fragment and still permit use of a strobe light to monitor the condition of the disc packs while in operation. Closed guards offer protect personnel from direct contact with the rotating elements and also collect possible oil leakage from machine seals and drain the oil to the appropriate area. Most coupling guards are <u>not</u> designed to retain flying debris in case of a catastrophic coupling failure.

CONCLUSIONS

Catastrophic coupling failures can be prevented with adequate consideration of the following factors:

- Proper coupling selection include the coupling type selected, machine demands and system risks for selection of the service factor and the materials of construction for the specific need.
- Proper installation and maintenance require quality training in best practices for fitting, assembly, lubrication, and alignment. Further, shaft alignment can only be adequately maintained with good foundations and control of piping loads. Don't use substitutes for OEM fasteners and honor finite life of fasteners.
- Vibration probes and systems applied need to account for the specific needs of the machine with realistic configurations that can respond in time. Instrument systems need to be maintained and tested with well-planned, periodic verification.
- Anti-Flail designs must retain a coupling spool long enough for other layers of protection to shut a machine down safely while accounting for the likely position of the shafts during a failure.
- Guards are primarily intended to keep people from touching moving parts. Containing an entire spool piece is, in general, not realistic and we can benefit from multiple layers of protection. Some limited bolt containment is possible. Using open mesh or perforated guards provides more cooling as well as enabling on-line strobe examination.
- Fail-Safe designs may be required if, for example, large torque shocks are likely to occur during the life of the machine, or where the consequences of even very rare events are especially severe.
- Torque monitoring during initial machine start up can avoid the uncertainty of calculated torsional critical speeds. Continuous torque monitoring can offer additional protection against operating near a torsional critical which may not be detectable with conventional vibration systems.
- Shear devices provide another option to deal with extraordinary torque shock loads which may not be avoidable for a particular system.

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