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PREVENTING UNDETECTED TRAIN TORSIONAL OSCILLATIONS

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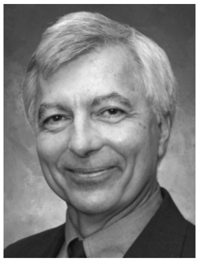

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ABSTRACT

In the past few years the authors have encountered numerous problems and coupling failures related to unexpected torque fluctuations on applications with reciprocating compressors and on VFD, synchronous motor, and gas engine driven trains. In some cases, dangerous coupling failures have resulted causing significant lost uptime, and requiring a tremendous amount of engineering resources to discover the root problem and the corrective action. Since almost all modern turbomachinery is outfitted with proximity probes to detect lateral vibrations, but not with probes or systems to detect torsional oscillations, these failures occurred suddenly and, in some instances, without warning.

In this paper, some of these cases and the work done to deduce the causes of these failures are discussed. The various methods used to measure the oscillations and the advantages and disadvantages of each are presented.

INTRODUCTION

In the past few years, numerous problems associated with torque fluctuations have been encountered by the authors. These have occurred on applications ranging from reciprocating compressors, variable frequency drive (VFD) and synchronous motors and gas engine driven trains. In some cases, dangerous coupling failures have resulted causing significant downtime, requiring large amounts of engineering resources and posing a potential for safety related incidents. As opposed to lateral vibrations where proximity probes, accelerometers and velocity meters are used to monitor levels, few if any, applications monitor torsional pulsations on a regular basis. This has led to failures occurring in some cases suddenly and without warning.

These factors, sudden failures, long downtimes and potential safety incidents, have raised the importance of recognizing turbomachinery trains susceptible to torsional oscillations. In this paper, several cases and the work done to deduce the causes of these failures is discussed. The various methods used to measure the torsional oscillations and the advantages and disadvantages of each are presented.

TORSIONAL BEHAVIOR

Train configuration can play a role in determining the likelihood of torsional pulsations. Multibody and geared trains have increased likelihood of suffering from problems due to the increased number of torsional natural frequencies. However, a majority of failures are associated with excessive torsional pulsations or excitations and not the train complexity. These pulsations can arise from reciprocating machinery (compressors or engines), synchronous motor starts, or use of VFD motors control and conversion technology.

In several of the examples presented, VFD motors play the integral role in the coupling failure. Recent trends in the oil and gas industry are toward higher power density trains and greater adoption of VFD motors. This is particularly evident in the liquid natural gas (LNG) business where VFDs are used in both the primary refrigeration trains and off-plot compression trains. The size, both physical and horsepower, of these VFD inverters have kept pace with the growth in size of the LNG plant. Each inverter type has a particular torsional behavior and excitation associated with it. The design of modern systems for large LNG trains is described by Baccani, et. al. (2007).

Numerous methods exist to model and predict torsional behavior in turbomachinery trains. American Petroleum Institute (API) 684 (2005) and Corbo and Malanoski (1996) describe many of the methods and practices in general and those specific to many types of turbomachinery. These include transient analysis as specified in API 617, Seventh Edition (2003). Depending on the train configuration, this may involve using nonlinear analysis techniques.

Even with the advanced analysis and techniques, failures continue. Uncertainties in the damping present in the torsional system, excitation magnitude and the interactions in the electromechanical system due to VFD control algorithms are some of the reasons why. For typical oil and gas industry compressor trains, the level of damping available to the torsional system may be limited to the material damping due to internal friction. Industry typical values range from 0.6 percent to 1.7 percent of critical damping. These small changes in damping, while minor compared to the range of damping available to lateral systems, may cause torsional response levels to change dramatically effectively eliminating any useful life in the train. The uncertainties persist due to the lack of readily available data to verify torsional predictions, modeling assumptions used and the understanding of interactions between the electrical and mechanical systems.

As previously noted, torsional problems can be dangerous. With the lack of consistent monitoring in a system of lightly damped natural frequencies, failures can occur with little or no warning with relatively small excitations. Even with geared trains, where torsional oscillations can be seen as lateral vibrations on either gear or pinion or gear tooth rattling, the relationship between the lateral and torsional magnitudes is seldom known. Kita, Hataya and Tokimasa (2007), have presented an initial attempt to predict that relation. While the analysis procedure is based on an actual test stand measurement of the lateral vibrations resulting from measured torsional pulsations, application to other gear configurations remains to be tested.

Large torsional oscillations may involve stress reversals and can occur quickly upon startup of the train. At 3600 cpm, low cycle fatigue (1000 cycles) can occur in ≈16 seconds. Endurance limit (1,000,000 cycles) can be reached in ≈4.5 hours. The cyclic failure frequently occurs at the coupling, many times the weak link in the train by design. However, torsionally induced maximum stresses may not occur in the flexing element. Failures in other components of the coupling may release parts. With coupling guards not designed to contain release of these parts, personnel safety can be at risk.

As with any unexpected failure, downtime of modern turbomachinery trains can be expensive. The costs are incurred not only by the user for lost production, but also by the vendor for significant engineering/service support and priority manufacturing of replacement components. However, the majority of torsional problems are not caught in "typical" mechanical testing of train machinery. Unless full load testing of the entire train is performed, a costly option with significant project schedule impact, the torsional behavior of the train is not truly tested until startup of the machinery for operation. Air runs or partial load operation may not show the problem in significant magnitude to draw attention, leaving the failure to occur just when new or revamped equipment is put into service.

Even methods to prevent failures, as noted by Feese and Hill (2001) for reciprocating machinery has its uncertainties. Beyond tuning of the torsional train to remove interferences between natural frequencies and excitations, one popular method is the introduction of damping into the system through implementation of an elastomeric coupling. However, complexities with these designs exist based on their nonlinear behavior with torque and degradation of properties over time. Careful analysis of not only the machinery train but also the operating environment is needed when using these couplings.

When an elastomeric element coupling or device is required to tune an equipment train away from torsional resonances, or even to dampen the magnitude of oscillations, it is because the torsionally soft material (compared to steel) either moves the resonances away from running speeds, or dampens the vibration through hysteresis damping, which converts the vibratory energy to heating of the material. However, there are some drawbacks to using elastomeric elements for these purposes.

- Elastomers, urethane, natural rubber, SBR, nitrile rubber, Viton® and others are not stable material.
- Many have a shelf life and get harder over time.
- There is a tolerance on the hardness (durometer), affecting stiffness values.
- Torsional stiffness varies with torque and temperature (Figure 1).
- Elastomeric materials are sensitive to oxygen (ozone), light, water, hydrocarbons, and various other agents.
- These materials have a vibratory torque capacity, sometimes missed, related to temperature.
- If used to dampen a resonance, the internal heating of the material must be taken into account.
- If it is the block element style (Figure 2) they can be difficult to install, especially for couplings over 24 inches in diameter, unless a silicone fluid or other low friction inert substance is applied liberally, and hydraulic tooling is used.
- Because of all of the above, the condition of the material should be checked regularly, at least once a year, thereby increasing maintenance costs.

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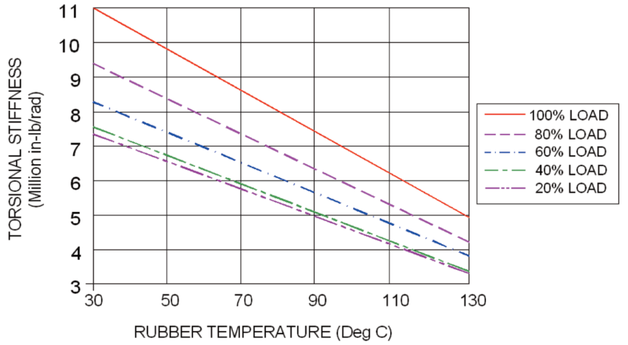

Figure 1. Graph of Elastomeric Coupling Stiffness Versus Torque and Temperature.

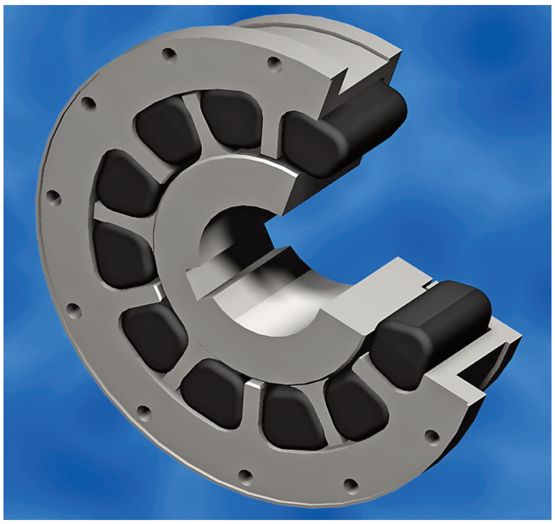

Figure 2. Block Element Style Coupling.

Even with these limitations, these type couplings are frequently used, and in most applications these couplings run fine with the proper maintenance checks, sometimes for periods up to 10 years. It is the times when they do not run well that cause significant problems. Steps can be taken to improve life, like ozone resistant additives, and shelf life protection for spares (e.g., sealed opaque containers).

But if this type coupling is causing downtime issues, one can attempt to carefully design a standard diaphragm or disc coupling to do the job, but this requires special materials and more advanced torsional and stress analysis. In Table 1 is a comparison of properties of a case (Mancuso and Corcoran, 2003) where a special diaphragm coupling replaced an elastomeric resilient coupling on a synchronous motor train. This potential change-out is not guaranteed, just a possibility. The system in the end may still require the damping from one of these type couplings, especially if expensive options like soft start electronic controls or a redesigned drive train are not feasible.

Table 1. Comparison of Elastomeric Hybrid Coupling Properties with the Special Diaphragm Coupling that Replaced It.

Coupling Type	Driving OD (in)	Driving Half Weight (lbs)	Driven OD (in)	Driven Half Weight (lbs)	Max Start-Up Torque (million lb-in)	Coupling Transient Torque Capacity (million lb-in)	Axial Misalign. Capacity (in)	Angular Misalign. Capacity (deg/end)
Resilient Disc Hybrid	26.88	509	41.75	3546	4.00	4.26	+/- 0.200	0.25
Marine Style Diaphragm	27.41	610	27.41	911	5.35	6.60	+/- 0.080	0.17

CASE STUDIES

Case Study Notes:
1) In the case studies below, the authors have presented the facts as they know them, after reviewing gigabytes of analysis, reports, e-mails, photos, etc. There is no attempt to assess any blame to any of the many parties involved in the failures presented. In each case, all or nearly all the parties made mistakes, minor or major, and all parties paid a significant price financially. These studies are presented strictly to educate the industry to avoid these failures.
2) Only seven studies are presented, but there could have been many more that are not presented due to time limitations. For example, problems related to synchronous motor transient start-up torque oscillations are not covered.

Case Study 1

On a pipeline gas transmission train three couplings (two different types) failed before it was realized that the torsional analysis was incorrect, and that instead of the train running away from any resonances, it was running directly on one at operating speed. The train was relatively complicated, consisting of an induction motor (1200 hp at 1150 rpm) to a jackshaft (with couplings on either end, and bearings in the middle) to a four cylinder reciprocating compressor. Attached to the jackshaft was a belt-driven pulley, driven by a diesel engine, for alternate power. A starter motor was part of the train also (Figure 3). In the actual train where coupling failures occurred, the pulley was disconnected. The original analysis predicted a sharp resonance at 740 rpm (Figure 4).

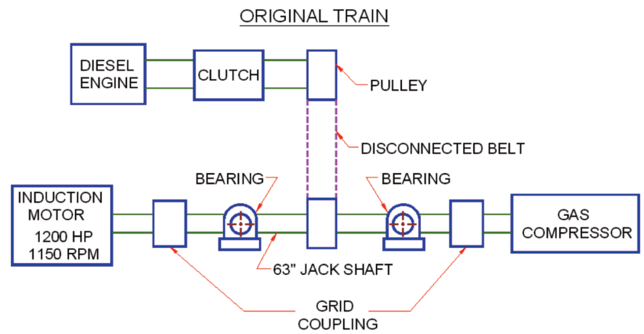

Figure 3. Case 1 Train Layout.

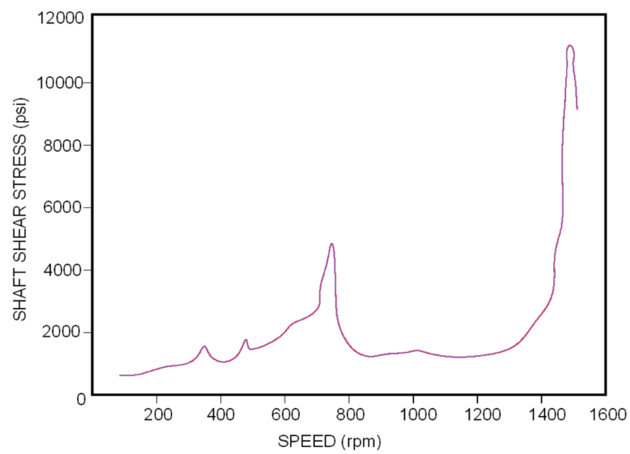

Figure 4. Case 1 Predicted Resonance.

So when excessive vibration (and noise from the coupling closest to the motor) shut down the train at start-up, it was felt that if the train could get past that critical, it could operate at normal speed. Arrangements were made to bypass the trip and run.

This was done. The first coupling, a grid type design, lasted one hour (Figure 5). Since it was likely the coupling had already been damaged during a few start-up attempts previously, and that is why it failed so quickly, a second coupling of the same size and type was installed and an attempt to get past the resonance was made. It was done, and this coupling failed.

Figure 5. Case 1 Failed Grid Coupling.

In previous similar applications a disc type coupling had worked. A larger disc coupling (larger to match the torsional stiffness of the grid coupling) was manufactured on an emergency basis and installed. It lasted one hour.

The original torsional analysis was reviewed, and it was found that the train without the pulley connected was not modeled properly. When done, it was found that the normal running speed was right on a different, more violent critical.

The final solution to move the criticals out of the speed range were flywheels added to the compressor shaft, and an even larger disc coupling installed, supplied, unfortunately for the original coupling supplier, by another coupling manufacturer.

What could have been done to reduce the excessive cost and downtime caused by this problem? Unbeknownst to the coupling supplier, the torque oscillations were calculated to be at 51 percent of the coupling continuous rating at the originally predicted critical, a fact that should have been reviewed with the coupling vendor. A red flag could have been raised at this point.

But even then the problem might not have been found, because of the incorrect torsional analysis. The best thing would have been to instrument and measure the torque oscillations at the first sign of trouble.

Case Study 2

A 21 inch diameter single flex disc coupling connecting a four cylinder gas engine quill shaft to a bull gear driving a reciprocating compressor at a gas pumping station failed after 231 hours of operating (Figure 6). The coupling transmits 3,700 hp at 480 rpm and replaced a larger 23 inch forged steel gear coupling.

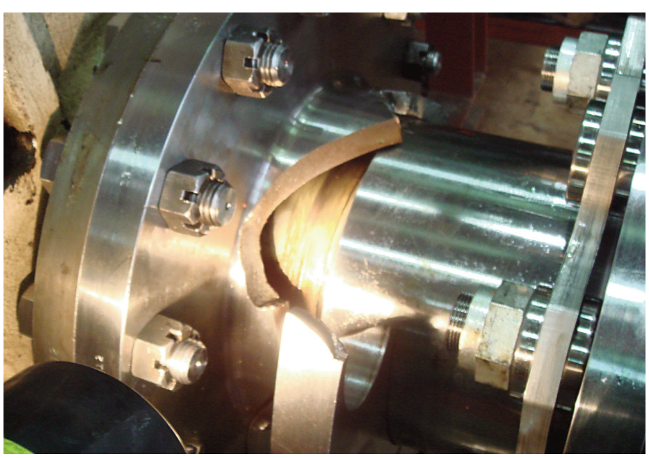

Figure 6. Case 2 Failed Disc Coupling.

The failure of any disc pack adapter flange is uncommon and was caused by torsional vibrations. Under normal continuous unidirectional loading, the adapter is not subjected to high stresses, but in this application there was a high reversing load as evidenced by the pattern of fretting of the disc pack washers on the flange faces (Figures 7 and 8). Beach markings on the fractured surfaces (Figure 9) indicate that initial failure occurred as a result of fatigue, and final fracture was a result of overstress. Metallurgical analysis showed that the material was acceptable per the applicable specifications. The disc packs were not extensively damaged, as they have a high fatigue life, mostly to accommodate misalignment bending loads, but which also helps them take high vibratory torsional loads. There was also extensive damage to the thrust bearing and shaft (Figure 10).

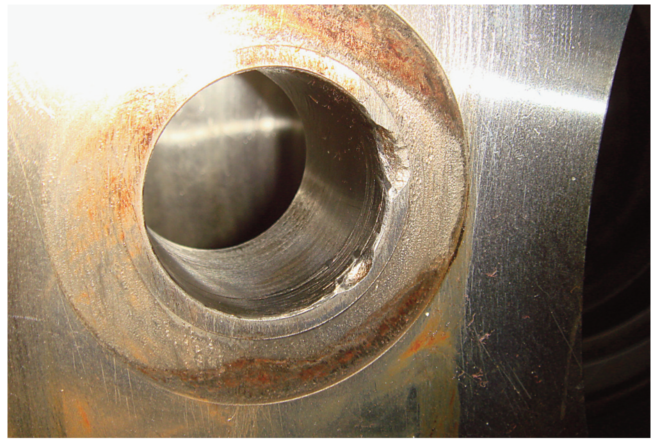

Figure 7. Case 2 Fretting on Disc Pack Flange.

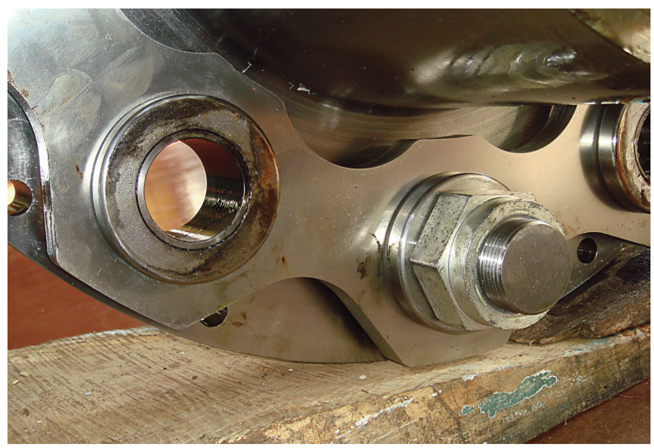

Figure 8. Case 2 Fretting of Disc Pack Washers.

Figure 9. Case 2 Beach Marks at Fracture Site.

Figure 10. Case 2 Shaft Damage.

The failure was noticed after the machine tripped on high bearing temperature. *It occurred suddenly and without warning from vibration probes*.

A check of the torsional analysis done for the newer low maintenance disc coupling was said to have been done, but was not reviewed with the coupling supplier. So the only things that could have been done to prevent this failure were:

- The coupling manufacturer could have insisted on reviewing the torsional analysis, possibly then picking up a problem, and
- Actual torque oscillation measurements could have been taken during the initial start-up after the disc coupling was installed.

A faulty engine fluid damper, which had degraded over time, was eventually cited by the equipment owners as the likely cause for the high magnitude torsional oscillations. The fluid damper color was dark indicating the need to be changed. The old fluid was replaced and the gear coupling has been reinstalled and is still running to this day, albeit with frequent (90 day) greasing, which is somewhat of an inconvenience.

Case Study 3

All of a disc pack coupling's disc pack attachment bolts failed on the engine end of a gas engine driving a reciprocating compressor at 2510 hp at 900 to 1000 rpm, after only three hours of total run time at the compressor manufacturer's facility (Figure 11). In addition, the disc pack failed (Figure 12). The three hours included a number of smaller runs over a period of three days. There was no load from the compressor; the only load was parasitic.

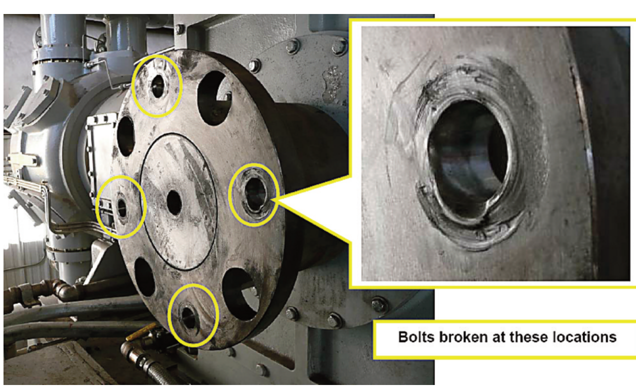

Figure 11. Case 3 Disc Pack Flange Bolt Hole Damage.

Figure 12. Case 3 Disc Pack Damage.

The conclusion of the compressor manufacturer was that these bolts were never torqued and were loose, and they were able to confirm this by checking quality records and noticing that there was no permanent paint put on the fasteners as an indication that they were torqued. The coupling manufacturer agreed.

But, this was not a typical loose bolt failure. There was a lot more going on as evidenced by the classic beach marks on the broken bolts (Figure 13) and the two direction (yielded) disc pack holes indicative of reversing torques.

Figure 13. Case 3 Beach Marks on Failed Bolts.

In the torsional analysis, some concern was expressed by the analysis company regarding the modeling method associated with the process. Field verification of predicted torsional critical speeds was recommended and should have been done. It is not known by the authors whether this has yet been done. This situation is still under investigation.

Case Study 4

In yet another expensive problem resolution, a shear pin coupling between a gearbox and balance machine at a gas turbine manufacturing facility kept having premature shear pin failures, significantly affecting production. The balance machine was driven by a VFD controlled induction motor through a gearbox at 13480/13480/8050 hp at 330/1500/2400 rpm, depending on the size of rotor to be balanced. The pins were an overload protection against a motor fault. On the other end of the balance machine was a brake, to reduce the rundown time of up to five hours between balance runs had there not been one.

The pins were failing in fatigue, so since bending moments from high misalignment could have contributed, first the machine alignment was checked and verified to be good. Since a VFD was involved, a torsional vibratory load was suspected. Oddly enough, this coupling (Figure 14) had been outfitted with a torquemeter, but this particular device was designed to measure steady torque loads, not high frequency torque pulses or quick spikes, so the torquemeter did not (could not) give any indication of fluctuating torque.

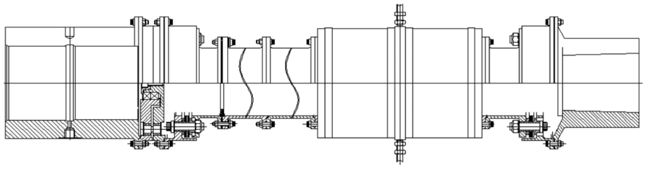

Figure 14. Case 4 Shear Pin Coupling with Torquemeter.

A torsional analysis was not seen or given to the coupling manufacturer. Whether one was ever done or not, the logic of the packager who put the train together was that the coupling manufacturer was given the peak torque of the application, and this torque should have been assumed to be fully reversing. Therefore, no matter what oscillatory loads were predicted by analysis, they would not have a magnitude higher than this. By the applicable engineering codes of the application site country, the packager was correct.

However, due mainly to communication issues, the code was not seen by the coupling manufacturer, nor did the coupling manufacturer understand that there was a brake in the train. To meet the code, and still have fault overload protection, meant a completely different shear device design. Plus, there would be needed torque oscillation measurements so that the device could be qualified to handle the suspected vibratory stresses without premature failure, while still providing fault overload protection.

The packager finally settled on a ball and spring overload device to solve the problem, though it is much heavier and of course more expensive.

Case Study 5

In February of 2007, a diaphragm coupling's diaphragm failed between a VFD controlled motor (1341 hp at 1040/1559 rpm) and gearbox driving a centrifugal compressor (Figures 15 and 16). The train had been operating on and off since July 2006. The failure was discovered when the train shut down on high lateral vibration. The failed diaphragm was on the gearbox side.

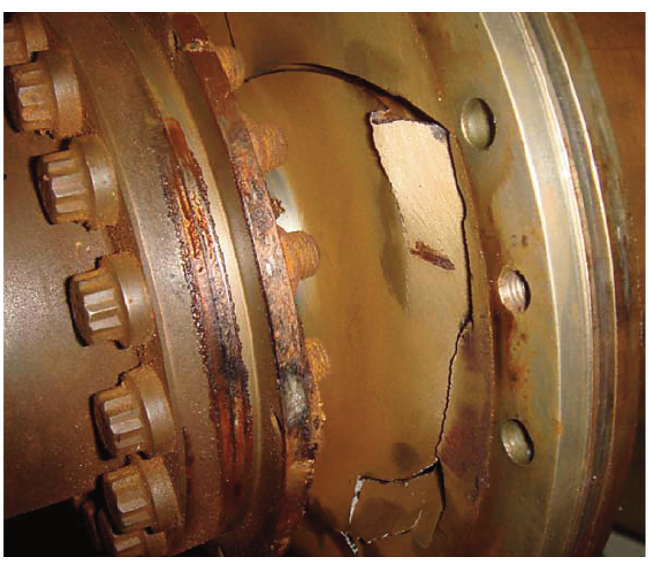

Figure 15. Case 5 Diaphragm Failure.

Figure 16. Case 5 Diaphragm Failure Opposite Side.

The diaphragm broken parts were analyzed by the coupling supplier, and the conclusion was that the diaphragm failed in torsional fatigue, as evidenced by the 45 degree crack emanating from the toroidal base (smallest diameter) of this integrally machined (nonwelded) diaphragm, and the condition of the crack surface (Figures 17 and 18). For this particular diaphragm, the base has the lowest resistance to vibratory torque loading, predicted by analysis. An independent test lab confirmed that the failure was from fatigue while also confirming that the material met design specifications. However, the torsional analysis for this train does not predict the loading that would cause this failure.

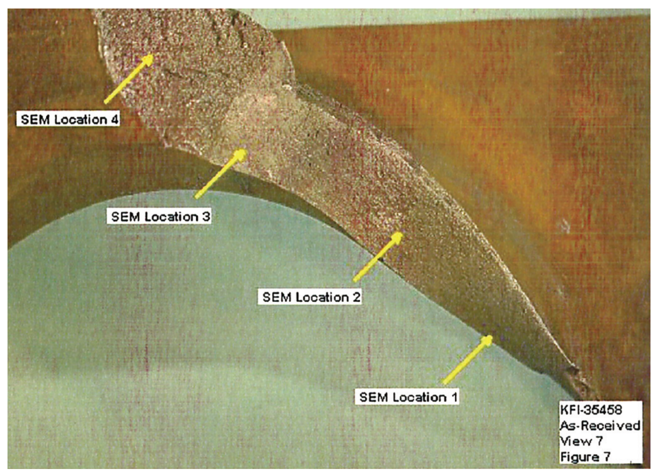

Figure 17. Case 5 Diaphragm Crack Magnified.

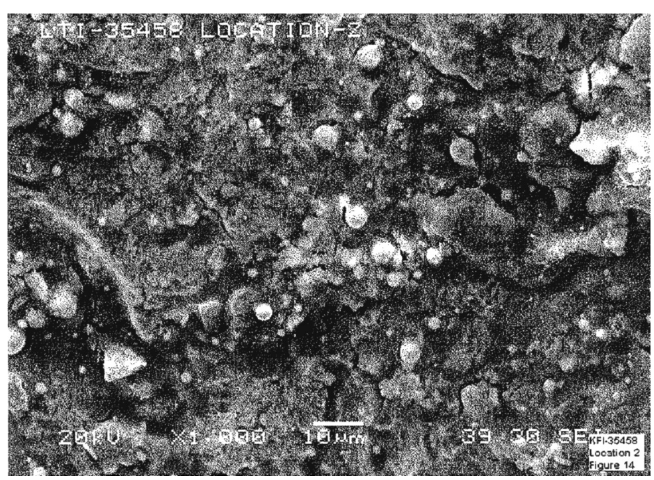

Figure 18. Case 5 SEM Photo of Crack Initiation Surface.

Nearly a year later, after the diaphragm was replaced on the gearbox side, the diaphragm on the motor end failed. The same conclusion was reached, that this second diaphragm failed in torsional fatigue, and again the material met specifications. Especially, no metallurgical discontinuities (laps, seams, excessive amounts of nonmetallic material, etc.) were observed in the coupling material that could have contributed to the observed crack initiation and/or propagation.

There, of course, is still some concern about the application as the oscillating torque required to fail the coupling was not predicted. The only way to be sure of these will be to actually instrument the coupling or connecting shaft with a device to measure the actual vibratory torques at all previously known operating conditions, and this has been recommended to the OEM and user. Again, this issue is still under investigation.

Case Study 6

This particular coupling failure took over one year to solve at the considerable expense of all parties involved. The failure occurred at a Middle Eastern LNG plant with equipment and engineering supplied by multiple international companies.

The train consisted of a VFD controlled induction motor (rated at 18305 hp at 1575 rpm) driving low and high pressure recycle gas compressors through a speed increaser. The high performance disc coupling between the gearbox and first (low pressure) compressor running at 6832 rpm failed catastrophically, with parts penetrating the coupling guard, after 2000 hours of service (Figures 19 and 20).

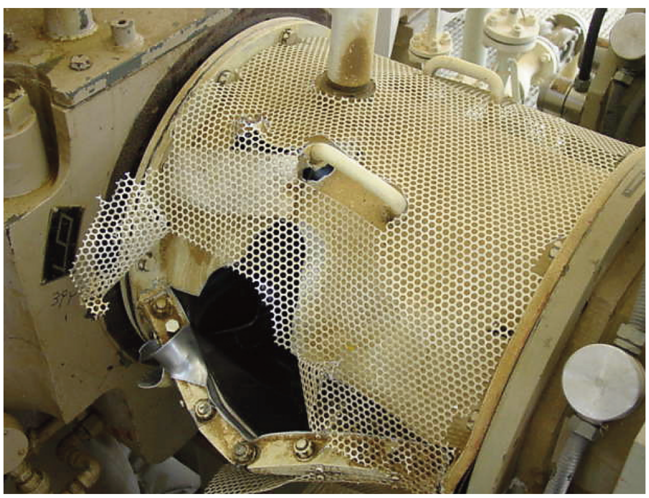

Figure 19. Case 6 Hole in Coupling Guard.

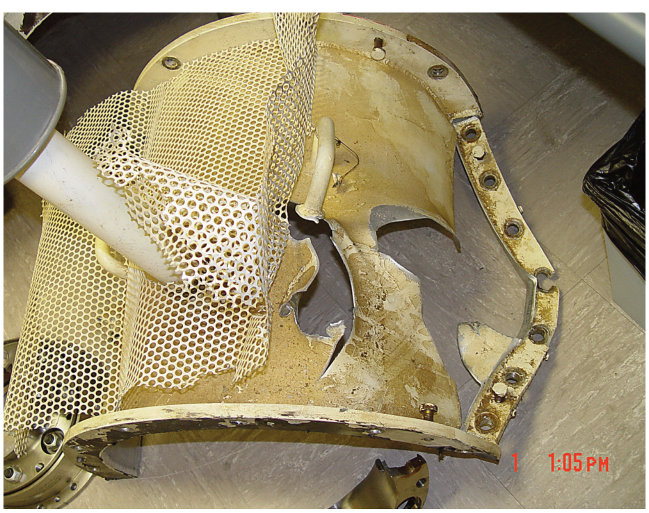

Figure 20. Case 6 Damaged Guard.

The inspection of the failed parts revealed that this coupling failed in torsional fatigue, as evidenced by the fatigue cracks and severe fretting between the disc pack bushings and washers and the mating flanges (Figures 21 and 22). Examination of the spacer tube revealed a spiral wound crack propagation typical of failures induced by torsional forces (Figure 23).

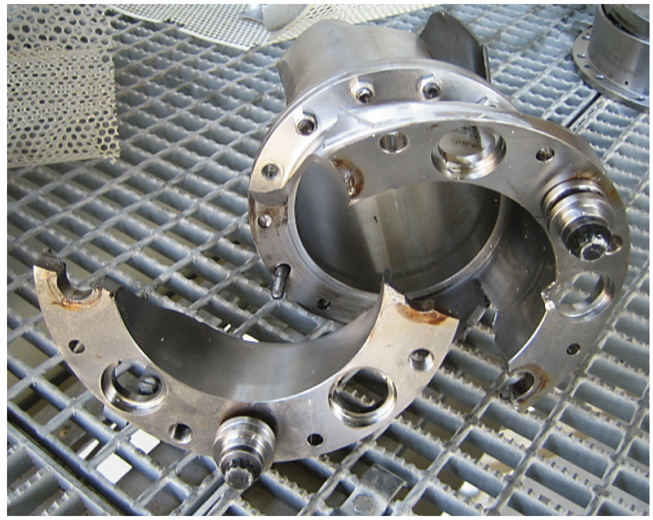

Figure 21. Case 6 Ejected Parts with Disc Pack Flange Hole Fretting.

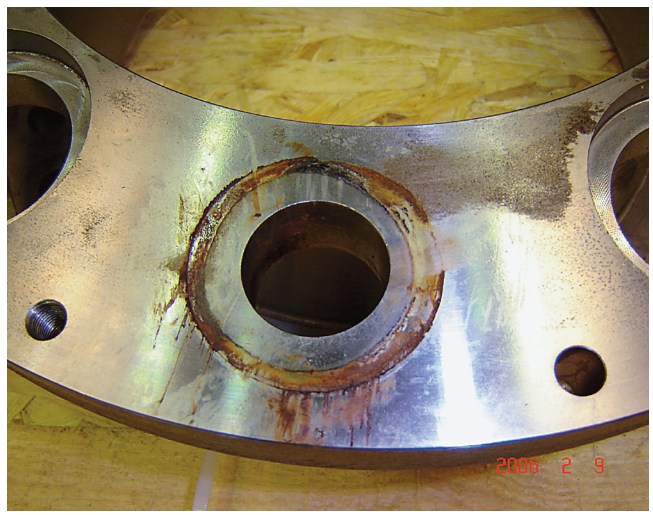

Figure 22. Case 6 Close-up of Disc Pack Flange Hole Fretting.

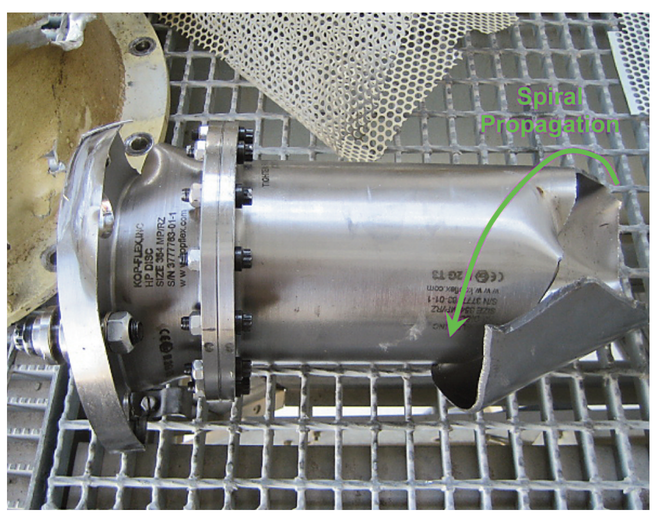

Figure 23. Case 6 Wound Spacer Crack.

Due to large economic losses from an unplanned shutdown of potentially several weeks to resolve the issue, the compressor train was put back into service with the spare coupling. Recognizing the safety and severity concerns of the coupling failure, several steps were implemented to mitigate these risks. A thick wooden catch was built around the problem coupling guard, and the area was deemed off-limits for personnel while the train was running. In addition, 500 hours following the startup of the train and every 1000 hours thereafter, the machines were stopped and all coupling parts were inspected. This continued until the root cause and solution were found at considerable expense (both time and money).

During a mechanical test of the entire train at the vendor shop, an unusual radial vibration and frequency were detected on the pinion shaft. The frequency of vibration matched predicted frequency of the first torsional natural frequency. From past experiences, the author diagnosed the problem as a VFD induced torsional pulsation. The gear mesh was hypothesized to be transforming the torsional pulsation into a radial vibration at the same frequency. Unfortunately, conflicting (and erroneous) data led the vendor to conclude that the VFD could not be the source. Subsequently, the train was shipped to the site.

After the coupling failure, finite element analysis (FEA) and lab testing with a strain gage instrumented coupling cycled on a dynamic torque applier was done. This however did not identify the reason for the coupling failure. Torque oscillation measurements were then done on the low speed shaft with a laser vibrometer. Once again, excessive vibratory torque was not found. Strain gage testing was repeated on the low speed coupling. These data were then used to calibrate the analytical torsional models to determine the magnitude of the torques at the high speed coupling between the gearbox and first compressor. A mix of measured and predicted data was used to infer the high speed coupling stress from low speed measurements assuming a torque pulsation originating at the motor. The test data calibrated model assumptions such as damping. Obviously, direct measurement of the high speed coupling would have proven more exact. However, owner representatives felt that this could not be achieved without undue risks to the machinery or site personnel.

One confusing fact of this failure was the apparent conflicting reports of high cycle fatigue (HCF) after 2000 hours of service. Over that period, the coupling would have experienced 190,000,000 cycles at the frequency in question greatly exceeding typical cycles necessary for HCF. What the low speed torsional measurements did reveal was the existence of VFD sidebands produced by the pulse width modulator. These sidebands are a function of the VFD output speed. Figure 24 displays the torsional oscillation of the train as one such sideband is coincident with and, then as speed is increased, moves away from the first torsional natural frequency (1TNF). When the two pulsations were coincident, it was found that the torsional oscillations produced exceeded the HCF stress level of the high speed coupling. Since HCF would happen only at specific operating speeds, this intermittent behavior explained why the failure did not occur sooner (closer to 1E6 cycles.) Torsional response levels of the high speed coupling were related to reexcitation of the 1TNF by the VFD control algorithm. In the end, the problem was resolved by installing a completely new algorithm to decouple the electrical and mechanical systems in play on the machinery train. Torque oscillation measurements had been done on the low speed shaft with a laser vibrometer, and did not pick up excessive vibratory torque. Testing was done again with a strain gage system, and the slightly different torques measured versus analysis were reinput into the models to determine the magnitude of the torques at the high speed coupling between the gearbox and first compressor. The torques were again found to be acceptable, but still did not explain the failure based on the coupling supplier's analysis.

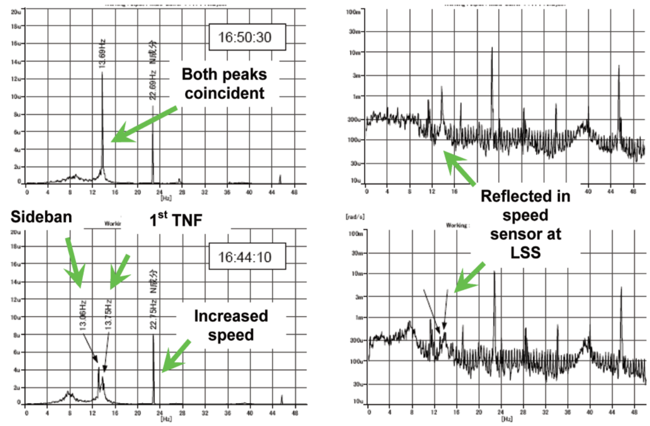

Figure 24. Case 6 Torsional Oscillation and Sidebands.

In the end, it was decided to just eliminate the odd frequency vibration, by reprogramming the VFD controller. This was done and the train is running now as designed.

Case Study 7

After 1835 service hours and 129 starts, high lateral vibration tripped out a train consisting of a VFD controlled induction motor (3594 hp at 795 to 1590 rpm) to a speed increaser to a high speed compressor (6136 to 12,272 rpm). Maintenance investigation revealed a cracked sleeve in the low speed high performance disc coupling (Figure 25).

Figure 25. Case 7 Cracked High Performance Disc Coupling.

Examination of the failed coupling revealed that the crack measured 10 inches in length and was oriented at an angle of approximately 45 degrees with respect to the coupling axis. No damage to the hub or the disk pack was observed. The crack was opened to expose the fracture surfaces, and detailed examination revealed a crack origin along the inner surface near the sleeve/spacer flange (Figure 26). The fracture features indicated that the crack had propagated through the sleeve wall and along the length of the sleeve. The crack path was predominantly planar. Scanning electron microscope (SEM) examination of the crack origin on the inner surface of the sleeve revealed limited plastic deformation, and the fracture surface displayed a transgranular morphology and crack growth marks (Figure 27). No metallurgical defects or material discrepancies were identified.

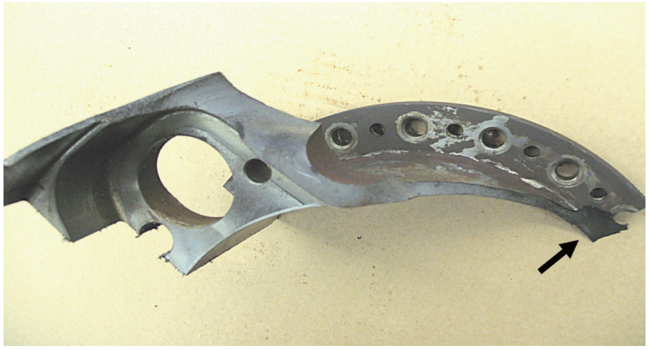

Figure 26. Case 7 Crack Initiation Site in Sleeve.

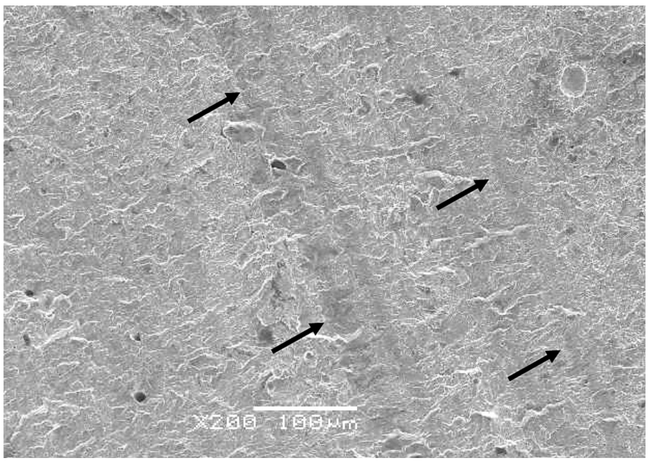

Figure 27. Case 7 SEM of Crack Initiation Site.

The cracking of the subject coupling sleeve was produced by a high cycle fatigue failure mechanism under torsional loading. One of the causes (under investigation) of the subject failure was a high frequency, cyclic loading of the coupling caused by the signal from the variable frequency drive. Another was excessive misalignment that was well above the target tolerances when checked (Figures 28 and 29).

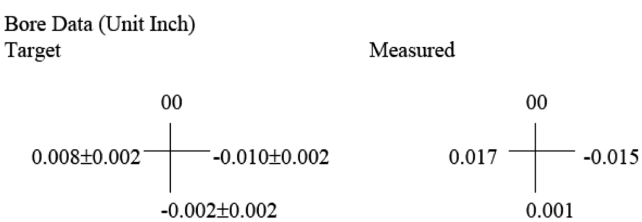

Figure 28. Case 7 OD Alignment Data .

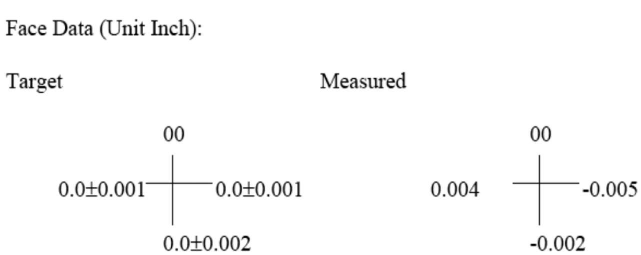

Figure 29. Case 7 Face Alignment Data.

A review of the torsional analysis revealed that there should not have been any 1× interference within the operating speed range, and a 2× and some higher harmonic pulsations from the VFD had interferences, but the resulting amplitudes were well within the stress levels allowable for the coupling.

The review of the coupling design showed that the highest stresses from a combination of angular misalignment and torque oscillations are in fact at the location of actual crack initiation discovered. Since the exact misalignment is known, but the actual torque pulsation magnitude and frequency are not, the coupling manufacturer, through FEA, is trying to figure out what the range of torque pulsations and frequency could have been to cause the failure.

Until these pulsations are known, the replacement coupling has been "beefed up" to resist torque pulsation fatigue in the area of crack initiation by increasing the flange/tube junction radius, with minimal change in weight and coupling torsional stiffness. Furthermore, the replacement coupling has been outfitted with a strain gage torque monitoring system that has the capability of measuring the actual torque pulsations. These data will be invaluable in determining the root cause of failure and getting data to verify the torsional analysis.

DISCUSSION OF VARIOUS TORQUE OSCILLATION MEASUREMENT SYSTEMS

Feese and Hill (2004-2006) describe four main ways to measure high frequency torque oscillations, but each has its limitations. A torsiograph (Figure 30) is an instrument that rotates with the shaft and is used to measure angular velocity (deg/sec) or displacement (degrees). One type, a torsiograph, operates on the seismometer principle, with a mass retained by springs whose relative motion compared to the stator is converted into an electrical signal by inductive proximity detectors. The device must be mounted on the free end of a shaft.

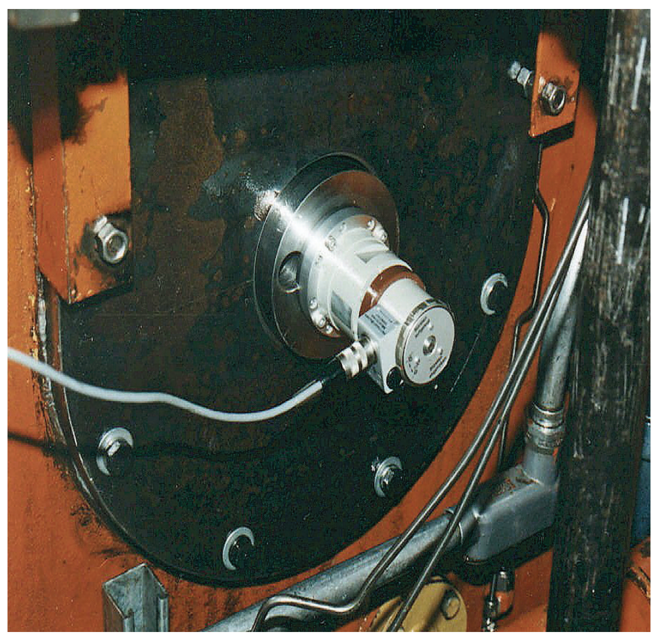

Figure 30. Torsiograph.

While the instrument is easy to install, it is sensitive to lateral vibration and will require that the shaft end be true and drilled and tapped such that the torsiograph is centered on the shaft. The issue with this device is that the amount of oscillation may not be an indicator of stresses. For example, high oscillation can occur in a system with a soft coupling, but the stresses may be low.

Torsiographs are no longer manufactured (Feese and Hill, 2009) and have been replaced in many cases by shaft encoders, or torsional lasers (see below). Like the torsiograph, a rotary shaft encoder is attached to the free end of a rotating shaft and generates pulses electronically or mechanically that can be processed by data acquisition systems to measure angular position of the shaft. These devices have the same measurement limitations as torsiographs, except that they are easier to install, and can even be installed with the machine running if the shaft has been prepared ahead of time.

Strain gage telemetry (Figure 31, 32, and 33) can be used, and it will measure actual stresses due to torque at a high enough frequency to detect torque oscillations at their typical resonant frequencies. The gages have to be placed at the right place depending on the mode shapes of the torsional oscillations. In addition, as it was in Case Study 6, attachment methods at high speeds are a safety issue, and it can sometimes take hours to mount the instrumentation.

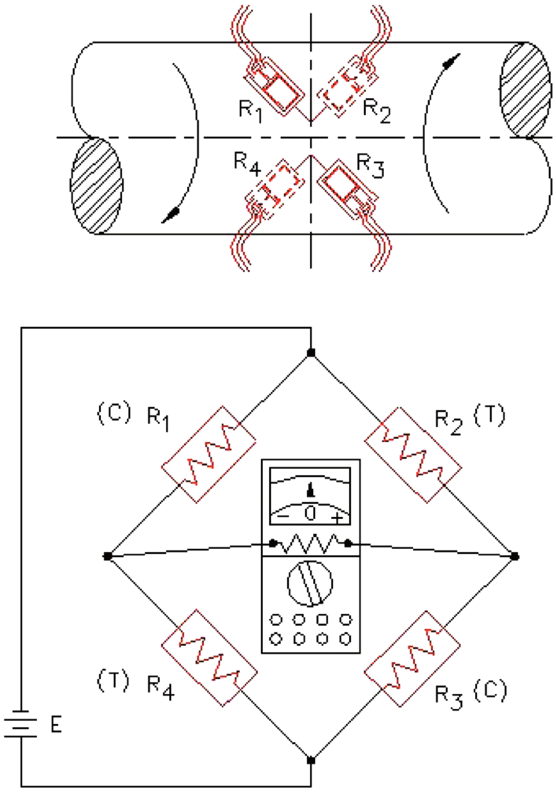

Figure 31. Strain Gage Bridge.

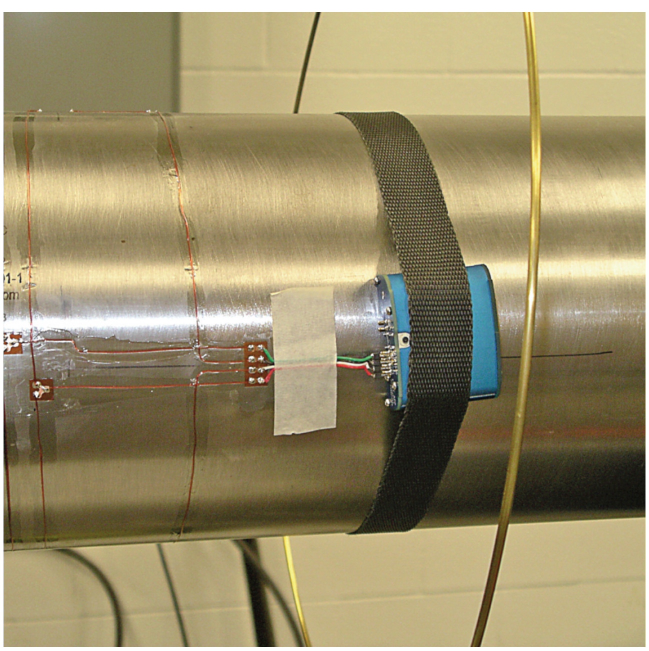

Figure 32. Strain Gage Telemetry Installation.

Figure 33. Strain Gage Telemetry Mounting Strap.

Historically, slip ring systems were used to provide excitation current to the strain gage bridge and to get the measured signal off the rotating components to the stationary instrumentation. Expense, signal noise issues, and reliability of the slip rings have made them rarely used today. Now, some version of a telemetry system is used.

The rotating components include signal conditioning electronics, a transmitter, and an antenna. The stationary components include an antenna, a radio signal receiver to decode the signal, and electronics either translate the signal to a voltage or current signal that a data acquisition system can measure and record. Of course the rotating electronics need power, and there are two ways of doing that.

One is to use a battery that is mounted onto the coupling. The advantage to a telemetry system like this is that it is easier to get a system up and running relatively quickly in the field on an existing coupling. The disadvantage is that depending on the battery life and the power consumed by the strain gage bridge and electronics, one will be limited to only three to seven days of data before the battery dies and needs to be replaced.

Power can also be transferred to the rotating coupling inductively. Power is transferred from the stationary to the rotating antenna. While this system has an added expense over the battery powered system, it is not limited in the length of time it can provide strain gage torque data. For either type of system, it is recommended to work with the shaft or coupling manufacturer so that the strain gage system can be properly incorporated.

A frequency modulation system uses proximity probes or a magnetic pick-up (Figure 34) to measure the pulse rate or gear tooth passing frequency of a train component (Figure 35). It does not measure actual stresses. By measuring the phase displacement between multiple sensor locations and correlating with torsional stiffness calibration data, the torque can be determined. However, for measurement of higher frequency torque oscillations, it is necessary to look at the pulse data at a finer resolution, perhaps down to even the tooth-by-tooth level. The highest frequency of torque that could be measured would be limited as a function of the tooth passing frequency.

Figure 34. Magnetic Probes.

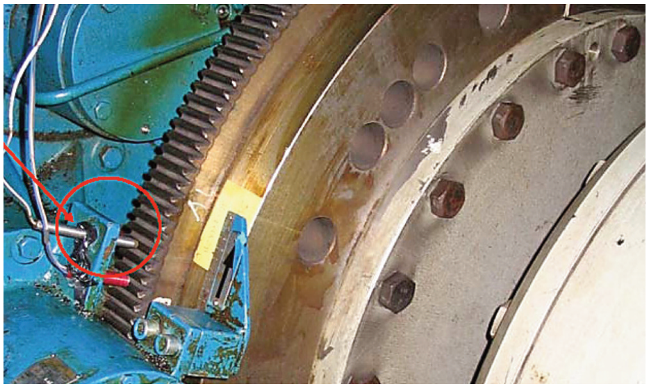

Figure 35. Proximity Probe Measuring Gear Tooth Passing.

A laser vibrometer (Figure 36) can be used to measure angular displacement in degrees. The laser requires reflective tape to be wrapped around the target surface (shaft or coupling tube). Two laser beams are produced by the vibrometer, which show up as two bright dots on the reflective tape. The signal differences in the reflected beams as the target part rotates is processed and filtered, and produces a voltage proportional to torsional oscillation in degrees. This system does have limits on measuring actual natural frequencies, and does not measure stress directly.

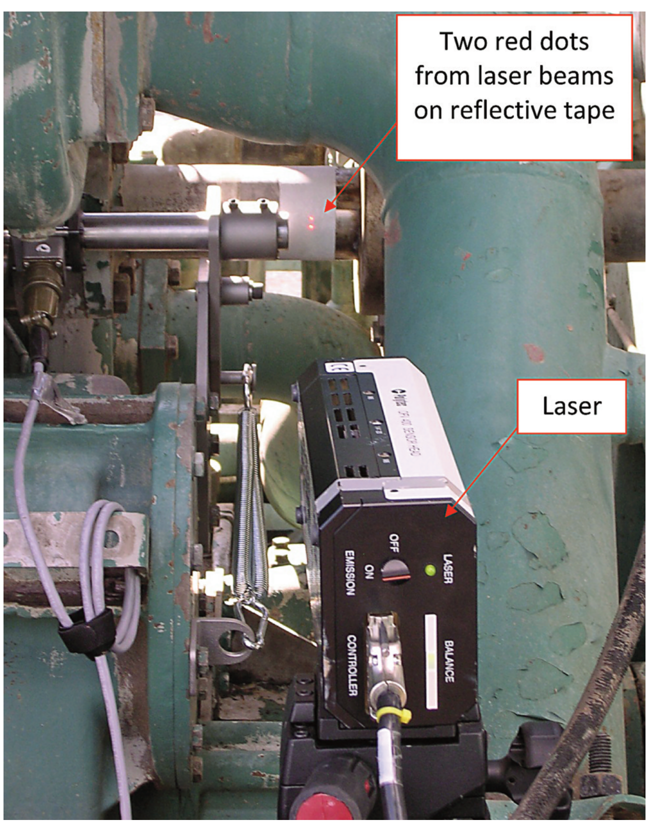

Figure 36. Laser Vibrometer.

As can be deduced from the description of these various torque oscillation measurement methods, they all have limitations, which need to be understood before choosing or using them for the particular application, and for interpreting the data resulting from the measurements.

CONCLUSION AND RECOMMENDATIONS

First and foremost, no matter what train is being modeled, a proper torsional analysis is required. Off-design conditions such as engine misfires and cold starts, should be modeled. In addition, the potential discrepancies in reported equipment inertias and torsional stiffness should be taken into account. For example, modeling an induction motor as a single mass-spring system can result in inaccurate and/or missed torsional natural frequencies according to Feese and Hill (2004-2006). Another example is that the torsional stiffness of the couplings, typically the softest in the system thereby giving them a large influence on the resulting frequencies, can vary by 5 to 35 percent from reported values. The authors have already discussed the nonlinearity of elastomeric coupling stiffness and the effects of temperature and age, but even a typical high performance diaphragm or disc coupling stiffness can be off slightly. The stiffness of these couplings are spot checked in the test labs, but they are not checked for every size coupling at every combination of conditions of speed, angular misalignment, speed, torque and axial stretch and in all the ranges of machining tolerances.

With trains incorporating VFD motors, based on the authors' experience, consideration of all possible sources of excitation including harmonics and sidebands of the pulse width modulator and the noise floor is recommended especially for geared units. In some cases, electromechanical modeling of the VFD system coupled with the machinery train may be required. The need is greater with new inverter designs or where power levels exceed previous applications. The intent, beyond modeling the impact of the inverter frequency fluctuations, is to determine the control algorithm impact on the first torsional mode. Stability of this mode is of primary importance. Software packages are commercially available to analyze these systems. However, a complete package (treatment of the mechanical and electrical systems equally) is not known to the authors.

Next, once a proper analysis is complete, make sure the predicted vibratory and mean loads for as many conditions as possible are checked with the equipment suppliers, especially the coupling manufacturer if the coupling is the weak link by design. There are many tools to model the predicted loads and ensuing stress levels on the equipment shafts and couplings, FEA being of course, one of them. Some people use service factors to check whether a component is suitable for the loads. That is fine if there is a lot of experience with a certain train, but with newly designed trains especially, the service factor could be too high, leading to excess weight causing lateral problems, or, much worse, too low.

Finally, and this is the authors most important recommendation, actually measure, at all phases of production and testing, and at all possible loading conditions, the actual vibratory torques, amplitudes and frequencies in an equipment train, and compare them to the predicted values, *before* a serious problem occurs. Refine the models if discrepancies are found.

Once a problem is found or failure occurs, use the appropriate measurement technique to actually measure, at the point of failure, the actual stresses. Then again, compare them to those predicted, discover the reason for any unforseen loads, and make corrections

The authors do not accept high speed couplings or centrifugal compressors designed by only analytical means. Couplings, for example, per API 671 Fourth Edition (2007), and rotors per other relevant API specifications, are actually balanced on balancing machines, and except for very low speed applications, are not allowed to have calculations alone verify the balance quality. In the case of compressors, mechanical and aerodynamic performance is measured. So, the case studies presented reveal that one should expect more than calculations for determining the torsional behavior and acceptability of turbomachinery equipment trains. Actual measurement of predicted torsional oscillations, especially on new applications prone to torsional problems, like VFD controlled driver trains, should be done during at least the testing and commissioning phases of prone turbomachinery trains.

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