IS THE DRY COUPLING REALLY CAUSING YOUR VIBRATION PROBLEM?

by

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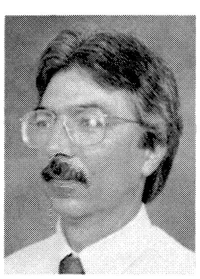

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ABSTRACT

From time to time vibration probes pick up high vibration at the coupling location on output shafts. Many times the coupling will then be removed and sent to the shop or to the manufacturer for inspection, repair, and rebalancing. In many of these cases, especially with dry couplings, however, the root cause is elsewhere [1]. The coupling vibration—and sometimes damage—can be a symptom of other problems in the train.

What these vibrations mean is addressed as related to the coupling and vibration signature. Is it the coupling, or is the rotor out of balance? Or, is there a problem with the fit between the two? What else could be wrong?

Field cases are presented where suspected coupling problems turned out to be something entirely different. In one case, an axially vibrating coupling turned out to be the result of a resonance condition in the feedback loop in the controller of a variable frequency motor. This resonance caused a torsional vibration that showed up axially in the coupling.

In another case, three different high performance disc and two different high performance gear couplings were installed before a vibration problem was traced to pipe strain induced misalignment.

Other cases are presented, including some where the coupling actually was the problem. Moreover, coupling balancing is covered, especially as it relates to the overall rotor balance condition. How can you be assured that a coupling balanced at

the factory will still be in balance when installed in the field? What is the importance of balancing tolerances and how significant are they when referenced to the out of balance due to fits and clearances? It is shown that the setup indication tolerances are a more significant contributor to overall coupling balance than actual balance machine unbalance tolerances.

All of the cases presented involve dry couplings—ones that don't have lubricated gear teeth. On almost all new high performance turbomachinery applications, dry couplings have replaced the gear type, and one of the reasons for this is the general lack of coupling problems associated with the dry couplings (except for an occasional windage problem). There are usually no wearing parts, and there are less clearances (like in gear tooth mesh), which can lead to significant unbalance and vibration in sensitive applications.

INTRODUCTION: WHAT CAN A COUPLING DO TO THE VIBRATION OF YOUR ROTATION SYSTEM

To understand whether the coupling could be the cause of your vibration problem, it is important to understand the coupling mechanics first—what is the role of the coupling in the machine dynamics. In a rotating system it can be categorized based on its influences/responses to its train vibration characteristics: laterally, torsionally, and axially.

Note that, herein, a coupling or component that generates excitation (excites itself and other components connected to it) is referred to as active. A component that must be excited by external sources is referred to as passive. The coupling's active or passive role in regards to different modes of vibration is shown in Table 1.

Table 1. The Coupling's Role in Vibration.

Vibration	Coupling's Role
Lateral	Active
Torsional	Passive
Axial	Passive
Misalignment	Active

Coupling's Role—Lateral Vibration

Lateral vibration is vibration in a direction perpendicular to the axis of rotation. As a part of the rotating system, the coupling can influence the lateral vibration in several ways:

- Unbalance

Every rotating component, including the coupling, is an active device as it exhibits, large or small, some amount of unbalance which generates lateral excitation forces under speed (proportional to speed^2) at 1× frequency. For this reason, unbalance of a rotating shaft (system) has become one of the most important and visible parameters for middle to high speed rotating machinery. The unbalance left in a machine balanced coupling consists of two components,

- *Residual unbalance* is measured on the balance machine relative to the balance machine rotating center. The magnitude of this can be as low as the balance machine permits.

- *Eccentricity/clearance induced unbalance* exists whenever there is a mass shift effect due to eccentricity between the actual coupling running center and balance machine rotating center, and/or clearance between mating pieces. The magnitude of this type of unbalance usually dominates the total unbalance of a coupling.

This unbalance can be effectively controlled through the use of interference pilot fits between mating pieces and tight concentricities of the coupling mounting surfaces (bores and pilots) to the center of rotation.

- Coupling Weight and Center of Gravity (CG) Location

The coupling weight and center of gravity location is important from a rotordynamic point of view as it affects the system natural frequencies and mode shapes. A heavy coupling with the CG far away from the support bearing will make the system more sensitive to excitations and expose the bearing to high load under speed. A train lateral analysis can be used to help identifying the allowable coupling weight and CG location.

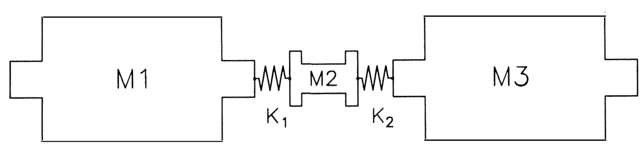
Coupling's Role—Torsional Vibration

A coupling is a generally torsionally passive device and needs external excitations in order to vibrate. A coupling is one of the components that make up the system torsional characteristics. A system can vibrate torsionally, even when away from any resonance when operated under significant torque oscillation (cyclic load). It can also vibrate significantly at system resonance with small torsional excitation. Persistent and high levels of torsional vibration may cause the equipment to fail in torsional fatigue. Generally the softest component in the system, compared to generally massive turbomachinery rotors, the system typically winds about the coupling.

Other than avoiding running at resonance, torsional vibration can also be improved by eliminating unexpected torsional excitation. For cases where significant torsional excitation is inevitable (for instance, synchronous motor and variable frequency drives), damping may be introduced—usually with elastomeric coupling elements—to reduce the magnitude of vibration.

Coupling's Role—Axial Vibration

Not all couplings have an axial natural frequency (ANF). Only couplings with axially elastic flexible elements (such as elastomer, disc, or diaphragm couplings) may resonate under axial excitation. The ANF exists for such couplings whether they are stretched or not. A diagram showing a simplified model of a direct drive train (mass - spring - mass - spring - mass) is shown in Figure 1. Without excitation, the coupling can not vibrate axially on its own. That is, the coupling is axially passive.



M1 – Mass of driving rotor and coupling driving hub
M2 – Mass of coupling floating section
M3 – Mass of driven rotor and coupling driven hub
K_1, K_2 – Stiffnesses of coupling flexible elements

Figure 1. Simplified Axial Model of a Coupling Between Two Rotors.

The authors know of no documented cases of a dry coupling resonating at its axial natural frequency and contributing to high train vibration. This is because of, in the opinion of the authors, the unlikelihood of encountering high axial excitation amplitude sources in general turbomachinery applications. Further, many couplings, such as disc pack and elastomeric couplings, have built-in axial friction damping and/or nonlinear stiffness that limits the vibration amplitude [2]. For these reasons, axial vibration due to a coupling ANF is unlikely a concern and, when there is vibration, the source of excitation could even be nonaxial (transverse effect). Examples are synchronous motor drives, or variable speed trains that could generate significant torsional excitation that actually vibrate the coupling and connected equipment axially. Another possible source of excitation includes excessive angular misalignment.

Coupling's Role—Misalignment

Proper installation procedures and alignment methods can reduce excitation and prolong the equipment life. Excessive misalignment can cause unwanted vibration and possibly fatigue the equipment.

If two machines are significantly out of alignment, excess vibration will result due to the flexing forces (bending forces) in the coupling acting on the connected shafts. These forces will cause 2 × and sometimes 1× frequency vibrations.

CASES

The following cases illustrate the above principles, and how they were or were not understood and followed.

CASE 1

Creeping Misalignment of a Process Compressor

The operators of a natural gas boost compressor station had already successfully commissioned and operated one boost compressor package. This package and a second one were each driven by 12,000 hp rated gas turbines. While trying to commission the second package, the boost compressor kept shutting down on high inboard bearing vibration. The shutdown occurred while downloading the unit, 16 seconds after initiating a normal stop, at about 60 percent speed (design speed was 7640 rpm). This resulted in poor control of the powertrain process, the down loading cycle, and general operation. Even though it was possible to download without a shutdown by manual control of the system, it was not acceptable as it was considered necessary to be able to conduct remote operation control.

The main goal posed to the troubleshooter was to find a quick and effective way of keeping the compressor operating adequately (with full control off the recycle valve and cool down cycle) during the contractual heating season. After the heating season there would be plenty of time for analysis and repairs.

One key feature of both packages' power train was the introduction of a dry disc pack coupling design between the power turbine and the boost compressor. All other components of the power train had been proved many times over by the station operator and by other operators.

As mentioned before, one unit downloaded with full control of the recycle valve and cool down cycle and the newer unit did not. The Bodé plot response depicting a shutdown episode is presented in Figure 2. Except for the apparent high runout at very low speed, the synchronous response did not yield a clear indication of the cause of the shutdown. At first it was thought that the shutdown was caused by an instrumentation problem. This was not the case, as it was found out later.

One of the plant operators familiar with the operation of the first unit was of the opinion that there was a noticeable alignment change in the second unit resulting from line pressurization alone. The operator had noticed the change when pressurizing the compressor with the alignment indicators attached to the coupling hubs the alignment figures changed noticeably! However, the only popular scenario around the station was that the second unit dry coupling was the cause of the shut downs.

Vibration data were tape recorded for a normal stop that included the shutdown. A waterfall is depicted in Figure 3 of the forward radial noncontact probe which experienced the shutdowns. The problem was transient high subsynchronous vibration after initiating the normal stop. The subsynchronous vibration only showed up during the controlled downloading and rundown, but not when the unit was operating at full load.

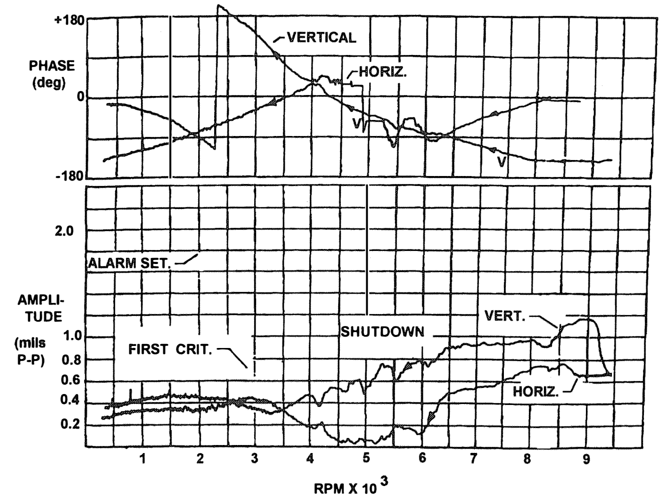


Figure 2. Axial Plot Response with High Vibration Shutdown.

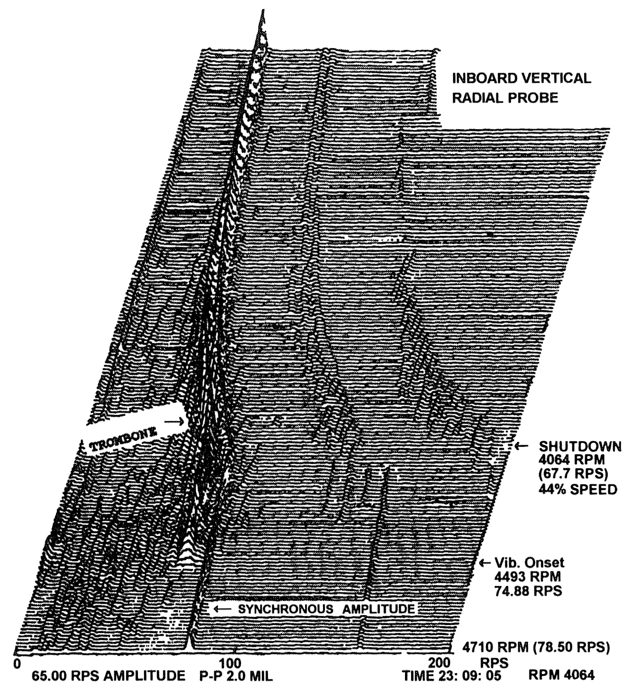


Figure 3. Unit 2 Normal Stop Including Shutdown at 60 Percent Speed; Vendor #1 Coupling.

Since the coupling was dry and there was no oil running through the interconnect spool the possibility of a flooded coupling was discarded.

The consensus was to try another vendor's dry coupling in the event that the first coupling was defective. (Note that both the original vendor's coupling and its spare had already been tried without good results. It was thought that the first coupling had been overstretched axially and damaged, and the spare was therefore installed.)

The same normal stop is depicted in Figure 4, including a shutdown at 60 percent speed, with the second vendor's coupling. This result was not expected and quickly made an oversimplified problem into a complex one. At this time, the shutdown subsynchronous vibration phenomena was called the "trombone," based on the audible tone heard from a speaker. The image of the unpopular plant operator who initially suspected an alignment problem improved dramatically among his peers. The results from the data also deactivated anyone's blame on any dry coupling design.

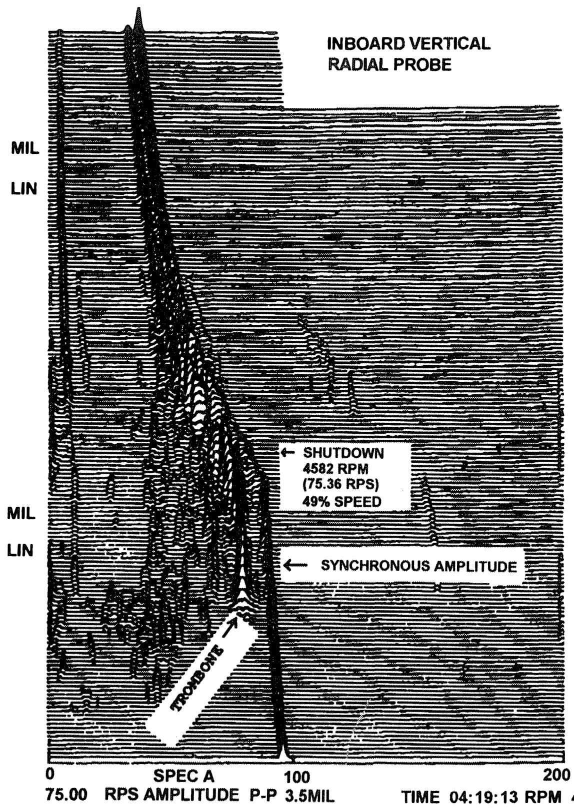


Figure 4. Unit 2 Normal Stop Including Shutdown at 60 Percent Speed; Vendor #2 Coupling.

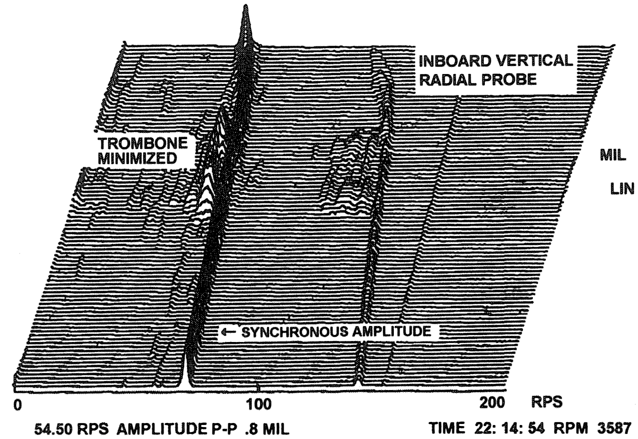


Figure 5. Unit 2 Normal Stop, Unit @ Alarm (But No Shutdown) @ 60 Percent Speed; Gear Coupling.

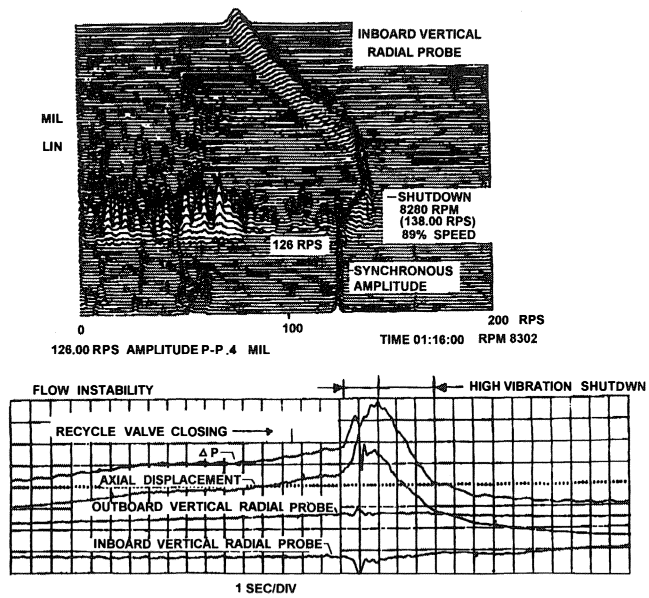


Figure 6. Shutdown Caused by Flow Instability.

There was vast operating experience with lubricated (wet) or splined/geared couplings for this type of compressor train. A wet coupling was installed as soon as it was available. The result of a normal controlled stop is presented in Figure 5. There was a repetition of the "trombone" phenomena, however, but the vibration amplitudes at the forward bearing only activated the alarm setting, not a shutdown. After repeating the stops several times, there was a spell of relief when the shutdowns did not materialize. However, the line pressure conditions were not at the maximum. As soon as the line returned to maximum pressure, the vibration did not always stay at the alarm level, and it reached the shutdown setting at least once.

It was then necessary to do a complete troubleshooting investigation comparing the shaft response as a function of gas pressure, gas temperature, and alignment.

It was already known that process delta pressure (ΔP) discontinuities across the compressor, during uploading or downloading, resulted in a coupled response of the compressor shaft axial position (Figure 6). The close relationship of the ΔP variations across the compressor and the axial response of the shaft is shown in Figure 6. Except for a small phase shift one can use the axial shaft position as an indication of the process ΔP.

The ΔP/axial shaft response phenomena was investigated in the subject compressor vibration problem along with a step by step survey of the alignment anomalies reported by the plant operator.

A normal stop comparison is shown in Figure 7 of the radial and axial shaft response of the unit that operated without problems (unit 1), the unit experiencing the shutdowns and alarms (unit 2), and a similar unit at another station. Data for the similar unit correspond to the same condition presented in the previous figure. Journal position is depicted in Figure 7 with respect to the bearing 6:00 position given by the inboard bearings' X and Y probes and the outboard bearing's Y probe.

From Figure 7, it is concluded that there was ΔP activity in unit 2 (Figure 7 (c)) after the normal controlled stop was initiated. Unit 2 ΔP activity was very different from the one experienced by unit 1 (Figure 7 (b)) which was deemed a unit with normal ΔP response. By analogy and comparison with a similar compressor, Figure 7 (a), it was concluded that a ΔP flat slope followed by a ΔP discontinuity was causing what was earlier called "trombone" vibration.

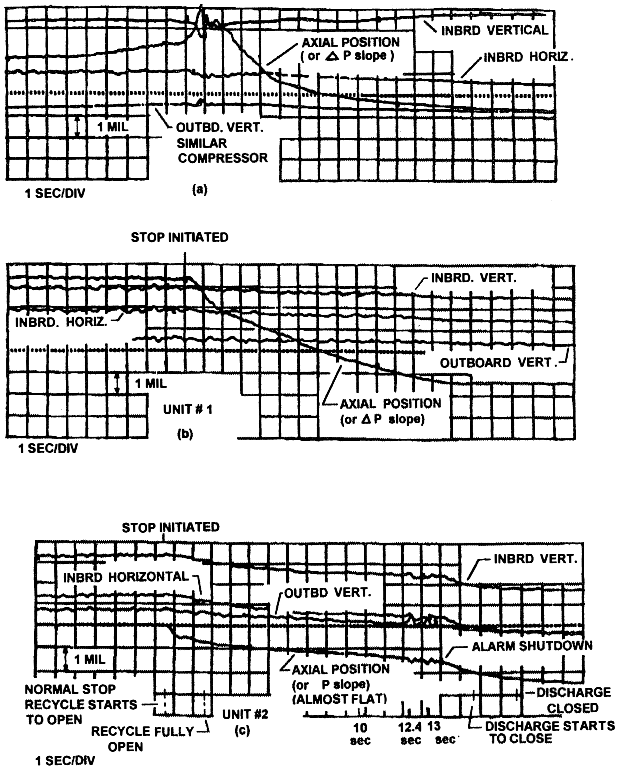


Figure 7. Unit 1 and Unit 2 (Vibration at 60 Percent Speed) and Similar Unit Radial and Axial Shaft Position Corresponding to ^ P Fluctuations.

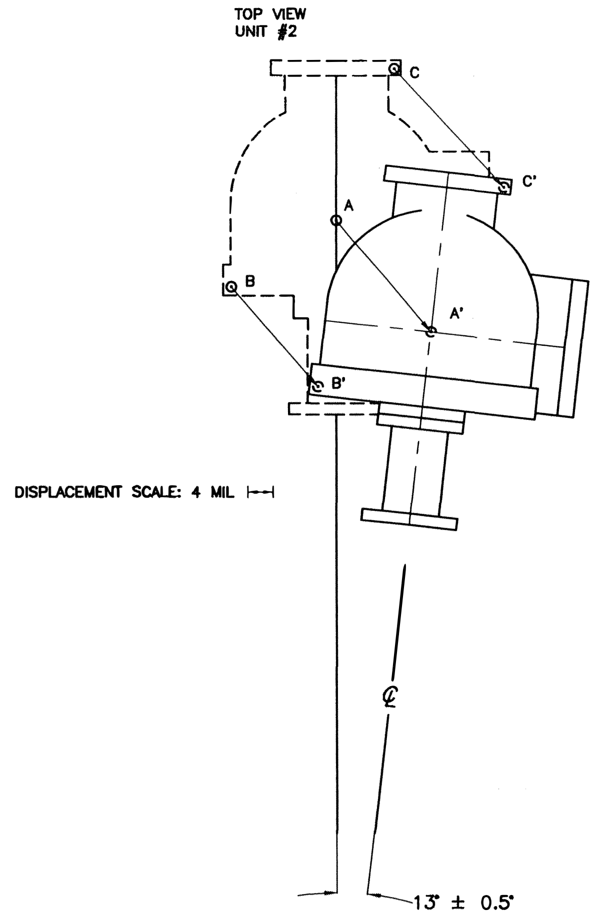


Figure 8. First Pressurization.

The results of the alignment study are presented in Figure 8 and Figure 9. The top view of the compressor is shown in the figures in terms of initial position (dotted lines), intermediate, and final alignment positions (solid lines). The initial position was a result of suction header pressurization (Figure 8). This was followed by operating thermal expansion of the discharge header (Figure 9). The final misalignment was considered severe with a vectorial magnitude greater than 0.125 in. The centerline of the compressor shaft actually went down and away from the expected self aligned condition.

Beyond the quantification of the alignment problem one temporary alternative to meet the troubleshooting objective was to calculate a cold offset alignment that would compensate for the known operating misalignment. The compressor was the "fixed" component and the turbine driver the movable one. The magnitude of the compensated cold, un-pressurized offset was large relative to the change of the turbine feet. The compensated offset would have put the front engine mount off the foot print of the base. Moreover, the compensating offset would have moved the engine exhaust centerline far away from the centerline of the exhaust bellow to a point that exceeded the installation tolerance. Finally, based on previous operator experience of having tried to "chase" or compensate for the pressurization misalignment more than once without effective result, it was opted not to compensate the problem anymore but rather to fix the root of the induced misalignment.

CASE 1 CONCLUSIONS

The root of the shutdowns during controlled stops was not the dry disc coupling designs of the two different vendors, but a combination of pressurization and thermal expansion of the piping and an inadequate recycle piping system. This combination resulted in the "trombone" or subsynchronous vibration phenomena and a misaligned power train. The end result of these two problems resulted in repetitive vibration shutdowns.

The gear coupling ran better than the three different disc couplings in the sense that when the gear coupling was used, the alarm was often tripped instead of the shutdown switch. The vibration was still there, but not as severe. The coupling vibration was a symptom of the problem, not the cause. Changes to the piping thermal growth and the recycle loop were the final solution to the vibration problem.

CASE 2

Spiking Gearmesh Vibration

The power train discussed in this case is depicted in Figure 10. There were features of this power train that made it different from a standard production package. These features were: a disc pack coupling (dry coupling) between the gearbox and the compressor, the compressor alignment parameters, which included a vertical offset and a skew offset, and the compressor itself. This compressor was a standard design but never proven in the configuration discussed here. The gas producer turbine, the power turbine and the gearbox were standard package components rated at 1000 hp at 24,000 rpm.

During package string test, the gearbox exhibited spiking vibration exceeding the shutdown setting. There were two main spikes in the vibration signature. The first one showed up at about 85 percent compressor speed (full speed is 11300 rpm) at the gear mesh frequency, at 14.4 kHz. This spike was followed by another one below the gear mesh frequency, between 90 percent speed and 100 percent speed, ranging from 11.05 kHz to

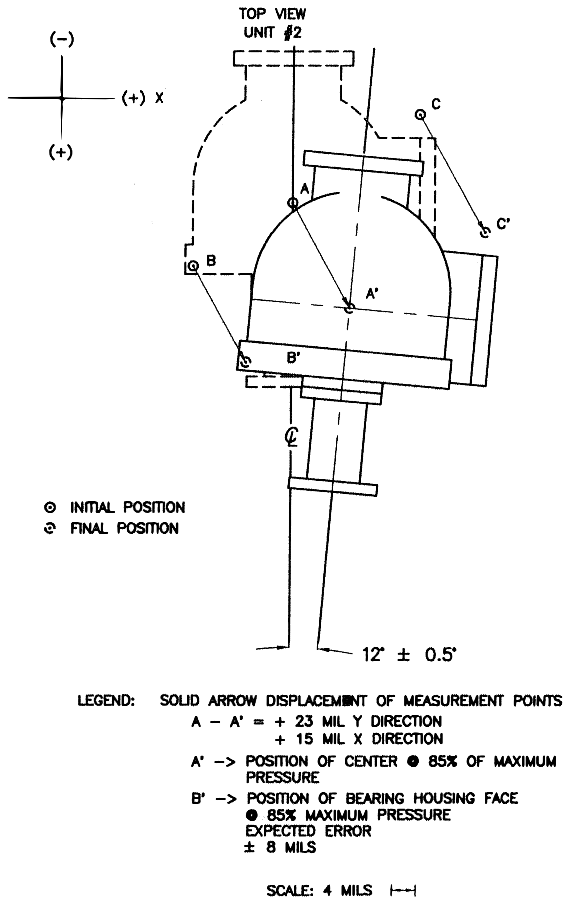


Figure 9. Normal Stop Attempted.

12.4 kHz respectively. The second spike was called 12.4 kHz mesh-subsynchronous (mesh SSV) vibration since the gear mesh frequency was 15982 Hz at 100 percent speed.

By eliminating several possible causes, and by using the concept of operating amplication factor of the spiking phenomena, the root cause of the spiking was narrowed to one gear element.

The compressor operated between the first lateral critical speed and the second. The first critical speed was measured at 95 rps (rev/sec) or 50 percent speed. The compressor was mounted on three pedestals requiring atypical alignment which included

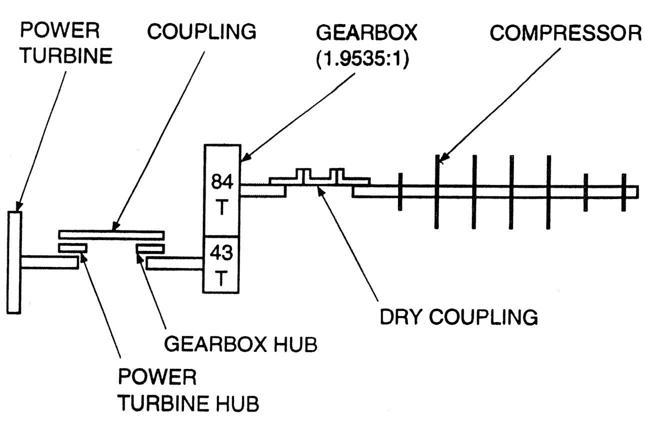


Figure 10. Case #2 Train.

cold horizontal and vertical offsets yielding both vertical and skew offsets. The maximum input speed to the gearbox was 371.7 rps and the maximum output speed was 190.3 rps.

When the package was string tested (rather than load tested) at the factory, the vibration monitor showed two spikes (Figure 11) reaching levels above shutdown settings. This was observed as the power train was accelerated from 75 percent speed to 100 percent speed. This type of phenomena had not been observed before in similar package configurations featuring the same driver, the same gearbox, a "wet coupling" (gear-spline lubricated style) and other vendor compressors. Because this new compressor required skewed alignment figures, it was thought that the dry coupling design was pushed to a poor performing condition, and thus the unusual gearbox spiking.

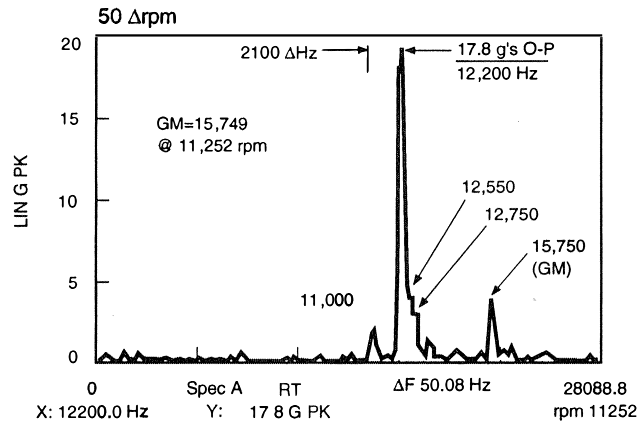


Figure 11. Spectral Analysis of Gear Vibration.

Since during the standard string test, it was not possible to apply a process load to the compressor, the power train, and especially the compressor, never became fully heat soaked to reach full thermal growth and a self aligned state. This lack of load hindered the chance of overcoming this three pedestal mount compressor's alignment figures for a vertical and a skew cold alignment offset shown in Figure 12. Also, test cell engineering reported that the spiking amplitude was greater with a cold power train than with a hot one.

The dry coupling was also tied into the source of the spiking by the following rationalization. While the gearbox output shaft reached almost full axial thermal growth, the relatively long

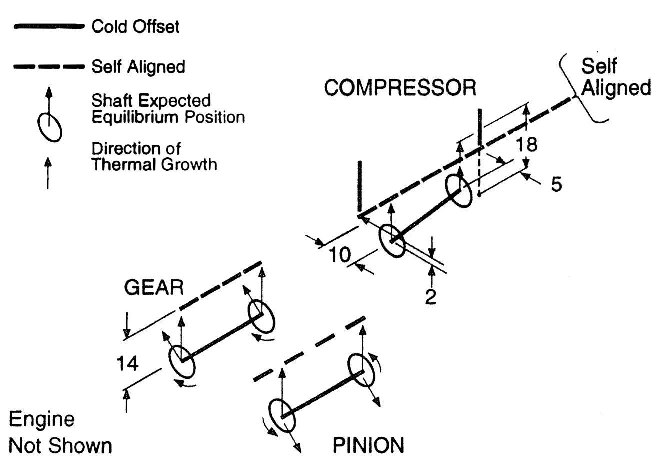


Figure 12. Relative Position of Shafts Showing Complex Alignment Between the Gearbox and the Compressor.

compressor shaft did not, due to a lack of load. This meant that anytime during the string testing, the dry coupling axial preload of about 0.070 in was never fully relieved, resulting in an axial spring load of the coupling disc pack yielding susceptibility to the coupling's axial natural frequency (ANF).

Therefore, the bidirectional cold offsets and the axial preload were thought to combine into a setting that caused a mismatch of the gear mesh that led to the multiple spiking.

The gearbox mesh, featuring an opposed double helix design which results in a self centered pinion and gear, and, consequently, minimal axial thrust loading, was not thought of as being the cause of coupling axial loading. However, the reverse was considered possible. If the dry coupling ANF was ever externally excited, this could result in axial reciprocating motion that was transferred to the gear and that, in turn, would disturb the pitch diameter mesh and cause the spiking.

Possible ANF external excitation sources were the gearbox and the compressor undergoing axial reciprocating motion. By itself, the coupling should not undergo self excitation through its ANF range.

There were several factors that interfered with resolving the spiking problem. Based on the above test conditions and limitations, it would have been premature at first to cast a direct fault upon the gearbox gearing as the cause of the spiking. Moreover, it was difficult to make a comprehensive assessment of the magnitude of the spiking phenomena and to determine the root of the spiking due to the lack of torque load. This being the case, it was conceivable the resulting spiking was caused by a mismatched pitch mesh, due to causes external to the gearbox like a partially aligned power train lacking torque load.

Another factor intervening in determining the cause of the spiking phenomena and the time available to conduct the troubleshooting was the possibility of running into the contractual shipping penalty period that was just a few days ahead. Regardless of the cause, there was a very limited time window to fix the spiking phenomena with a minimum of instrumentation.

SPIKING VIBRATION SPECTRUM

The spiking phenomena is depicted in Figure 13 by means of a "waterfall" display of the output accelerometer. The speed swept from 73 percent to 100 percent speed and returned to 65 percent speed. The largest amplitude was registered at 19.8 g 0-P at 14.4 kHz. Mesh subsynchronous vibration (Mesh-SSV) is shown at the 9 kHz and 12.4 kHz range. Gear mesh spiking is shown at 14.4 kHz. Actual gear mesh (GM) as a function of

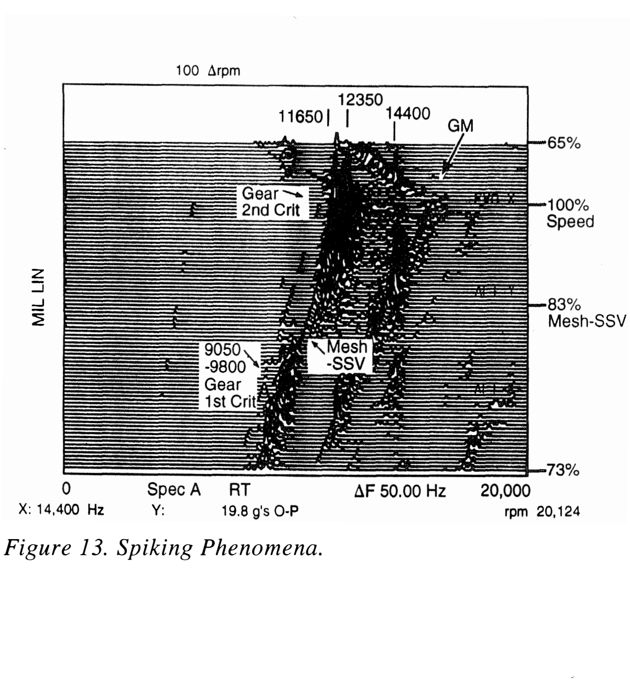


Figure 13. Spiking Phenomena.

speed is indicated by a soft diagonal line. The gear mesh vibration always tracked speed.

Before collecting more troubleshooting data, it was deemed necessary to make a list of possible causes leading to the mesh-SSV and the spiking gear mesh vibration. This list was ranked and prioritized by likelihood of cause. The list was put together to chart a troubleshooting course. Needless to say, the list was biased by trying to focus on the new power train elements as the possible cause of the spiking. At this point, the factory engineering and the customer had agreed that the spiking was not an acceptable phenomena, since it encroached into the operating range of the machine at very high g levels. The list is presented next.

Ranking of Possible Causes of Spiking Gearbox Vibration

- Partial misalignment and a stiff dry coupling leading to a mismatch of the mesh at high speed at no load and at 40 percent load.
- Dry coupling axial natural frequency (ANF), and/or lateral critical speed leading to a mismatch of the mesh at the high speeds.
- Power train torsional critical speed interference with gear mesh.
- Accelerometer/mount problem.
- Gearbox housing resonance due to a possible material inclusion at the output accelerometer mount location (both accelerometers showed similar response and the only difference was higher magnitude at the output accelerometer).
- A gearing problem (for all practical purposes this would be a long lead fix that was not commercially favorable due to the penalty clauses of the contract).

The gearing problems considered were:

- Gear mesh quality.
- Gear shaft resonance.
- Mesh mismatch resulting from the relative alignment of the bearing bores and hydrodynamic sleeve bearing performance.

One of the reasons that the coupling was ranked so high is that couplings have a bad reputation when it comes to vibration problems of power train components. Most of the times couplings are vulnerable to installation errors. These include handling abuse and mismatching of parts like substituting a damaged weight matched nut and bolt with an off the shelf unmatched nut and bolt. The latter leads to high unbalance before the coupling is even run.

Other possible causes not listed above (like compressor rotor instability) had not materialized in the data compiled by the test engineering during the string test.

TESTING AND TROUBLESHOOTING

Testing for the first two items on the list was carried out in three days. Each test is discussed separately.

Dry Coupling Axial Natural Frequency Test

The first test was the dry coupling preload axial natural frequency (ANF). The trace is shown in Figure 14 of the axial displacement of the shaft measured by the standard axial compressor probe. The mass elastic model consisted of an spring loaded in the axial direction with masses attached at each end. At the inboard end, the masses were the gear viscously coupled to the pinion at the mesh; at the outboard end the mass was the compressor shaft.

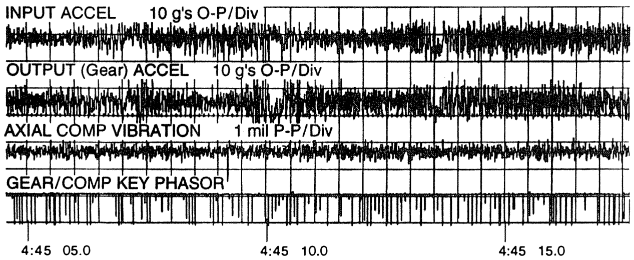


Figure 14. Trace of the Axial Displacement of the Shaft.

Physically, these masses were connected by the coupling disc pack that was a spring with a given preload rate. The result of this test is presented in Figure 15 and Figure 16.

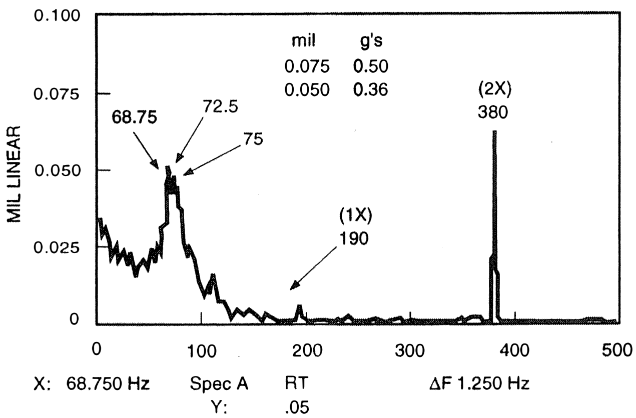


Figure 15. Peak Hold Spectrum - Axial Probe on Compressor Outboard End.

The peak hold spectrum is shown in Figure 15, from a hot break away to 100 percent speed (lube temperature 131°F), of the axial probe on the compressor outboard end. The ANF of the partially preloaded system is shown at 73 Hz and it appears well damped with an amplitude of 0.05 mil p-p. The 380 Hz spike corresponds to 2× the speed of the compressor shaft (190 rps) at 100 percent speed.

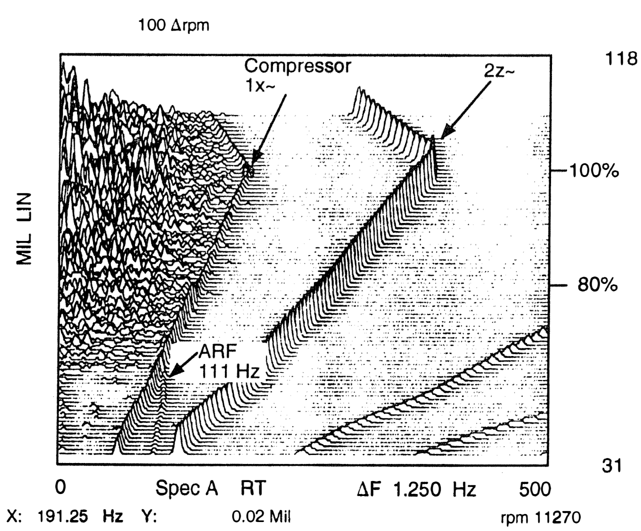


Figure 16. Waterfall of Axial Compressor Probe.

The waterfall of the axial compressor probe is shown in Figure 16 on the compressor outboard end at cold breakaway. The ANF of the pre-loaded dry coupling was measured at 111 Hz, with again, a low amplitude.

The spectral measurement of the axial response of the system showed that there was an axial natural frequency (ANF) excited with during runup but not amplified by 1× compressor speed. The power train was known to have a torsional critical speed very close to the excited ANF, thus the excitation seen on Figure 16 could have been caused by the nearness of the torsional critical speed. In spite of the excitation of the ANF, it was not the cause of the mesh-SSV or the gear mesh spike. This was deduced from the measurement that showed a cold ANF of 111 HZ (58 percent speed) and a hot ANF of 72 HZ (38 percent speed). Both of these ANFs occurred much earlier than the speed onset of any of the gear phenomena being discussed here (first onset at 75 percent speed).

Realignment Test Eliminating Cold Offsets

The realignment test followed the axial ANF test. The power train was realigned to simulate a thermally self aligned power train, but maintained the dry coupling axial preload. The realignment eliminated all the cold offsets between the compressor and the gearbox, but there were no changes of the vibration signature that would indicate a clear improvement (although the maximum levels at 14.4 kHz decreased by about 2.0 g O-P). The pattern shown in Figure 13 was basically duplicated.

The severity of the GM amplitude was then judged using the amplification factor (Q^o) scale in Table 2 as a reference for comparison:

This table was compiled from several documented sources. The amplification factor, Q^o, quoted in Table 2 should be used only as a reference for understanding the meaning or severity of a mesh response amplification factor.

Table 2. Amplification Factor Scale.

Q^o Range	Type of Damping
<< 2	Elastomeric
1 - 8	Viscous
9 - 20	Semi-Viscous to Structural
20 - 50	Structural

The Q^o of the mesh-SSV (12.4 kHz) was measured at 18 and 53 for a steady-state lube oil temperature of 131°F and at 90°F, respectively. A comparison is shown in Figure 17 of the waveform from the input (pinion) and output (gear) accelerometers during excitation of the 12.5 kHz spiking and 14.4 kHz spiking.

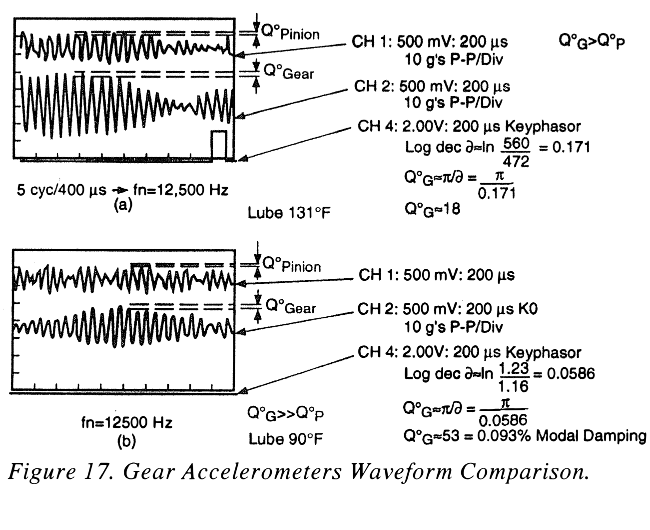


Figure 17. Gear Accelerometers Waveform Comparison.

Based on the high Q^{O}, the cause of both the mesh-SSV and a critical speed excitation was addressed as a gear teeth profile error. This was a gear teeth quality issue not a gear design issue. Both large amplification factors (Q^{O} = 18 and 53) made the spiking phenomena a non acceptable response based on the scales of Table 2. At this time, it was possible to attach a quantitative magnitude to an otherwise qualitative rejection due to gear spiking.

The gear vendor measured the quality of the gear and pinion teeth. The measurements showed the gear teeth profile error, rather than the pinion teeth error, were out of their specification. The most salient features before and after the fix are shown in Figures 18 and Figure 19. (Other measurements pertaining to the gear and the pinion are not presented in this paper to avoid conflict of interest or disclosing proprietary design features).

The before case in Figure 18 shows the lead tracing with two high points. Lead is a measurement taken on the tooth flank, along the pitch diameter. The magnitude of the high points shown by the traces exceeded the vendor specification.

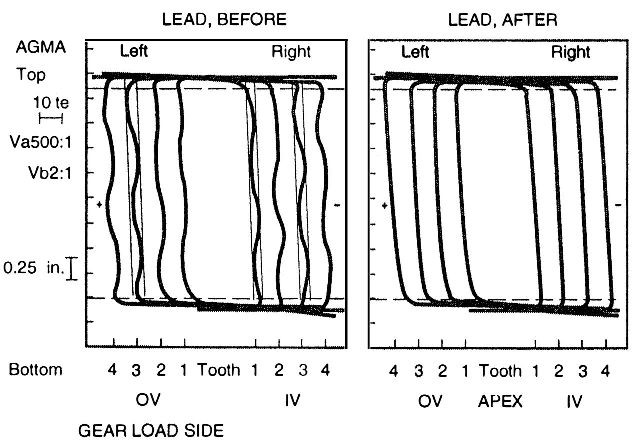


Figure 18. Gear Lead Before and after Rework.

The before case in Figure 19 shows the profile had a large protrusion just below the pitch diameter. Profile is measured along the tooth flank from the major OD corner of the tooth towards the root of the tooth. The magnitude of the protrusion shown by the traces exceeded the vendor specification.

CASE 2 CONCLUSIONS

These conclusions summarize the outcome of the major tests performed to find the possible causes of the spiking gearbox vibration:

- Gear vibration phenomena was not caused by poor accelerometer mounting or a housing resonance.

- Axial natural frequency, ANF, of the dry coupling was not the cause of the mesh-SSV or the gear mesh spike.

- Elimination of both cold vertical and skew offsets of the compressor did not show a significant influence upon the mesh-SSV and gear mesh spike. Therefore, the alignment was not the cause of the vibration.

- There was no torsional critical speed interference causing the mesh-SSV and mesh spiking. The mesh-SSV and mesh spiking induced a noticeable torsional response of the power train. The amplitude of the torsional vibration was measured at 1.2 degree P-P.

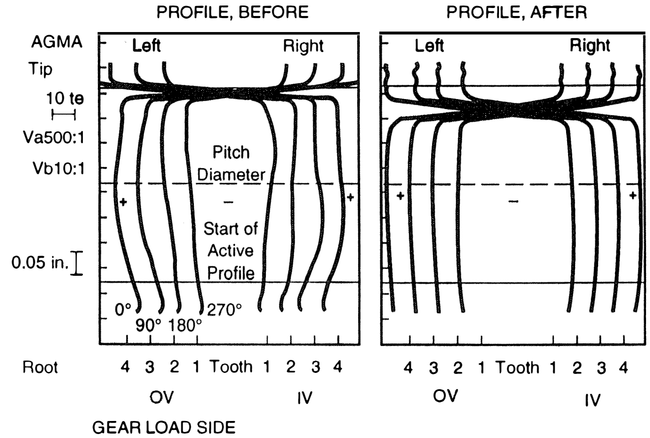


Figure 19. Gear Profile Before and after Rework.

- The cause of the mesh-SSV and mesh spiking phenomena was gear teeth profile error. This finding was firmed up from the measurement, the calculation and the comparison of the operating amplification factor of the gear at 131°F and at 90°F at 12.4 kHz, Q^{O} was equal to 18 and 53, respectively.

CASES 3A, 3B, 3C

These cases represent situations where dry couplings were actually at fault due to very simple and fundamental design and/ or manufacturing and handling errors.

CASE 3A

This case involves a coupling similar to Figure 20. Reports from an OEM's test stand indicated an 80 percent running frequency subsynchronous vibration, which was attributed to oil whirling inside the dry coupling.

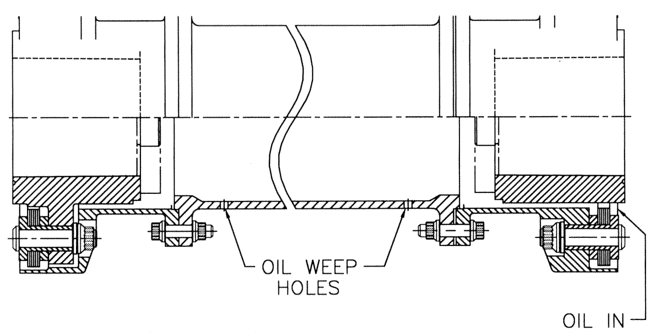


Figure 20. Oil Weep Holes.

Compressor oil meant to drain out of the coupling guard had gotten inside the coupling and could not get out. It formed a layer inside the spacer, which spun at a different speed than the coupling causing the subsynchronous readings.

Small, radial oil weep holes were drilled in the spacer to correct the problem.

CASE 3B

The coupling in this case was installed between a gas turbine gear driven and centrifugal compressor operating at a maximum continuous load of 9400 hp at 14830 rpm. The customer was experiencing 1.2 mils peak-to-peak vibration at 1× frequency at the gearbox shaft end. Acceptable limits for this location was 0.8 mils.

The coupling was indexed on the machine shaft, and the vibration phase angle followed the coupling. A coupling balance problem was then suspected and it was removed and returned to the manufacturer for analysis.

Upon inspection, it was discovered that there was a significant dent in the antiwindage shroud of the coupling disc pack assembly. This dent was not easily discernible to the naked eye. A flexible pad wrapped around the circumference of the shroud highlights the dent, as shown in the photograph of Figure 21. The displaced metal in the dent was calculated to produce a difference in unbalance of approximately 4.0 gr-in in the component assembly. This error was well beyond the allowable API 671 balance tolerance for this subassembly, and it was concluded that this error could have caused the excessive vibration.

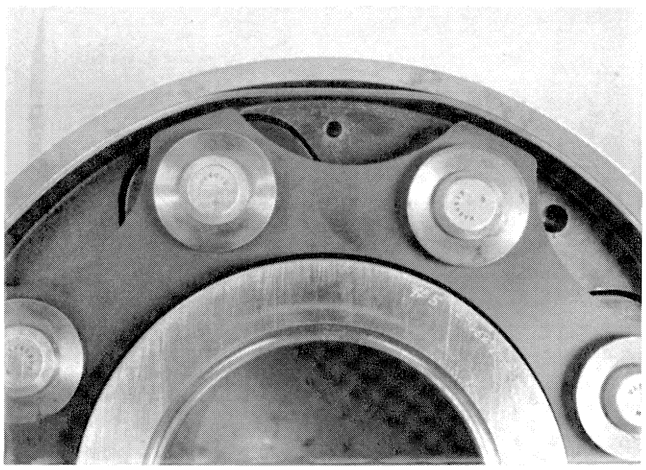


Figure 21. Windage Shroud Dent in Hub Assembly.

Not only was there damage to the shroud, but whatever caused the dent was of such a great force that the disc pack actually tin-canned slightly—meaning that there was elastic deformation in the pack that caused slightly buckled discs. These temporarily deformed discs were also not easily visible.

After the coupling was repaired, it ran well on test. However, it was not ever determined exactly when the damage occurred. It definitely occurred after the coupling was balanced at the coupling manufacturer's plant, but before the test run at the purchaser's facilities.

There was some suspicion that the shipping container was inadequate, so, to cover all bases, better boxing was supplied for the repaired assembly and subsequent shipments. Also, all personnel who could possibly handle the parts at both facilities were notified to be careful and to check for damage if any coupling components were dropped or bumped severely and to look for this damage in all areas, especially the critical areas of the component part locating pilots, and flexible elements.

CASE 3C

The dry coupling hub in this case was outfitted with provision for keyless hydraulic removal. All the oil porting and distribution grooves and O-ring grooves were machined into the coupling hub. The coupling was to be installed on an induction motor/gearbox driven centrifugal compressor, 5000 hp at 12,560 rpm.

Note in Figure 22 that the design of the oil porting in the hub included a high pressure 1/16 in pipe thread plug. This design made it easier to machine the oil distribution holes.

The problem that occurred in the field was that this plug had inadvertently been left out of the hub, and this error was not discovered until the user tried to hydraulically install the coupling hub. The hydraulic fluid leaked out of the unplugged hole and the hub could not be installed.

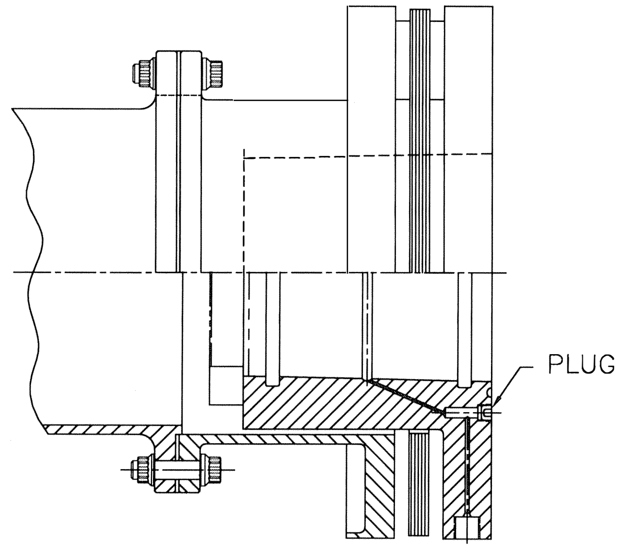


Figure 22. Oil Distribution Hole Arrangement and Plug.

The solution to the problem was not to simply install a replacement plug. It was more complicated than that. The coupling had been precision balanced without the plug installed. To add the plug would mean introducing an unbalance of 2.8 gr-in (the plug weighed 1.4 grams and was located on a 2.0 in radius). This was well above the subassembly allowable unbalance and could have caused excessive vibration if the rotors were very sensitive to unbalance forces. More than that, the coupling had been assembly check balanced, so the entire coupling had to be shipped back to the factory and rebalanced (with the plug installed in the hub) on a rush basis to get the plant back on line on schedule.

CASE 4

The original drive train consisted of an induction motor with a variable speed drive (VSD) connected to an induced draft (ID) fan with a 6-link (3-drive bolt) standard type disc coupling, as shown in Figures 23 and 24. The drive was rated 600 hp at 600 rpm, with a general operating range of 400-600 rpm and a maximum fan demand of 6000 ft-lb. The motor rotor was supported by sleeve bearings and the axial float was restricted by the thrust bearings of the fan and the inherently limited end float disc coupling.

During the first startup, loud "clicking and popping" noises were heard in the vicinity of the coupling. Plant personnel found

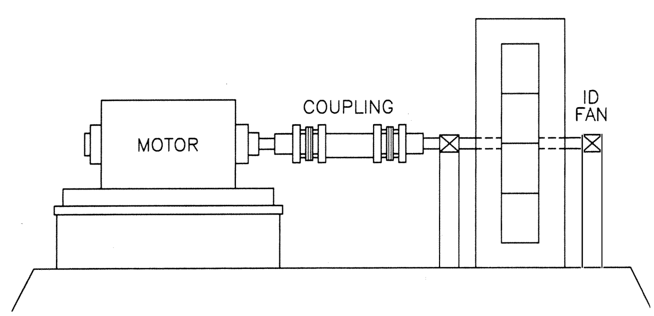


Figure 23. Case 4 Train Schematic.

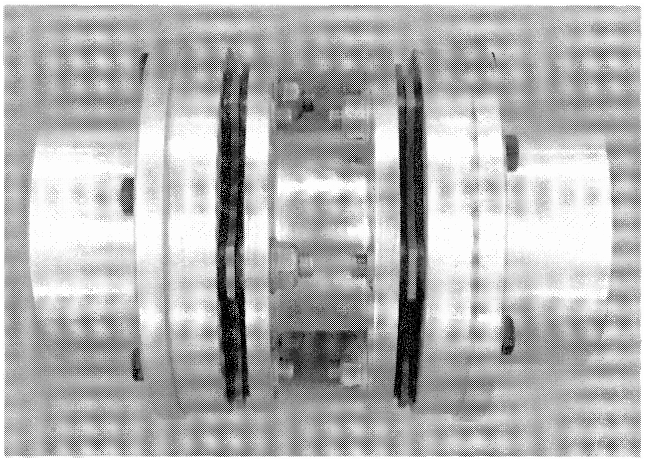


Figure 24. Standard Type Disc Coupling.

that the disc packs in the coupling had developed gaps between discs and looked "buckled."

A new center member was shipped with two new packs and installed. It was reported that the noises occurred again, starting at 500 rpm.

Coupling field engineers were on site for the next startup and confirmed the clicking noise, which appeared this time at 400-420 rpm. Inspection of the coupling revealed that it had developed "tin-canning" (an elastic deformation of the disc packs where the disc pack links are buckled) as in Figure 25. In addition to the noise and the buckled discs, the spacer was shuttling axially between the disc packs during the startup.

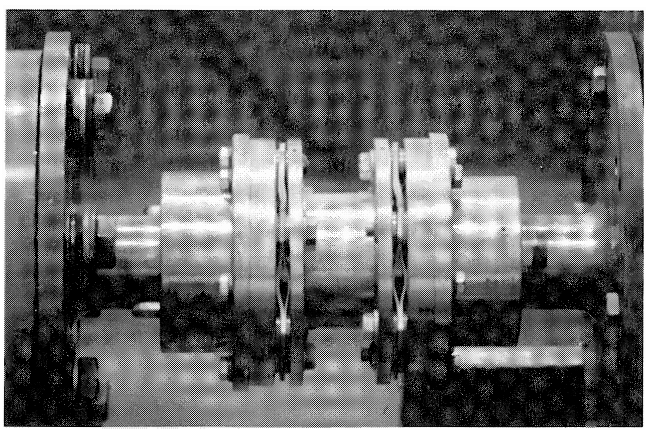


Figure 25. Buckled Disc Packs.

This "tin-canning" generally means the coupling has been overextended axially or has been over torqued. The rating of the coupling (21,500 ft-lb peak capacity) however, should have been enough to handle the torque loading. There was also plenty of axial capacity to handle the axial change in alignment during operation.

It was decided to "beef-up" the coupling in case there was unforeseen torque loading. Fifty percent thicker disc packs and special tight fitting bolts replaced the existing design.

This solution didn't work. With minor differences, the problem persisted. There was the noise at 450-520 rpm accompanied by axial spacer shuttling and buckled discs.

It was found that a torsional analysis of the train had not been performed, mainly because the application seemed routine. But based on the customer's perception of a faulty coupling design, again it was decided to "beef-up" the coupling further and make it axially and torsionally stiffer. This time an eight-link (four driving bolt) coupling was installed which was rated at 4× the expected required maximum load.

Once again the noise occurred, this time at 575-620 rpm and a couple of days later it started at 400-420 rpm. After some analysis, the first torsional resonance of the train was determined to be in the 1600-1800 cpm range, well above the operating speed range.

Then, an axial natural frequency (ANF) of the coupling was suspected, but this was found by calculation to be well above the running speed. Also, every time the disc packs were changed, the ANF of the coupling did too, yet the problem persisted. Note also that because of the nonlinear stiffness and self-damping characteristics of disc couplings, ANF problems are unlikely.

Further developments shifted the focus on the controls of the variable speed drive. Direct drive train runs across the line at 600 rpm, bypassing the VSD controls, were successful. No vibrations or chattering noises were heard and the spacer axial shuttling stopped.

Apparently, the motor-fan-coupling system had been responding to an oscillating torque output of the variable speed drive and motor. The torque oscillation amplitude was approximately 30 percent of the mean torque output with a frequency of about 25 Hz (1500 cpm), regardless of motor speed. This 25 Hz corresponds to the first torsional critical of the train. The torque oscillations were found to be caused by closed loop feedback response of the variable speed drive's voltage regulator. Electronic dampening eventually solved the problem.

The noise generated by the coupling was the result of individual links or groups of links snapping against each other whenever there was a vibratory torque of sufficient magnitude. One of the plant personnel working on the problem jokingly suggested that this "sound alarm" was a nice design feature.

This is another case of a coupling problem being a symptom and not a cause of a train problem. The coupling is an axially passive device. It will not excite itself and must be excited by an outside source.

The other lesson is that it's not usually a good idea to correct a problem by continually modifying the coupling, as was tried repeatedly (4×) in this case with poor results. Much time and money was wasted on this approach, when time would have been spent more wisely searching for other possible causes of the problem.

CASES 5A, 5B, AND 5C

These three cases deal with test stand vibration problems originally thought to be a coupling problem, but eventually found to be caused by parts which mated to the coupling and were made by other than the coupling manufacturer. These parts had been either improperly designed, or mismanufactured, or both.

CASE 5A

The first train in question was a gas turbine/gearbox driven compressor. Between the gearbox and the compressor was a reduced moment style high performance disc coupling designed for a normal speed of about 13,000 rpm. During load testing of the gearbox at the gearbox manufacturer's facilities, excessive vibration was reported.

Previous speed testing had been successful. The test setup consisted of a 2500 hp motor driving the contract gearbox through a speed increaser for the speed test. On the output side of the contract gearbox was one hub / disc pack / sleeve assembly of the contract reduced moment coupling with an attached moment simulator used to simulate the effects of rest of the connection on the shaft end (Figure 26).

For the load test the simulator was removed and the contract coupling half was connected through shop coupling parts to a speed reducer connected to a load applying water brake as in Figure 27.

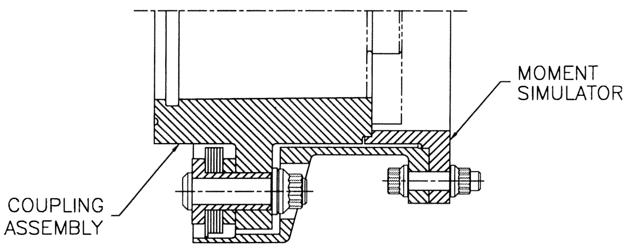


Figure 26. Moment Simulator.

The excessive vibration was reported to be synchronous with speed at levels up to 1.0 mil radial on the contract gearbox shaft and the water brake reducer shaft. The levels also appeared to "shift suddenly" indicating the possibility of a clearance problem. An identical half coupling with simulator from a "sister" project was also tried and the results were similar.

The contract gearbox was "checked out" by the manufacturer, and they felt that there was no problem with the gearbox, but possibly something wrong with the contract coupling design. The half coupling was returned to the coupling manufacturer for inspection and a balance check. No problems were found.

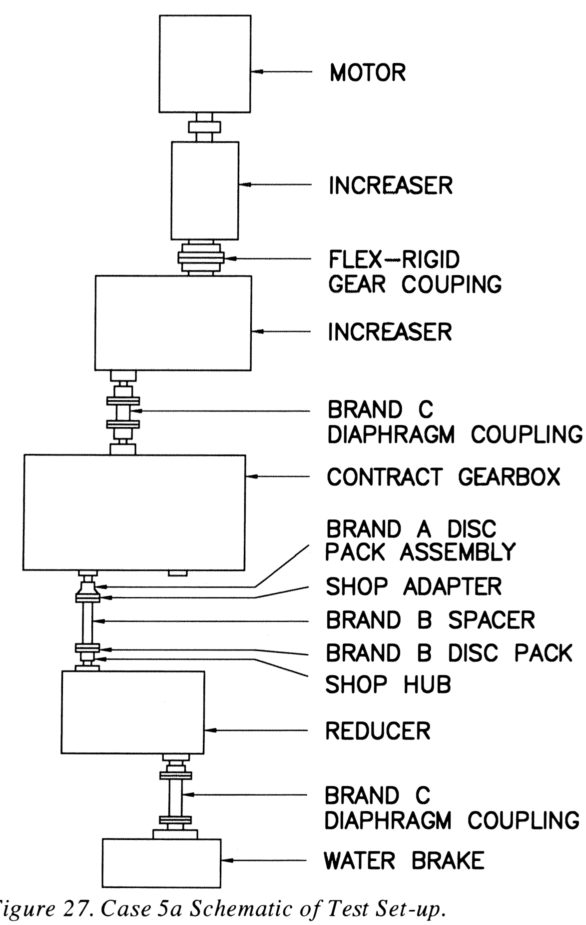


Figure 27. Case 5a Schematic of Test Set-up.

Concurrently, initial measurements were made on the shop coupling pieces, and it was found that there were excessive clearances between the mating parts. The actual "shop" coupling between the contract gearbox and the brake arrangement consisted of the contract half coupling manufactured by brand A, connected to a shop manufactured adapter, connected to a brand B spacer, then a brand B disc pack bolted to a shop manufactured hub on the reducer shaft.

Complete measurements were taken on the shop "coupling" parts, and excessive (by the coupling manufacturer's and API standards) clearances and runouts were found everywhere. The adapter plate pilot to the contract coupling sleeve clearance was 0.002 in, while the plate pilot clearance to the spacer was 0.003 in. (API 671 allows a maximum of 0.001 in clearance.)

The spacer faces ran out 0.006 in to each other, while the spacer pilots ran out 0.0045 in to reference (datum) bands machined into the spacer for balance purposes. In addition, only three non-body fitted bolts attached the adapter to the spacer. The bolts were 3/8 in and the clearance holes were 7/16 in, a difference of about 0.060 in. (API 671 allows only 0.005 in clearance at a piloted connection). The bolts had been tightened without torque wrenches.

It is hard to believe that with this shop coupling rig made up of different manufacturer's coupling parts, and with the excessive clearances and runouts, that the contract coupling manufacturer's parts would be "suspected." But this gear manufacturer's test stand procedure had been to put together loose fitting parts, indicate them in to what they thought were reference diameters, then field balance to make up for any errors. They claimed that they had no problems previously using these methods, and they probably didn't. It was discovered, however, that these particular coupling parts had never before been run together, nor had this particular gearbox ever before gone through similar load testing.

The shop adapter was redesigned and remade by the contract coupling manufacturer, and the other parts were skimmed, reworked and rebalanced to get them in as good a shape as possible. The spacer balance was still not to the coupling manufacturer's required specifications, but in the interest of time, it was decided to go with it, to get the gearbox through testing.

The "coupling" was installed and at no load, full load, and over speed conditions. The contract gearbox passed the API criteria for vibration levels. The levels were still a little higher than the compressor OEM's internal standards, but was accepted by the user's representative, figuring that the actual arrangement with the complete contract coupling would run better.

CASE 5B

Another case of an adapter problem on a test stand occurred in late 1994. The train involved was a gas turbine driven centrifugal compressor rated at 4700 hp at 13000 rpm. The coupling was again a high performance disc coupling which was a semireduced moment type.

Late on a Friday evening the coupling manufacturer received a call that there was a balance problem with the coupling causing excessive vibration on the test stand. The test stand setup was a steam turbine driven gearbox to the contract compressor. This was a "hot" job, so the coupling parts were "hand delivered" to the coupling manufacturer's facilities and an emergency crew was called in to inspect and rework and/or rebalance the coupling if necessary.

Upon inspection, it was found that the test stand coupling was a combination of a 1986 manufactured reduced moment high performance disc half coupling, attached to an adapter plate of unknown manufacture, connected to the contract spacer and compressor high performance disc half coupling, shown in Figure 28.

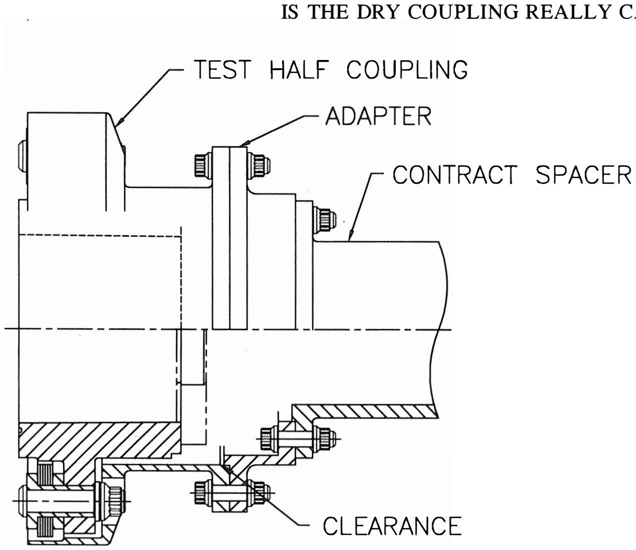


Figure 28. Case 5b Adapter Arrangement.

When measurements were taken on the adapter plate, it was easy to see where the vibration problems were coming from. Excessive clearances and runouts were measured—0.008 in clearance between the adapter and the spacer and 0.008 in runout from coupling bore to adapter pilot.

The adapter plate was reworked with new pilots machined, then the entire "test" coupling was component then assembly check balanced. This arrangement successfully passed further testing.

CASE 5C

The final of the three test stand adapter problems involved significant machinery and coupling damage. The train was a turbine driven speed increasing gearbox to a centrifugal compressor. There was a high performance reduced moment disc coupling between the gearbox and compressor, which had a rated load of 2682 hp at 19,612 rpm.

The contract half coupling, mounted on the contract gearbox output shaft and outfitted with a solo plate designed and manufactured by the gearbox manufacturer, failed in an apparent spin-burst mode during gearbox performance tests.

On the test stand, the gearbox was driven by a variable speed induction motor connected to the gearbox with another manufacturer's disc coupling. The coupling had at first operated for approximately two hours at various speeds up to 21,965 rpm.

Due to high vibrations, the machines were realigned and testing resumed two days later. After approximately 30 minutes of operation at 21,573 rpm, the half coupling failed. The damage is shown in Figures 28 and 29.

Parts of the coupling blew through the hat shaped guard. The gearbox shaft was bent and significantly damaged.

The main cause of the failure was the existence of large unbalance forces which generated considerable stresses which, when superimposed on the existing centrifugal stresses, led to the initial fracture in the coupling sleeve and subsequent damage (Figure 30).

The source of the unbalance was traced to the improper design and manufacture of the customer manufactured solo plate. This plate, which was supposed to act as a moment simulator, was in fact nothing more than a weight simulator, simulating the weight of the unconnected parts of the coupling. (See Figures 31 and 32 for damaged plate and the actual arrangement.) The half coupling was rigidified with cap and set screws.

Since the center of gravity of the half coupling with the plate was significantly altered when the plate was bolted to the rigidified half coupling, the dynamic characteristics of the pinion gear rotor were probably altered. In this case, the center of

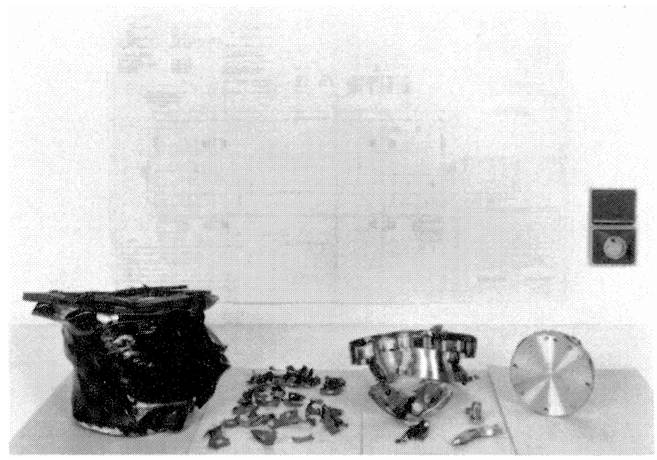


Figure 29. Case 5c Coupling and Guard Damage.

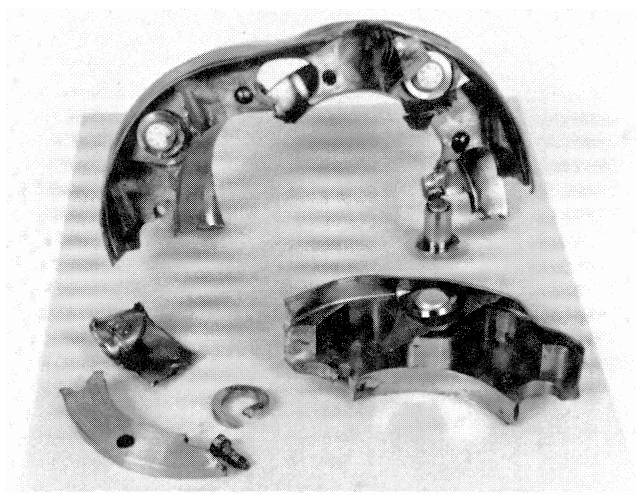


Figure 30. Sleeve Damage.

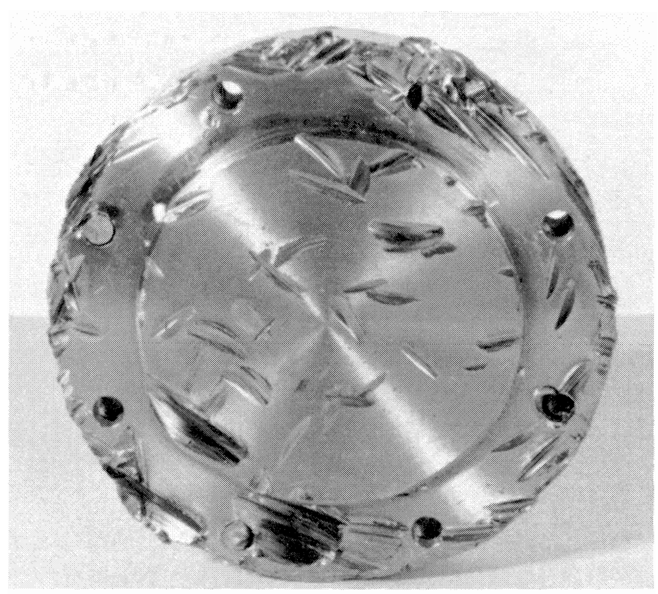


Figure 31. Customer Manufactured Simulator Plate.

gravity of the half coupling with plate was 0.06 in from the end of the shaft (toward the gearbox). The actual contract coupling has a calculated center of gravity of 1.83 in toward the gearbox.

Furthermore, the actual plate was not machined to the customer's drawing. The actual measured pilot diameter was such that there would have been an 0.008 to 0.010 in clearance with the coupling sleeve pilot. The plate was not attached to the sleeve with body fitted bolts but with four capscrews threaded into the plate through the holes in the sleeves.

Even if dialed in (indicated in) and balanced at low speed (1000 rpm) as this plate and half coupling were, it is easy to see that at high speeds any unbalance or vibration could easily have caused the plate to shift and then be the source of the enormous forces required to fail the coupling.

Indentations of the rigidifying setscrews in the coupling hubs attest to these forces, statically determined to be 8000 pounds force to make the largest indentation. Knowing the depths of indentation, as shown in Figure 32, the worst case angle was determined to be 0.32 degrees.

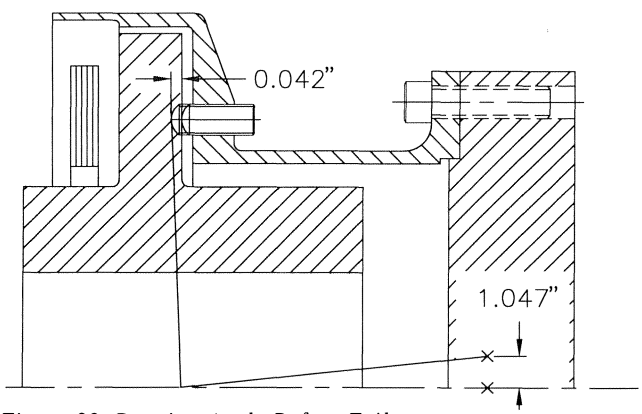


Figure 32. Running Angle Before Failure.

Moreover, the rigidifying screws may have loosened due to the high vibrations, causing further escalation of the failure.

The solution for this case was to supply a double piloted moment simulator plate that was manufactured to tight tolerances and made for zero clearances at the pilots. This plate also was designed such that when bolted to the half coupling, the proper moment was obtained with respect to the gearbox bearing, as in Figure 26.

All of the above three cases, again, point to the importance of keeping the eccentricity-induced (or clearance-induced) unbalance to a minimum and making mating parts tight. But, what makes the eccentricity and clearance such big concerns?

ECCENTRICITY AND CLEARANCE

The possible unbalance of a balanced coupling mounted on the rotor shafts (U_{total}) in most high performance couplings is mainly due to: 1) the residual unbalance from the balance operation (U_{res}), 2) the sum of the unbalances from the eccentricities between the various interfaces of the shaft and coupling component parts (U_{eccn}), and 3) the sum of the unbalances from loose fits between interface pilots and/ or coupling components (U_{clr}) [5].

So, the total unbalance (neglecting bolting unbalance) can be simplified to:

$$U_{total} = U_{res} + U_{eccn} + U_{clr}$$

The authors' experience shows that in many instances, when the unbalance of a coupling is considered, the emphasis is placed on the residual unbalance left after the balancing operation. Other major parts of the total unbalance are sometimes overlooked. However, the unbalance due to eccentricities and clearances can be much greater than the balancing operation residual unbalance.

For instance, a coupling balanced to the lowest possible tolerance on a precision balancing machine can still cause significant vibration if there is excessive runout in the rotor shaft and/or the coupling bore. Furthermore, a perfectly balanced coupling fitting up to an integral flange with a clearance fit pilot may also cause vibration in a sensitive application. This is why the balance operation setup tolerances and the allowable fit clearance tolerances in such specifications as API 671 are so tight [6].

To illustrate the point as far as eccentricities are concerned, a study of over 600 cases of assembly balanced applications indicated that for 85 percent of the cases (assuming perfectly straight and round shafts),

$$U_{res} < 35 \text{ percent of } U_{eccn} \text{ (Figures 33 and 34)}$$

Note that U_{res} came from the allowable balance operation tolerance, and U_{eccn} came from allowable balance machine set-up eccentricity (between the coupling bore and the center of rotation on the balance machine).

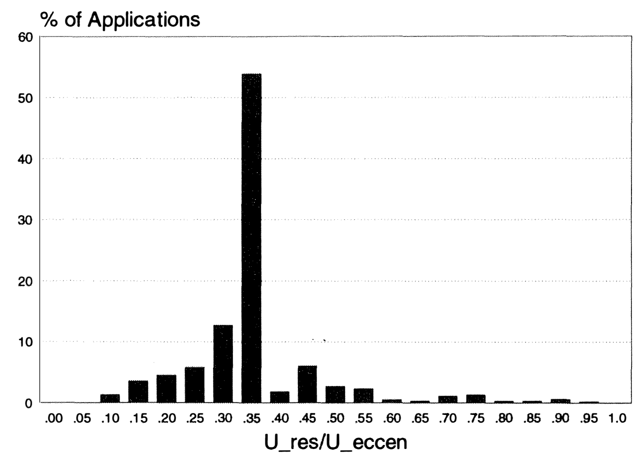


Figure 33. Ures/Ueccn Distribution.

As far as clearance is concerned, Figure 35 gives an indication of the possible magnitude of the unbalance due to clearance

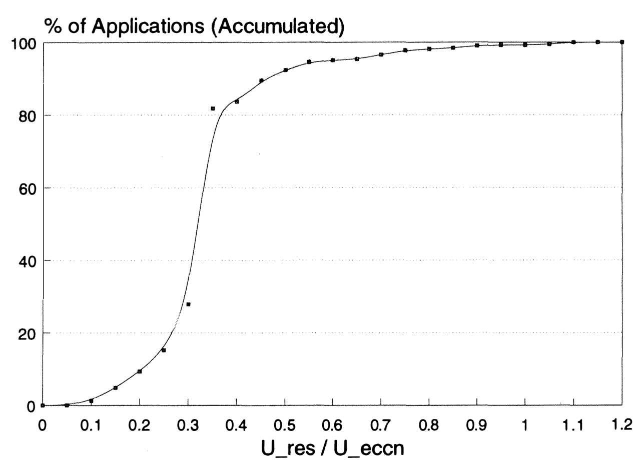


Figure 34. Accumulated Frequency.

relative to the unbalance without clearance in a normalized case where Ures equals 0.35 × U_{eccn}.

The graph is a plot of the percent of the normalized unbalance *vs* a ratio of the coupling component weight affected by a clearance fit (W_{clr}), compared to the total coupling weight (W). This affected coupling component could be, for instance, a coupling center section or spacer connected to a clearance fit pilot to a hub or hub assembly.

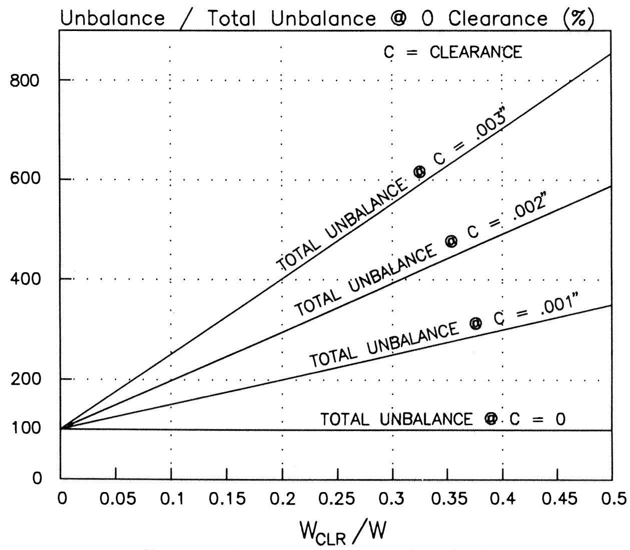


Figure 35. Effect of Clearance on Total Unbalance.

As can be seen, the unbalance due to clearances can be significant. If a coupling spacer weighed 20 percent of a total coupling weight, and the total clearance at the spacer coupling interface was 0.001 in, then, from the chart, the worst case normalized unbalance ratio could be 200 percent. In this case, the actual balance machine residual unbalance would be 17.5 percent of the total unbalance. In terms of actual unbalance numbers, if this coupling were balanced on the machine to 50 μ-in balance tolerance, the total unbalance would be 286 μ-in, with 93 μ-in due to eccentricity and 143 μ-in due to the clearance. Note that these numbers don't include any effects of rotor shaft runout.

As another example, lets look at the instance described in Case 5B, where the adapter was attached to the balanced 1/2 coupling and spacer with an estimated 0.008 in clearance and 0.008 in runout. The unbalance is calculated as shown in Table 3. It can be seen how big the clearance induced unbalance can be (34 × that without clearance). It falls outside the range of Figure 35. In the actual case, by remachining the parts and enforcing the API 671 requirements, the unbalance of the entire coupling dropped to within expected range and the coupling arrangement ran well on test.

This is why excessive clearance and eccentricity are serious considerations when it comes to unbalance. It is recommended that, whenever possible, light interference pilots be used to ensure minimum unbalance, and bore and shaft runout tolerances be kept to a minimum.

CONCLUSION

Couplings have long been recognized as one of the critical components of high speed and/or high power trains, due to their

Table 3. Case 5b Unbalance Calculation.

	Unbalance (g-in)	Mass Shift (μ in)	Unbalance Distribution based on Total Unbalance @ 0 clearance
Residual Unbalance (API 671)	.37	50	25 %
Eccentricity Induced unbalance	1.08	150	75 %
Total Unbalance without clearance	1.45	200	100 %
Clearance Induced unbalance	48.3	6690	3330 %
Total Unbalance with clearance	49.8	6890	3430 %

influence on the train lateral, torsional, and axial vibration characteristics. They are many times seen, rightly or wrongly, as the cause of undesired train vibrations. This seems to be particularly true when there is a pressure to bring the train back to operation; the coupling, hoped to be the cause, represents a relatively low cost and quicker turn around solution than many others. Also, lack of understanding of coupling mechanics, especially those with which many are unfamiliar, can tend to make someone see the coupling as the problem.

It is important to keep in mind, however, not to overlook what effect the coupling can really have on vibration. If a coupling is likely involved, it is important to work with and provide all necessary information to the coupling manufacturer to help in identifying the cause(s). A coupling manufacturer who analyzes and tracks field problems provides a valuable knowledge base that can be very helpful. Modification of a coupling on a turbomachinery train without the coupling manufacturer's review and approval can lead to unwanted trouble.

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