AN EFFECTIVE WAY FOR MAINTENANCE REDUCTION AND VIBRATION ELIMINATION OF PEAKING GAS TURBINES

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ABSTRACT
A US power generation company had 36 similar high heat rate gas turbines in two of their plants. These units were used for peaking power in a large metropolitan area. On two of these units in the early 1990's they tried switching from lubricated, gear-type couplings to single, machined diaphragm-type couplings for reduction of maintenance and improved flexibility. The gear couplings also caused a great deal of vibrations during startup, and many times units were shut down because of excessive vibrations.

Due to the extreme thermal growth of these applications, along with the stop-start nature of the application, the couplings were required to handle enormous amounts of misalignment. While the standard diaphragm couplings seemed like a good choice for the application because of their large misalignment capability, they proved to be unable to handle the excessive offset movements caused by thermal growth. The problem with the standard diaphragm was that it was too stiff or in other words, the spring rate was too high. In one case the coupling failed catastrophically very quickly under normal running conditions.

In the mid 1990’s the power company tried a different, unique style of diaphragm couplings that used multiple diaphragms with holes in them. The diaphragms were made of thin, uniform thickness steel instead of the more common machined ‘profile’ diaphragms. Because of the holes, these new diaphragms provided much more misalignment capability, and they also had far lower spring rates than the previous diaphragm couplings. The lower spring rates caused an enormous reduction in reactionary forces from the coupling, and they also eliminated many vibration problems caused by the original gear couplings.

The overall effect of this discovery was for the company to begin changing all similar units to this type of coupling. Now these 36 units which previously had severe vibration problems during startup, run very reliably. They have also eliminated the need for maintenance, which was an issue with the original gear couplings.

INTRODUCTION
In the early 1990s, a large northeast utility experienced an unusual coupling failure on a 45 MW gas turbine expander-generator set. The application was a Worthington ER-224 expander coupled to a 45 MW air-cooled generator. These units were built in the late ‘60s, early ‘70s, and into the ‘80s, and used FT-4 Pratt and Whitney engines (actually, the gas generator section) exhausting radially inward into the hot gas expander. The gas path was directed from the radial thrust of the jet engines axially through the expander and then radially outward and exhausting to the air.

All these units were built near the advent of jet engine and gas turbine usage in the power industry, and these particular units were built on the premise that the expander be coupled to a generator. The driving force came from two jet engines, which produced the thrusting hot gas. The jet engines were ordinary
jet engines that would have been found on any Boeing 707. There are literally thousands of these engines in existence.

The simplicity of the set-up was a selling point because any engine could be quickly and easily replaced with another ordinary engine from the local airport. The engines could be easily altered for this application, and there were a number of shops that would readily modify the jet engines for use in these units.

These Worthington expander units are installed throughout the U.S. and, in fact, there are units located around the world. They are not particularly efficient units in terms of heat rate, but they were built for fast and easy start to provide a significant amount of power on short notice. Thus, they were bought and utilized, and continue to be utilized today, largely in peaking service when there is a high demand, and cost of power production is not the immediate issue.

There were two distinct configurations built. The first was what we might call a single-ended unit. That is, it had one expander using two jet engines driving a 30-45 MW generator. The second configuration used two expanders coupled on each end of a 120-180 MW generator. These multiple-expander units were often referred to as Hi-caps. Figures 1a and 1b show the two configurations.

Peaking units are particularly difficult from an alignment standpoint because they are started and run for only a few hours at a time. Thus, thermal position of casings and bearings of a driver to a driven piece of machinery are always changing. This start and stop cycling on any large machine system is usually a difficult operating scenario, especially on couplings and bearings.

**PROBLEM**

Virtually all of these expander units were manufactured and sold with a gear coupling between the expander and generator shaft. There is approximately 18 inches between shaft ends. Both shafts are approximately 10 inches in diameter, with a single or double key. In most of these cases, the gear couplings were continuously lubed by the pressurized bearing oil system.

The Worthington units have a somewhat unusual mounting arrangement. The casings are centerline mounted, but there is an anti-rotation pin underneath the middle at the bottom of each unit. Since these units were mainly used for peaking service, the regular heat-up and cool-down cycles produced growth and contraction issues. Vertical, horizontal, and axial growth proved to be rather substantial. In other words, the expander was expected to grow approximately 0.045 inches upward, and there were different amounts at the coupling end versus the opposite end. In addition, the shaft of the expander heated and grew axially. The casing itself would grow, but at different rates than the shaft. In a normal set-up, it would have been preferable to align the expander and the generator to grow predictably into an approximately aligned condition; however, because of the relatively short duty cycles, it is quite possible to have rather radical movements of the turbine with respect to the generator. If the turbine is running for only one to three hours, it may not reach equilibrium until near the end of the running schedule, if at all.

The predicted nominal cold offset for these units, as stated above, is normally about 0.045 inches low on the turbine with respect to the generator. That is, it is presumed that the expander shaft will rise upward about 0.045 inches with respect to the generator. It is interesting to note here that actual measurements of growth on these ER units differ on units around the country. It appears that the turbine grows both vertically and horizontally and with different magnitudes at each bearing. It also appears that each unit may have its own growth signature, and each unit’s history may change from day to day. The large anti-rotation pin located beneath each of the expander unit’s casing, as well as the other alignment guides, need to be kept free and lubricated to aid in casing movement and growth.

The axial growth is another interesting event. There is a single thrust bearing located in one end of the expander. Since the casing and shaft grow at different rates, the actual distance...
between shaft ends changes. Generally, it is estimated that after the turbine has heated to normal steady-state conditions that the distance between shaft ends will shrink by about 3/16 of an inch. Thus, the coupling is usually pre-stretched by this amount. But, because these are peaking units, a unit may be started, brought up to speed, and synchronized to line before everything is warm and at steady-state conditions. Thus, it is desirable that the axial spring-rate of the coupling be rather soft (low) so that the generator may run on magnetic center even though the expander is not at steady-state conditions.

The Hi-cap units had some other unique issues. The sleeve-bearing generator nested between the four expanders would be pulled and pushed from both ends depending on which expanders were fired. Expanders on one end of the generator could have their thrust bearing away from the generator shaft, while on the other end, the expander thrust bearing was adjacent to the generator. This posed additional axial growth differences, especially when one, two, or three expanders were operated instead of all four and eight engines. It would be possible to have, for instance, one or two outboard expanders delivering power while the inboard units merely transmitted torque and did not heat up. Obviously, large misalignment conditions could exist even if the proper growth conditions were anticipated.

As a result of the operational misalignment on these machines, the gear couplings regularly had to take a large amount of misalignment. These couplings would wear and, oftentimes, the teeth gradually became worn to the point that vibration due to the couplings would increase. The gear couplings could lose balance or have a tendency to become locked up. After a period of time, couplings would have to be replaced or refurbished. This process, hopefully, would reduce the vibration due to gear tooth wear.

At some point in the late ‘80s, utilities tried to go to non-lubricated couplings instead of gear couplings. The approach taken was to leave the large hubs intact and merely install a machined-diaphragm coupling between the flange faces of the two coupling hubs. Thus, the marine-style gear coupling was merely removed, and a single machined-diaphragm spacer coupling was installed.

A number of power companies experimented with these machined-diaphragm couplings in this application. This was marginally successful, but there were several cases where single machined-diaphragm couplings cracked and produced serious failures. In one case, a diaphragm started to crack, and the crack rapidly propagated circularly around the metal. The tearing of the metal under torque and high speed twisted the spacer severely off center. The other diaphragm at the opposite end bent and broke, and the free spacer flew out of the machine. Fire protection lines were cut and a troublesome fire ensued. In another case, at a northeast utility, a coupling with only 90 hours of service cracked shortly after being brought up to speed after a period on turning gear operation. In that case, one end cracked, and the crack propagated so that the coupling tilted and pried the turbine and generator apart. Both of these cases required significant repair work and were costly rebuilds.

In the early 1990s, after the costly failure, that large northeast utility came to Coupling Corporation of America (CCA). This utility had seven of these units, all with machined-diaphragm couplings. They asked CCA to make a more flexible coupling that could regularly withstand the large amounts of thermal offset growth, as well as axial growth. Through their studies, the utility came to the conclusion that low-cycle stress fatigue was excessive on the machined-diaphragm couplings and that it was probably too excessive an offset, even during turning gear operation, such that the stress induced by the bending fatigue at very low RPM and no torque was sufficient to initiate a crack. When a turbine was brought up to speed and torque applied, the crack propagated around the entire diaphragm.

The utility requested CCA to provide a different-style coupling that would accommodate the misalignment, be more predictable, and permit inspection of its mechanical integrity in the course of its duty cycle. At the same time, other utilities that had experienced the same maintenance worries with the gear couplings and the questionable reliability of machined-diaphragm couplings were also searching for a solution.

**SOLUTION**

When approached with the specific requirements, CCA was confident that its FLEXXOR diaphragm design would be able to transmit the required torque at the speed of these units, as well as be able to handle the axial growth and the angularity induced as a result of the offset of the shafts.

The flexible part of the FLEXXOR coupling is the unique spoked diaphragm. These diaphragms are designed in such a way that they form a spoked membrane, and torque is transmitted through the spokes much like tension in spokes of a bicycle wheel. A coupling diaphragm appears to be a flat sheet with a lot of holes. In reality, however, the diaphragm allows multiple paths of torque transmission. In fact, if one spoke were to fail, a crack would not propagate; instead, it is stopped at the adjacent hole.

Each coupling assembly uses a set of diaphragms at each end of a tubular spacer. Thus, there are two planes of flexure. This allows a coupling to accept offset, as well as angular misalignment between the two shafts. Axial expansion, or movement of one shaft with respect to the other, is accommodated by the umbrella deflection of the diaphragms.

A FLEXXOR coupling has multiple diaphragms in parallel. For example, on these 800- size couplings, capable of about 1.6 million lb-in torque, there are 4 stainless steel diaphragms in
parallel separated by thin spacers. 24 sets of holes form the 24-spoke pattern. Thus, there are essentially 4 x 24, or 96, paths of torque transmission. The units are overtorqued as a stress relief operation. The couplings are dynamically balanced but are also provided with balance correction holes for field trim balance.

The axial and angular flexibility of these FLEXXOR couplings is such that the coupling can be flexed by simply applying hand pressure. Thus, separate installation hardware to squeeze a FLEXXOR to put it in position within its rabbit fits is not required.

The low spring rates correspond to the fact that these thin tension-stressed diaphragms can accept an unusual amount of angular and axial deflection without becoming overstressed. The major stress in the diaphragm is due to torque; consequently, of the combined stresses of torque, bending and fatigue, torque predominates. Therefore, with the large misalignments witnessed in these peaking unit applications, the bending stresses, which cause fatigue, are not nearly as critical a factor as in a machined-diaphragm coupling design.

Another interesting feature is that these couplings can be easily inspected for cracks while in place. A simple visual inspection is all that is necessary to detect crack initiation, with cracks at the edges of the inner diaphragms also visually detectable through the holes. This can actually be performed using a strobe light with the coupling spinning. (We are not advocating this for obvious safety reasons, but, in fact, we have inspected these couplings and other FLEXXOR couplings this way on occasion.) The more likely inspection method is with the unit at rest or, at worst, on turning gear.

In order to simplify the installation at the northeast utility, CCA decided not to replace the existing gear coupling hubs, but merely adapt the FLEXXOR element to those hubs. An intermediate ring adapter was designed to bolt on to the hub flanges of each shaft using the existing gear coupling mounting bolts. A second set of holes on the ring adapter was the attachment for the FLEXXOR spacer element.

In another utility, where Hi-caps were used exclusively, the requirement was for supplying ready stand-by reserve. These units were intended to start reliably and supply power to the Northeast grid. With the gear couplings in place, there was a regular guessing game whether the units would come on line or be kicked out due to high vibration. Often the gear couplings would “lock up” and cause high vibration at one of the bearing locations.

Another utility in the south operates a fleet of 36 Worthington ER-224 units, all originally equipped with gear-type couplings. As these couplings wore, the resulting vibration and/or bearing overload would affect generating capacity. Whether these couplings were refurbished or replaced, unit performance remained unsatisfactory.

Several machined-diaphragm couplings were tried, but after an undetected diaphragm crack resulted in this unit sustaining major fire damage, the utility became more wary of this type of coupling. After learning of the successful operation of the multiple-diaphragm FLEXXOR at the northeast utility, the southern utility followed suit by initially trying three couplings.

Extensive testing of these couplings indicated that the low angular stiffness of the FLEXXOR design essentially nullified the forces known to cause the previous vibration and/or bearing loading problems.

One co-author of this paper has much experience in measuring misalignment, movements, and vibrations on these Worthington ER-224 units. Relative off-line-to-running elevation change was measured using proximity probes mounted to water-cooled stands, and corroborated with Essinger bar measurements. Although a typical measurement session is depicted in Figure 2, significant variation in elevation change was noted among the units. These variations were attributed to differences in localized heating of the expander’s support columns; however, these variances are believed to be within a range that is normally anticipated when reasonable methods of control are applied.

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Figure 2: Alignment data between Generator and Expander

It follows that the coupling between the expander and generator is likely to experience variable and, sometimes, unpredictable amounts of flexing during its life.

**FINDINGS**

The southern utility was able to get some excellent vibration and temperature data showing the difference between the original coupling and the FLEXXOR coupling. The data
confirms three problem areas with the equipment. The first is oil whip on the generator bearings. The second is increased bearing temperatures. The third is heavy vibrations due to the harmonics of the system. The data in the appendices show these problems very clearly with the gear coupling in place, and then the problems almost vanish completely with the change to the FLEXXOR. The following list explains what each appendix shows.

Appendix A has two documents (A-1 & A-2) that were taken in the 1980’s and provide a good baseline view of a typical running condition on the inboard generator bearing. A-1 shows a waterfall plot where x is frequency, y (vertical axis) is vibration magnitude, and z (depth axis) is time. The vibration on the low end of the frequency scale that gets larger with time happens to be at the first natural frequency of the generator. A-2 is a time slice of A-1, showing the last data time line from A-1, where the peak vibration is highest. The A-1 graph shows that the vibration is over 4 mils, and is found at 1170 RPM. The behavior of this vibration shows a clear indication of oil whip.

Appendix B has three sets of paired data, all coming from the same machine. The pages are labeled B-1 through B-6. In each set, the first data sheet is taken on June 14, 1995 with the gear coupling in place. The second data sheet in each set is taken two weeks later with the FLEXXOR coupling in place. This data clearly shows some serious oil whip problems with the gear coupling installed, and virtually no problems with the FLEXXOR coupling installed.

B-1 shows hand-written data from a running session lasting almost three hours. In the ‘Exp I’ column (Expander Inboard) in the ‘BRG Metal Temp’ group, it is plain to see that the temperature of the expander inboard bearing steadily increases while running. At the same time the vibration data also shows a steadily worsening condition. Eventually at time 1835 the vibrations were bad enough to trip the system at 3 mils. B-2 shows the same type of data taken with the FLEXXOR coupling in place. It is easy to see that at full load the bearing temperatures and the vibrations were constantly at very safe levels. It should also be noted that there was no alignment done before installing the FLEXXOR coupling. The note at the top right shows that the alignment was actually worse than would usually be tolerated.

B-3 is a polar plot showing the shaft centerline of the generator drive end. At startup in the no-load condition (NL), the shaft goes to the lower-left quadrant. As the unit gets up to base load (B) which is 32 MW, the shaft starts moving dramatically away from the lower-left quadrant. At one hour (1) the shaft is already starting to move up towards the top quadrants. By two hours (2) the unit is well on its way to tripping at 2.5 hours. In B-4, with the FLEXXOR installed, the unit starts up and goes to its base load position, and it continues to sit there for the entire run. For the direction of rotation, the lower-left quadrant is the stable running location for this bearing.

B-5 shows a vibration analysis of the time just before the unit tripped out (as shown in B-1). There is a 22 mil vibration at 1080 RPM, a 1.4 mil vibration at 3600 RPM (running speed) and a 1.23 mil vibration at 7200 RPM. B-6 shows the massive improvement with the FLEXXOR installed. The vibrations at every level virtually disappeared.

Appendix C is a different unit that had a different type of problem. This unit did not have bearing issues, but instead had general harmonic vibration problems. As seen in C-1, there is a significant vibration of 2.0 mils at the running speed, 3600 RPM. At twice the running speed (7200 RPM) there is a much higher vibration of 4.3 mils. C-2 shows the same unit two weeks later with the FLEXXOR installed. The vibrations at every harmonic frequency dropped significantly.

Since the original installation, the utility in the northeast has converted all of the same units to the CCA coupling design. The change has resulted in elimination of virtually all maintenance on the coupling, and the equipment runs consistently with almost no vibration. Figure 3 shows a picture of an installed FLEXXOR coupling.

Figure 3: FLEXXOR coupling installed between a 160 MW generator and Worthington Expander (Hi-cap)

Other utilities with these units have also made the same change to their equipment with the similar results. Utilities all across the country have, in fact, made the FLEXXOR coupling retrofit the de facto standard for these Worthington ER units, both in the single-ended models and in the four-expander Hi-cap configuration. Well over 60 of these couplings have now been
installed. The same basic design is supplied with or without electrical insulation. They can be made with limited end float. One has been supplied with axial damping for one generator, which tended to hunt.

CONCLUSION
To conclude, the FLEXXOR coupling has been able to successfully accept a rather large and variable misalignment on these Worthington ER peaking power combustion units. It is the hole-filled, thin diaphragms with the very low spring rates that produce very low forces and bending moments on the shafts and bearings of the coupled machines. This permits the driver and driven elements to operate in their independent, natural and stable shaft to bearing orientations. The result is that the two coupled machines operate with lower vibration and less maintenance problems. These machines can now be depended upon to reliably start and run at minimal vibration. This has resulted in much needed reduction in maintenance and eliminated the need for coupling lubrication and maintenance. The end result is that the utilities have saved many dollars and have improved their online response time and reliability.

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Appendix A-1: Waterfall plot with original gear coupling. Frequency on bottom axis, time on depth axis. 1X is running speed (3600 RPM) and large peak is 1/3 running speed.

Appendix A-2: Vibration data with original coupling. This time slice is the top line from A-1.
Appendix B-1: Temp and vibration data with gear coupling. Note the dramatic temp rise in the expander inboard column along with increased vibrations.

Appendix B-2: Data from same equipment with FLEXXOR coupling.

Appendix B-3: Shaft centerline movement in generator bearing with gear coupling. Circled numbers represent running hours.

Appendix B-4: Shaft centerline movement in generator bearing with FLEXXOR coupling.
Appendix B-5: Vibration data with gear coupling. Peak of 22.0 mils is at 1/3 of running speed (~1200 RPM).

Appendix B-6: Vibration data with FLEXXOR coupling.

Appendix C-1: Vibration data showing problems at the harmonics, especially 2 times running speed.

Appendix C-2: Major reduction in vibrations with FLEXXOR coupling.