AN EXPERIENCE WITH NONSTANDARD COUPLING APPLICATION

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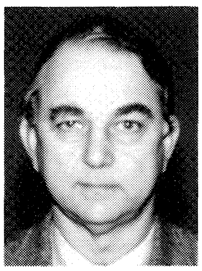

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ABSTRACT

The selection of flexible couplings is often considered to be routine because there are a variety of proven designs available. A compressor retrofit that required variance from standard coupling design practice is discussed, along with the problems that occurred, and how the problems were resolved.

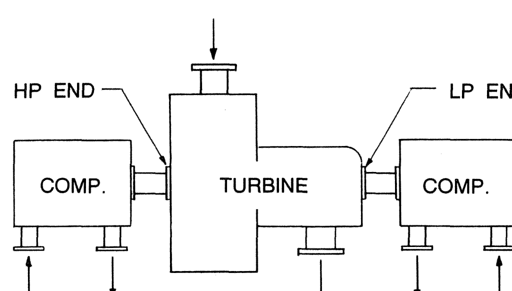
In the early 1980s, two new compressors with a drive turbine were retrofitted to an existing foundation. Marine type gear couplings were applied and have subsequently exhibited hub to shaft galling, fatigue failure of the rabbet pilot, abnormally rapid wear of the coupling teeth, and coupling bolt failure by hydrogen embrittlement. Changes to the existing design have helped, but the final solution will be installation of a different style coupling.

INTRODUCTION

The turbine and compressor designer can choose between a variety of flexible couplings with proven performance. Coupling selection and application is straightforward and routine—unless application needs deviate from standard practice. What can and did go wrong because the application required subtle differences in design is discussed.

In the early 1980s, a process unit at Du Pont's Beaumont, Texas, site installed several new turbomachinery trains as part of a process modernization. One of these new trains experienced several unusual drive coupling problems. The train consisted of a 40,000 hp, 9,000 rpm steam turbine driving two compressors, one on each end of the turbine (Figure 1). The train was installed on an existing foundation, which imposed severe space limitations on the drive couplings. The turbine had integral hubs on each end, and the thrust end hub was the outboard side of the thrust disk.

Gear couplings, size four, were selected because of past success in similar applications at similar speeds and torques. The compressor coupling hubs were designed to be mounted hydraulically, because the potentially volatile environment does not permit heating the hubs for removal or installation in the field. Because of



TURBINE AND COMPRESSOR CASES

Figure 1. Entire Compressor Train.

the integral turbine hubs, marine style gear couplings were used. Marine couplings have the teeth on the spacer rather than on the coupling hub. To be consistent, the compressor ends of the couplings were also marine style. The thrust end of the turbine had an integral flange that was larger than normal to accommodate both the thrust disk and the bolt circle for the coupling (Figure 2). The bolt circle for the flange was 13 1/2 in, compared to a normal 9 11/16 in, thus requiring a special sleeve design. Because of space limitation, the integral turbine hubs were located within the turbine bearing housings, as shown in Figure 3. The top half of the turbine bearing housing had to be removed to assemble the coupling, as can be seen in Figure 4.

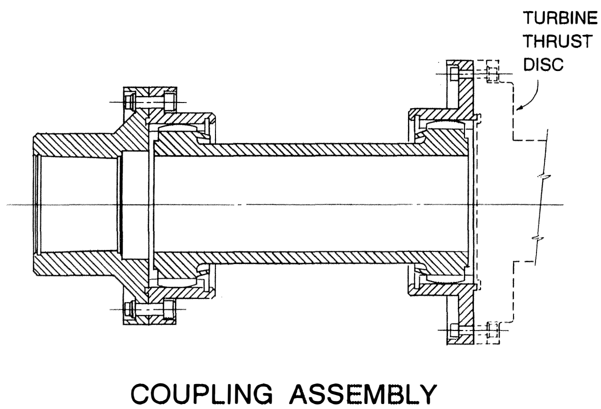


Figure 2. Turbine Thrust End Flange and Coupling.

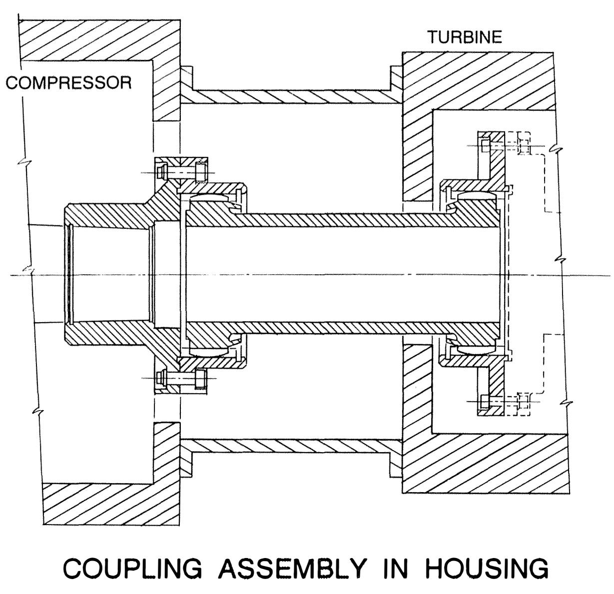


Figure 3. Turbine Hubs within Bearing Housing.

Coupling lubrication was by oil sprayed into antisludge design gear meshes (Figure 5). The turbine manufacturer provided the spray nozzle for the turbine end, and the compressor manufacturer provided the spray nozzle for the compressor end.

Coupling problems began even before the initial startup. The O-ring groove in the compressor coupling hubs was not a standard design, and neither metric nor American standard O-rings would fit (Figure 6). The "best-fit" O-ring was selected and used for the first installation. When one of the compressor hubs galled during

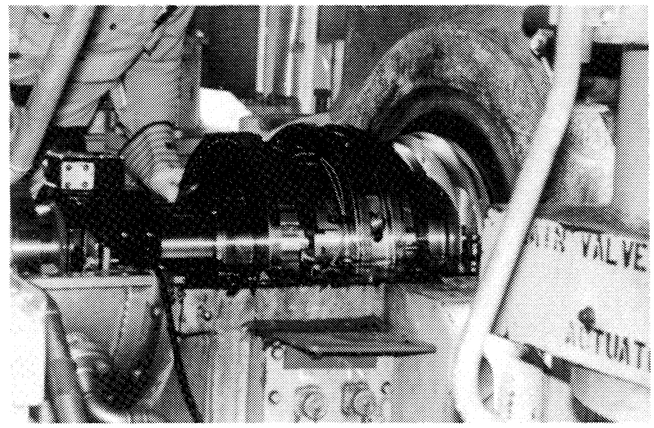


Figure 4. Bearing Housing Top Half Removed for Coupling Installation

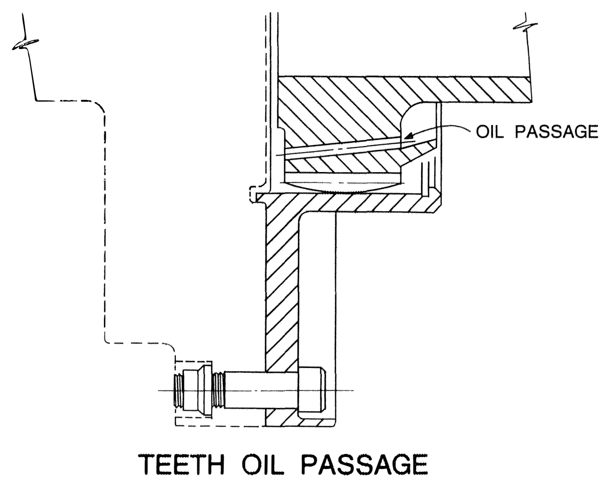


Figure 5. Coupling Antisludge Design Feature

installation, it became a concern that a problem of O-ring leakage, due to its being the wrong size, could make the coupling hub nonremovable in the field.

When the coupling was installed, the turbine flanges were found so close to the bearings that it was not possible to get a torque wrench on the nuts for the coupling bolts, and it became necessary

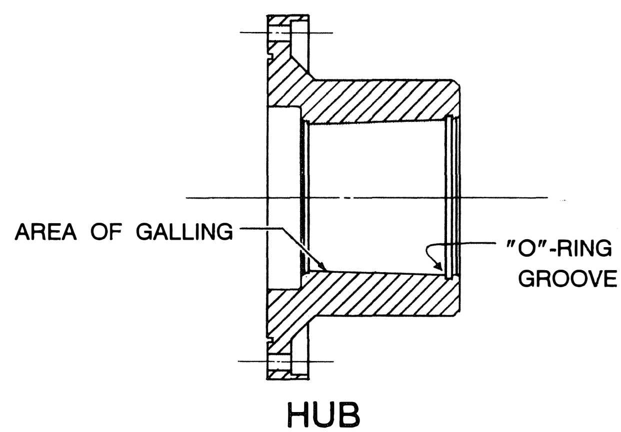


Figure 6. Compressor Hub, "O"-Ring Groove, and Galled Area.

to torque the bolt heads. There was some concern that the potential for additional frictional forces on the bolt body could result in a less accurate torque setting than obtained by torquing the bolt nut. The coupling bolts were installed with the bolt heads on the spool side of the coupling to maximize bolt bearing surface on the thin flange of the sleeve.

An inspection in 1983, after one year of operation, revealed significant wear on the coupling teeth adjacent to each end of the turbine (Figure 7). The couplings were replaced, and one year later an inspection revealed the same rapid wear problem. No corrective action was taken because the plant was mothballed due to poor product demand.

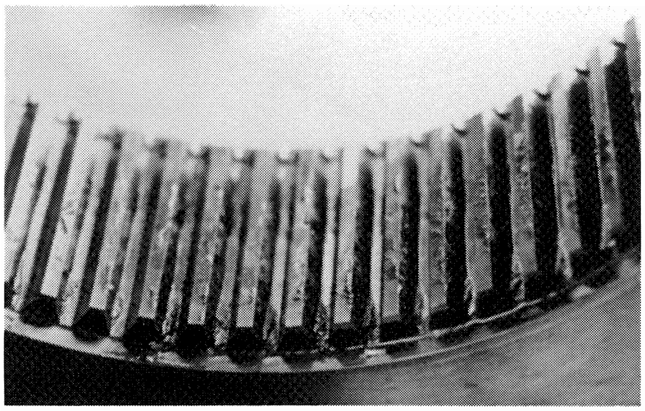


Figure 7. Worn Coupling Gear Teeth.

New couplings were installed when the plant was returned to production in 1986. After 2½ years service, the coupling sleeve rabbet ring at the thrust end of the turbine was found to have failed by fatigue (Figure 8).

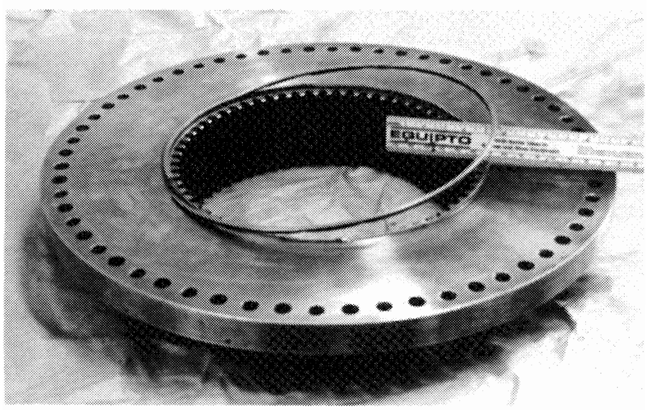


Figure 8. Coupling Rabbet Failure.

In February 1989, one of 30 3/8 in coupling bolts sheared off near the bolt head.

The variances from standard design that led to all these problems were:

· integral coupling flanges on the turbine tight against the bearings.

· oversized coupling sleeve for the thrust end of the turbine.

· thick outboard flange on hydraulically mounted compressor hubs.

· coupling oil spray nozzle on compressor side supplied by compressor vendor, and spray nozzle on turbine side supplied by turbine vendor.

DISCUSSION OF PROBLEMS

Hub Galling

The most persistent, long term problem with the coupling hubs of both compressors has been galling during installation and removal. To do a normal hub installation and removal, the hub is expanded by injecting pressurized oil between the hub bore and compressor shaft. The hub is positioned on the shaft and pushed by hand onto the shaft taper until the O-rings form a seal. A portable hand-operated hydraulic pump is connected to a predrilled oil passageway in the shaft (Figure 9). The hub is then hydraulically expanded and pushed further onto the shaft taper until it reaches a predetermined hub location. The final location depends on shaft taper and required hub-to-shaft interference for torque transmission by friction. For example, with a ¾ in on diameter per ft taper, a 0.002 in per in of diameter shrink requirement, and a 3.0 in diameter shaft, the hub needs to travel axially 0.096 in ($3.0 \times 0.002 \times 12.0/.75$) beyond the metal-to-metal contact [1].

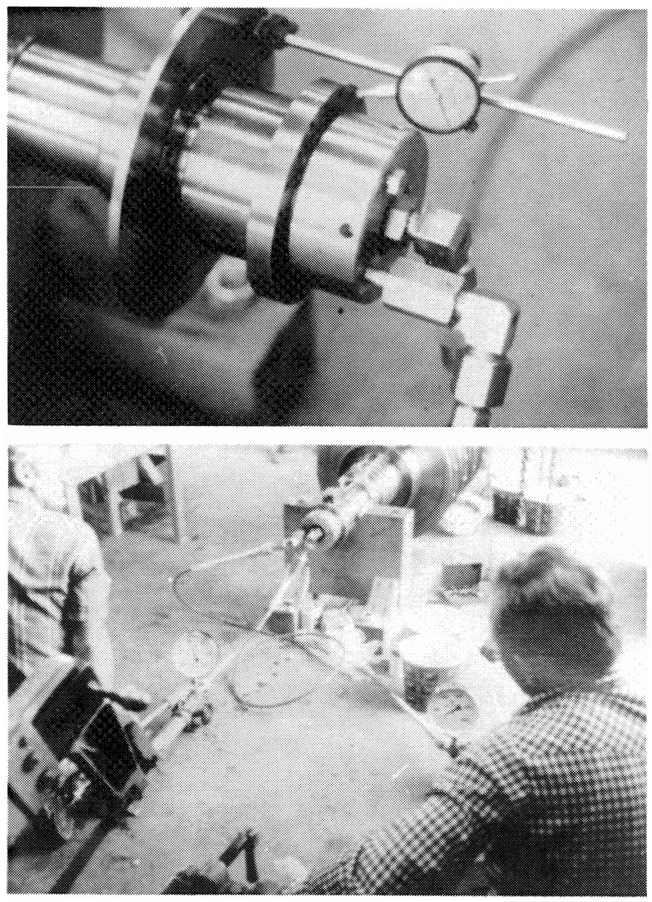


Figure 9. Hydraulic Rig for Coupling Installation.

The space limitation of the installation required that the outboard flanged face of the mounted hub be closer to the shaft end than normal. This large flange so close to the shaft end inhibited proper bore expansion during installation. As a result, the outboard end of the hub could drag and gall as it was pushed onto the shaft (Figure 6). The hubs were installed and removed 12 times with galling occurring on three occasions. So far, the damage has required only light dressing of the shaft, but the possibility for greater damage is a concern.

Attempts at repairing the galled hubs were unsuccessful. The 0.001 in deep gall marks in the hub bore were removed by regrinding the bore. Unfortunately, even this small change in the bore diameter caused the final hub location to move too far onto the shaft, exceeding the allowable axial spacing for the coupling spool piece. For the taper involved, this 0.001 in increase in bore diameter moved the hub 0.030 in further onto the shaft. This was unacceptable, and new hubs were needed. The spare couplings were installed, but later removed prior to the November 1982 startup in favor of new instrumented (torque monitoring) couplings, which arrived just in time for the restart. Since the initial startup, the galling problem has resulted in three hubs being scrapped because they advanced too far onto the shaft after grinding.

A finite element analysis was done to model bore expansion during hub installation with hopes of finding a possible modification to stop the galling (Figure 10). No modification was found that satisfied all concerns.

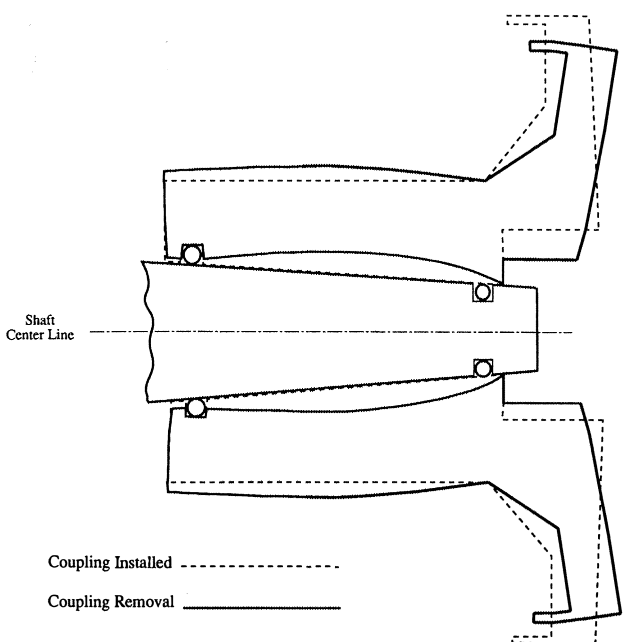


Figure 10. Hub Expansion from Hydraulic Pressure.

The following changes were made in attempts to alleviate this problem:

• During hydraulic installation, the hub was now pushed onto the shaft in steps separated by one minute pauses to allow for bore expansion and complete oil dispersion. A higher maximum pressure for hub bore expansion during installation was also put in use. The coupling vendor increased this by 20 percent to approximately 25,000 psig.

• The maximum pressure for hub removal was raised by 30 percent to 28,000 psig.

• The shaft end was chamfered in the area where galling occurred as per vendor recommendation (Figure 11).

Coupling Tooth Wear

During the first shutdown in 1983, the gear teeth on the turbine side of both couplings were found to have significantly more wear

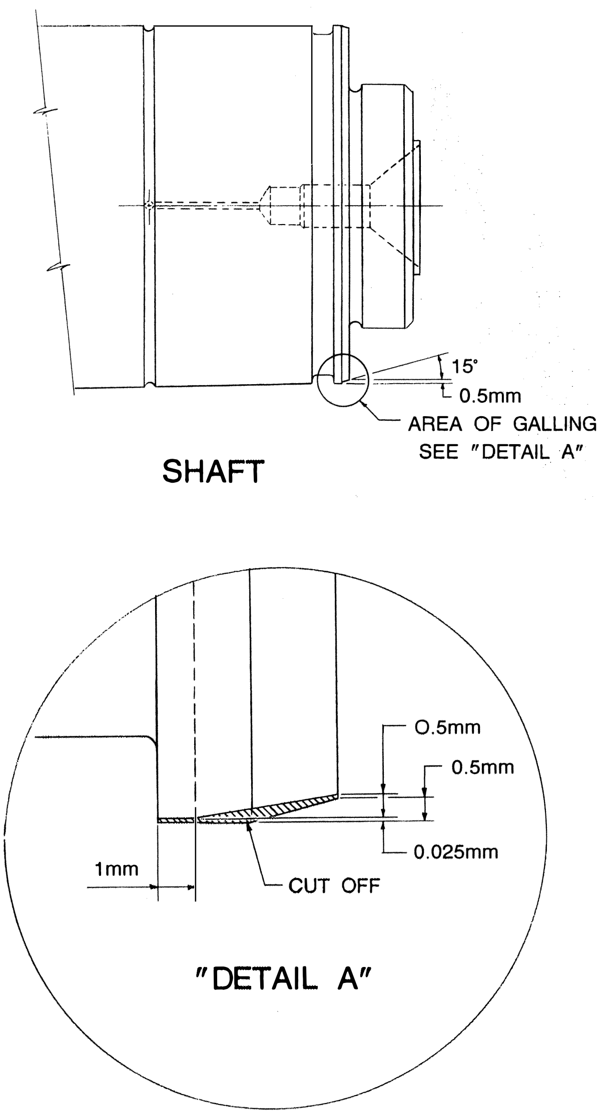


Figure 11. Shaft End Chamfer Modification.

than those on the compressor side. This rapid wear occurred even though the coupling teeth were nitrided to a Rockwell C-50. The couplings were replaced, but similar tooth damage was again discovered during the second shutdown in 1985.

Trying to understand the problem, a search began in the available literature for information on abnormal wear problems with couplings. The Conti-Barbaran number, an indication of the coupling floating member eccentricity, is said to be useful in predicting overall coupling performance [2].

$$e = 1E10N/(GD^2n^3)$$

e Conti-Barbaran Number
N Horsepower
G Weight of floating member, kg
D Pitch circle diameter, m
n Operating speed, rpm

For this particular coupling, the computation is 13. Trouble free performance was predicted for numbers greater than 10, so no problem here.

In evaluating potential heat generation within the coupling for severity of lubrication requirements, the expected tooth velocity was calculated to be approximately 4.0 ips [3].

$$v = r\omega\theta$$

v Relative tooth velocity, ips
r Pitch circle radius, in
ω Operating speed, radians/sec
θ Misalignment angle, radians

The upper limit for trouble free operation was 5.0 ips, which was not a very large margin of safety. This, combined with the heat coming from the turbine, could produce a potential problem. Carrying the analysis further, the expected temperature rise of the exiting oil was a moderate 38°F for the recommended 5.0 gpm supply, 2.5 gpm per mesh. However, if a blockage or misdirected flow reduced this to 1.0 gpm on the turbine side, the expected temperature rise was 98°F, and with 120°F inlet oil and the heat from the turbine, temperatures were quickly approached where the lubricating qualities of oil could begin to break down [4].

$$\Delta T = 0.184 \cdot PWR/(\gamma \cdot C \cdot GPM)$$

$$PWR = 8\mu\theta HP$$

ΔT Temperature rise of oil, F
HP Drive horsepower transmitted by coupling, hp
GPM Total oil flow to coupling, gpm
PWR Heat generated by coupling teeth of both meshes, hp
C Specific heat of oil, .555 BTU/lb*F
μ Coefficient of friction between coupling teeth
γ Weight density of oil, .0297 lb/in^3
θ Misalignment angle, radians

The problem was finally diagnosed as insufficient oil flow through the turbine coupling oil spray nozzles. The turbine manufacturer provided increased capacity spray nozzles for the turbine side of each coupling, and the problem has not recurred.

External Rabbet Failure

The turbine hub on the high pressure end was actually the outer portion of the turbine thrust disc. An unusually large coupling bolt circle had to be used on this end to avoid interference with the thrust disc. This led to a fretting problem between this flange and the mating coupling sleeve.

In August 1988, a coupling inspection revealed that the coupling on the thrust side of the turbine again had extreme wear and fretting damage on the turbine side teeth. This was after the oil flow had been increased to the turbine side of the coupling and had been operated long enough to believe the abnormal wear problem was corrected. The inspection also revealed that the external rabbet on the coupling sleeve had fretting damage and a portion had failed from fatigue. The approximately 1/8 in × 3/16 in × 8.0 in diameter external rabbet had completely sheared about 200 degrees around the circumference. Looking for the cause, no dimensional errors were found, but the bolts and bolt holes exhibited wear marks.

One possible explanation could have been loose coupling bolts. Loose bolts would have allowed the sleeve flange to move, causing the fretting damage and a side loading to be placed on the external rabbet. This could have resulted in the fatigue failure. Once the rabbet failed, oil could escape this area and leaked between the flanges, starving the coupling teeth of lubrication and causing the tooth fretting damage.

The damage evidence seemed to support the conclusion, but loose bolts were not reported when the coupling was disassembled. Just to be safe, extra care was taken to properly torque each of the 30 bolts on the turbine side of the coupling during reassembly. The coupling manufacturer's recommended installation torque is 50 percent of the bolt material yield strength. To make sure the bolts were tight, a change was made to the bolt manufacturer's recommended torque of 80 percent of the yield strength.

Another possible explanation for the rabbet failure could have been that the bolts were tight, and the coupling sleeve flexed under load. The large bolt circle (13.5 in) on this end of the turbine required an equally large coupling sleeve mating flange. With the large diameter of the flange and relatively thin cross section, distortion from misalignment forces could have been enough to cause relative motion at the locating rabbet. This could have created an interference loading, causing the external rabbet to fail.

According to Roark [5], the relative motion at this rabbet is proportional to α/t^3, where alpha is the angular rotation coefficient for the ratio of flange OD over ID, and t is the flange thickness. On the turbine end, α/t^3 is 2.2 *vs* 0.5 on the compressor side of the coupling. For any future couplings, if α/t^3 is much over 0.5, a close look will be taken at the flange design.

Broken Coupling Bolts

After increasing the installation torque on the coupling bolts in August 1988, the train ran for 12 months before the next shutdown. Inspection at that time revealed one of the coupling bolts had failed. The head of the bolt was missing, and dents in the turbine bearing housing indicated the head had broken off while the turbine was running. Material engineers examined the broken bolt shank and concluded the failure was caused by stress cracking from hydrogen embrittlement. But there was no process source for the required ionized hydrogen.

Further inspection revealed fretting damage on the mating coupling flange faces in the bolting area. The original machining marks and balance hole numbers of the turbine flange face were impressed on the mating sleeve face. The markings were in a scalloped pattern around the bolt holes as if influenced by the face pressure from the bolted connection. The wear and fretting on the sleeve face had become much worse since the coupling bolt installation torque was increased. Surprisingly, at the original lower installed torque value with expected greater relative motion there was very little fretting damage. The correspondingly lower contact pressure must have been the reason. The materials engineers believed the fretting mechanism was the source for the ionized hydrogen, causing hydrogen embrittlement of the coupling bolts. The higher installation bolt torque not only made the fretting worse, producing ionized hydrogen, but resulted in higher bolt stresses, further aggravating the situation.

Nonstandard Hub O-ring Size

The hub O-ring groove was incorrectly sized by the compressor manufacturer. The supplied standard metric O-ring was too small and the next size was too large. No standard metric or American O-ring fit this particular groove size. The OEM's proposed "solution" was to stretch the smaller O-ring to fit the larger bore groove. This was done on the initial installation because of the time constraints, but was unacceptable for a permanent solution. If the O-ring ever failed to seal, the whole rotor would have to be removed for coupling removal, because heat cannot be applied in the field.

Resizing the O-ring groove would have required additional reworking of the hubs and was not a desirable option at the time. Eventually, a domestic O-ring manufacturer was found who could custom make the correct size, an enormous quantity had to be purchased for them to take the "special" order.

UNIVERSAL LEARNINGS ON COUPLINGS

Face Friction is Not the Major Torque Carrier

Prior to investigating these bolting failures, many people believed most of the torque transmitted by the coupling was carried by face friction between mating flanges. It was learned from the vendor that a significant portion of the torque is transmitted by shear on the coupling bolts. One manufacturer stated that they design for 100 percent torque transmission by bolt shear.

Relying on bolting shear to carry the torque requires proper coupling bolt fit. Matching bolt hole diameter and concentricity must be maintained for uniform bolt loading. High strength bolts are not very ductile and cannot yield to accommodate much variation. If the hole and bolt diameters vary significantly, the load may not be evenly distributed and some bolts could fail. The coupling manufacturer normally assures bolt hole matchup, but with integral hubs, the responsibility falls on the user. In this case, the turbine manufacturer recommended reaming the bolt holes in the field. The coupling manufacturer drills the bolt holes slightly undersized so the holes can be reamed to size using the turbine hub as a template. Other alternatives are to finish the holes in the shop by using a jig for reaming, or using the precision of a numerically controlled machine.

Torque Coupling Bolts When Clean and Dry

The coupling manufacturer recommends not using any lubricant on the threads when torquing the coupling bolts during installation. The reason for this is that, depending on the lubricant, the friction range can vary greatly. Rather than get into specifying torque *vs* type of lubricant, the vendor recommends torque values with no lubrication for consistent bolt stretch. If threads are lubricated, the installation torque may need to be reduced by as much as 20 percent to prevent overstressing the bolt. Also, it is preferable to torque the nut and not the bolt head.

Mating Flanges Should Be Clean and Dry

Coupling mating flange faces should be cleaned of any lubricant during final assembly. This allows maximum torque transmission by friction between flange faces and reduces some of the load the bolts must carry.

CONCLUSIONS

Interim Solution
Hub/Shaft Galling Problem

Compressor hubs are no longer removed for a coupling change out. Coupling parts such as sleeves, spools, and bolts are now wet magnetic particle inspected for cracks every two years. The compressor hubs are wet magnetic particle inspected any time they are removed from the shafts, but they are removed only when absolutely necessary, because of the concern for shaft galling during installation.

If a coupling must be replaced, the hub is mismatched to the spare coupling pieces. However, the sleeves and spool with matched gear teeth remain a unit. This approach can be taken because:

- Mismatching coupling parts should not cause unbalance problems.
  - All coupling parts except the sleeves are individually balanced.
  - Sleeves are balanced during assembly balance.
  - The tolerance on the locating rabbet between the hub and sleeve is sufficiently tight to prevent unbalance problems.
- Mismatching coupling parts should not cause fit problems.
  - Hub and sleeve design permits dimensional interchangeability.

Broken Coupling Bolts

A return was made to the coupling manufacturer's bolt torque recommendations after the bolt failure. However, a switch from bolt material to Inconel 718 was made, because it resists hydrogen embrittlement. To obtain the required critical bolt fit, special grinding and machining are used to meet strict bolt diameter tolerance and finish. No readily available generic bolts were found to meet this need.

Broken Locating Rabbet

No way is known to correct this problem short of a major redesign of the existing coupling or changing to a different style coupling.

Final Solution:
Flex Element Style Coupling

Retrofit to a dry flex element coupling is being planned. A gear coupling with reduced moment hubs for the compressors (gear teeth on the hub) and a thicker sleeve flange for matchup with the turbine was considered, but rejected, because it is believed the flex element style will require less overall maintenance. The dry flex element coupling by the nature of its design will greatly reduce the misalignment forces suspected of causing the flange flexing problem, and this will not change over time, as it could with a gear coupling when teeth wear. The flex element coupling is able to use a much thinner flange than this gear coupling and has been successfully used in previous hydraulic installation applications.

The new flex element coupling should be installed later this year.

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