# ANALYSIS OF A HIGH SPEED COUPLING FAILURE

Investigation of why a heavy duty coupling failed while driving an ammonia compressor showed there were several contributing factors, not the least of which were misalignment and vibration.

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At a Canadian Industries, Ltd. fertilizer manufacturing plant, a flexible gear type coupling connecting a recovery turbine to an ammonia compressor recently failed rather suddenly. The coupling, which transmits 5,550 hp. at 10,530 rev./min., was thrown clear of the drive line with considerable impact but with no severe consequences.

Many of the parts that could be located were collected and a rather extensive stress and metallurgical analysis was undertaken to establish the mode and reason for failure and the modifications that should be made in the replacement coupling.

## Coupling description

The general coupling arrangement is shown in Figures 1-3. Flexibility of the coupling, as a result of the gear action, is such that axial thermal expansions can be absorbed as well as some misalignment. In the misaligned mode of operation the system is analogous to one where two Hooke joints with a splined shaft are employed. The flanges at either end of the coupling are attached by a bolt group forming a prestressed bolted connection.

Properly designed this coupling is an excellent system. However, misalignment at this speed of operation is quite critical, even though some can be tolerated by the coupling, since the unbalance may cause severe stresses. Further, it is difficult to realize the necessary flexibility in the coupling under high load conditions and marginal lubrication situations that exist here. These may result in an oscillating tensile stress on the bolts in the coupling flanges.

From the imspection of the failed coupling, it was determined that failure was initiated in the coupling bolts.

Microscopic examination made it obvious that many of the bolts failed due to a fatigue mechanism accentuated by significant pitting and fretting in the shoulder of the bolt where the coupling torsional load is carried. In some cases, a large number of parallel cracks formed several thin wafers in the failure process.

Microscopic examination also was made. The bolt material was extensively studied for abnormalities in structure and heat treatment. None were found, however, indicating that the fatigue failure was not attributable to heat treatment cracks or other such mechanisms.

## Stress analysis

The analysis considered here refers only to the bolt group design. Involved were:

1. *Static load consideration:* Assuming that the load and speed are steady at 5,550 hp. and 10,530 rev./min., repectively, and with

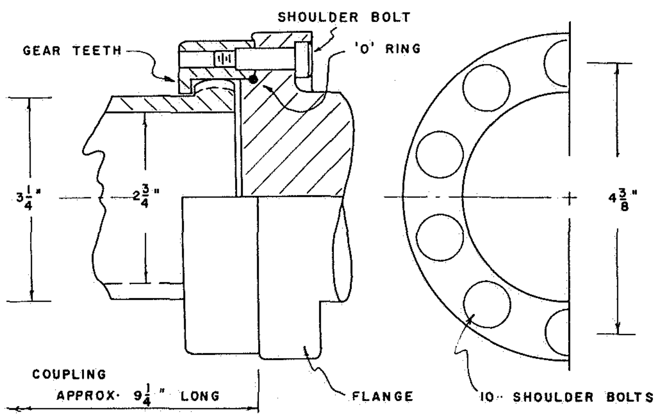



**Figure 1. Schematic of coupling geometry.**

the several reasonable assumptions brought out in the appendix, the resulting safety factor based on yield is about 2.3.

2. *Fatigue load considerations:* Here the analysis is more difficult since the precise load history of each bolt is unknown. However, it has been shown experimentally that mean shear stress has little effect on fatigue life, implying that the bolts must also be *subjected to an oscillating tensile load, a condition resulting from* misalignment and vibrations. The evidence indicated that this varying tension in the bolts was possible, indeed probable but to estimate the degree would be difficult to do reliably. Hence, several cases were proposed (see appendix for calculations).

These cases and their conclusions were:

1. Assume *perfect alignment,* no *vibration but the torsional* load fluctuating in a completely reversed condition.
   Conclusion: Safety factor = 1.4
2. Assume that misalignment and vibration causes a reversed tensile stress equal to one half of the transmitted torsional shear stress, and that these are both completely reversed and in phase.
   Conclusion: Safety factor = 1.3
3. Assume that misalignment and vibration allows the initial tightening preload on the bolts to be relaxed to zero once each revolution.
3. Conclusion: The system does not have infinite life and the expected life is less than 1 million cycles.
4. Assume similar conditions as in (3) except that the point of failure is considered at the shoulder of the bolt rather than at the fillet radius of the threaded section.
   Conclusion: Safety factor = 1.4

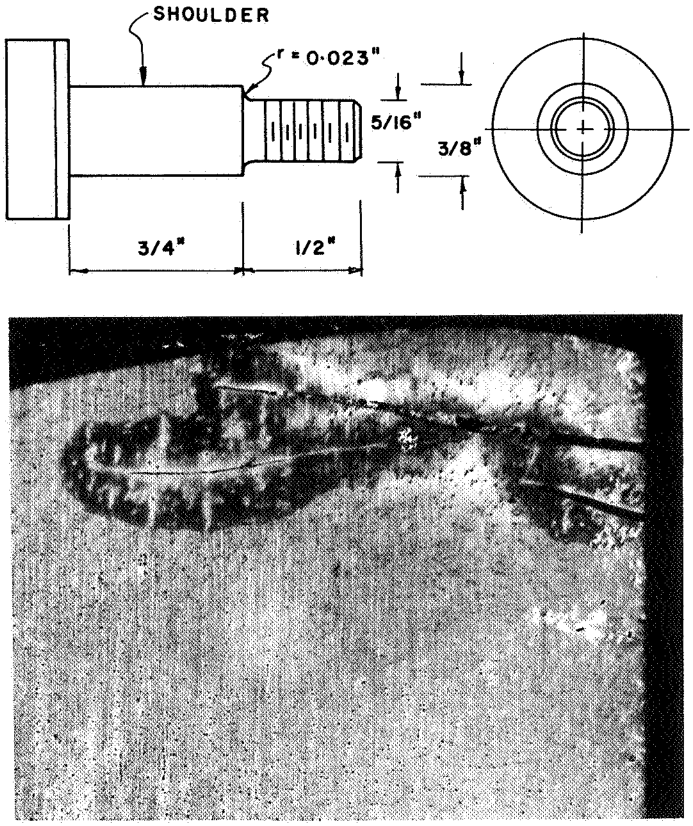



Figure 2. Shoulder bolt geometry and crack.

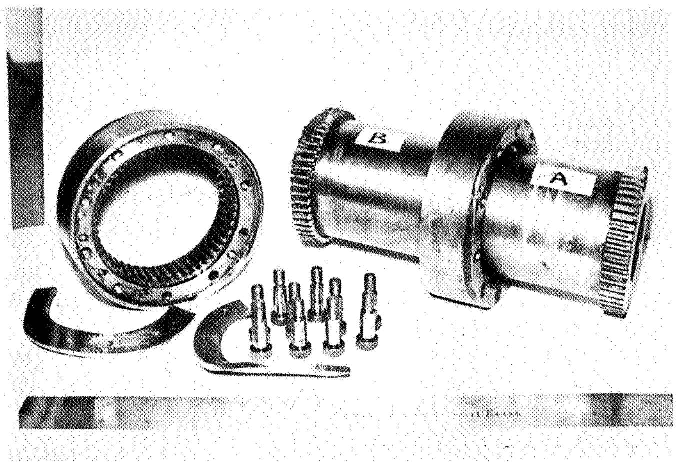



Figure 3. The coupling after failure.

## Energy load considerations

A bolted connection has tremendous advantage in a fatigue environment if it is properly designed and assembled. When no gasket is used in such a connection and the initial tightening load is as high as possible (eg. 90% of the proof stress) then the bolt will not see any external fluctuating tensile load until this external load exceeds the tightening load hence, no fatigue problems result. However, in the design of such a connection the bolt must be long such that a maximum stretch results since a certain amount of relaxation occurs during operation.

The fact that this type of flexible coupling does not allow significant misalignment under high load conditions was obvious in this case. The high contact stress in the teeth actually causes the teeth to weld and then pull apart, resulting in the formation of pits on the gear tooth profile. In addition, since relatively small motions are involved, lubrication to prevent this is difficult.

It was found then that there were no metallurgical reasons for the bolts to fail. However, if a good prestressed bolted connection was not made during assembly then there is good reason to believe that a fatigue failure would result. There was ample evidence that during assembly several unacceptable conditions resulted that provided the necessary conditions.

In particular, it was found that an "O" ring used to seal in the lubricant was oversized. It thus acted as a gasket in the prestressed connection, possibly the worst situation that could be imposed. Further, it makes it almost impossible to align the connection precisely. As well, the bolts were as much as 0.003 in. undersized, the holes were oversized, some bolts had no locking arrangement, all results of rather marginal tolerance and assembly inspection. There is no doubt that these factors allowed an oscillating tensile load on the bolts in conjunction with the shear stress resulting in a fatigue failure.

## Conclusion

From these investigations it thus was concluded that:

1. The mode of failure was fatigue on the shoulder bolts of the coupling.
2. The reason for failure was largely due to poor tolerance control and assembly procedures.
3. In the redesign, long bolts are recommended and good tolerance and assembly control are mandatory.

## Stress analysis calculations

Note that in all cases the critical section is assumed to be the shoulder of the bolt except for case B-4 where the critical section is assumed to be at the nominal diameter of the threaded section.

A. *Static load stress*

Bolt material - ASA B18.3

- $S_{ultimate} = 160{,}000$ lb./sq.in.
- $S_{yield} = 130{,}000$ lb./sq.in.

Assume 1. steady load 5,550 hp., 10,530 rev./min.
2. overload factor 1.5.
3. 75% of the bolts resist the torque.
4. neglect stress concentration and tightening stress.

The resulting shear stress in the bolts then is 28,000 lb./sq.in. and the safety factor using the shear stress theory of failure is then

$$\frac{130{,}000}{2 \times 28{,}000} \text{ or } 2.3$$

B. *Fatigue stress.*

*Case 1:* Assume: (a) perfect alignment and neglect tightening stress;
(b) a completely reversed oscillating torque.

Using Von Mises' fatigue theory to determine the maximum allowable alternating shear stress for this material gives

$S_{se}' = 0.577 \times \frac{1}{2} \times 160{,}000 = 46{,}000$ lb./sq.in.

The following relation considers the reliability, stress concentration and other factors such that the completely reversed shear stress endurance limit is $S_{se} = k_a k_b k_c k_d k_e k_f S_{se}'$ where:

$k_a = 1.0$ surface factor
$k_b = 1.0$ size factor
$k_c = 0.872$, 95% survival reliability
$k_d = 1.0$ temperature factor
$k_e = 0.95$ stress concentration and notch sensitivity effects
$k_f = 1.0$ miscellaneous factors
$\therefore S_{se} = 38{,}200$ lb./sq.in. and the resulting safety factor is

$$\frac{38{,}200}{28{,}000} \text{ or } 1.4$$

*Case 2:* Assume (a) a tensile stress of 14,000 lb./sq.in. is superimposed on the 28,000 lb./sq.in. shear stress in the bolts and that these are completely reversed and in phase.

Von Mises' combined stress fatigue criterion indicates that

$$\sqrt{\delta_{1a}^2 - \delta_{1a}\ \delta_{2a} + \delta_{2a}^2} \leq \text{Se for no fatigue problems}$$

here

$$\begin{matrix}\delta_{1a}\\ \delta_{1a}\end{matrix} = \frac{14{,}000 \pm \sqrt{(7000)^2+(28{,}000)^2}}{2} \text{ are the principal alternating stresses}$$

or

$\delta_{1a}$ = 35,000 lb./sq.in.

$\delta_{2a}$ = -21,900 lb./sq.in.

hence

$$\sqrt{\delta_{1a}^2 - \delta_{1a}\ \delta_{2a} + \delta_{2a}^2} = 50{,}000 \text{ lb./sq.in.}$$

again $Se = k_a k_b k_c k_d k_e k_f Se'$
where $Se' = \frac{1}{2}$ Su $= \frac{1}{2} \times 160{,}000 = 80{,}000$ lb./sq.in.
since the values of the k are similar to the first case
Se = 66,000 lb./sq.in.

The resulting safety factor then is $\frac{66{,}000}{50{,}000}$ or 1.3

Case 3: Assume (a) that misalignment and vibration relaxes the tightening stress to zero each revolution and consider the critical point to be at the fillet radius on the nominal diameter of the threaded section.

In this case, the section resists no shear due to the torsional load and only carries the initial bolt tightening tension. The proof stress for this material is of the order of 120,000 lb./sq.in., hence the maximum preload is approximately 108,000 psi at the stress area of the threaded section (5/16-18 UNC thread). Based on the nominal diameter of 5/16 in. this is equivalent to a stress of 74,000 lb./sq.in. Since this uniaxial tension varies from 0 to 74,000 lb./sq.in. the mean stress and alternating stresses are

$$\delta_m = \delta_a = 37{,}000 \text{ lb./sq.in.}$$

The maximum allowable completely reversed stress is again

$$S_e = k_a k_b k_c k_d k_e k_f s'_e$$

Here $k_e$ 0.57 which results from stress concentration due to a fillet radius of 0.023 in. with a notch sensitivity in this material of 0.8. The remainder of the k values employed are similar to those previously used, hence

$S_e$ = 40,000 lb./sq.in.

Applying these stresses to the following Goodman Fatigue Diagram, Figure 4, indicates that infinite life is not possible. Furthermore, when a representative stress-cycles fatigue diagram is constructed for this material the estimated life is less than one million cycles. Failure did in fact occur at this point.

Case 4: Assume (a) that misalignment and vibration relaxes the initial tightening stress to zero each revolution and consider the critical point to be the shoulder of the bolt transmitting steady torque.

A preload of 108,000 lb./sq.in. at the stress area of the threaded section is equivalent to a stress of 51,400 lb./sq.in. at the ⅜ in. diameter shoulder section. Further, the torsional stress of 28,000 lb./sq.in. also exists resulting in the stress conditions shown along with the corresponding principal stresses.

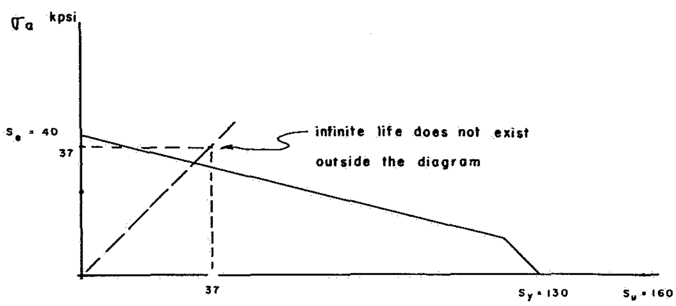
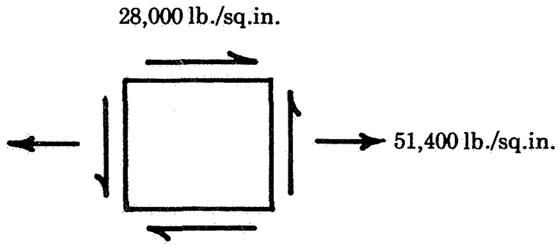
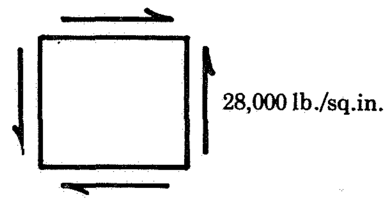



**Figure 4. Goodman Fatigue Diagram.**




$\delta_1$ = + 63,700 lb./sq.in.

$\delta_2$ = - 12,500 lb./sq.in.




$\delta_1$ = + 28,000 lb./sq.in.

$\delta_2$ = - 28,000 lb./sq.in.

The tensile values yield a tensile mean stress and alternating stress of

$\delta_{1m}$ = 45,900 lb./sq.in.

$\delta_{1a}$ = 17,900 lb./sq.in.

Similarly for the compressive stresses

$\delta_{2m}$ = - 20,300 lb./sq.in.

$\delta_{2a}$ = - 7,800 lb./sq.in.

Assuming Von Mises' biaxial fatigue theory applies then the Von Mises' mean stress and Von Mises' alternating stress is computed as shown.

$$\delta'_m = \sqrt{\delta_{1m}^2 - \delta_{1m}\ \delta_{2m} + \delta_{2m}^2} = 58{,}600 \text{ lb./sq.in.}$$

$$\delta'_a = \sqrt{\delta_{1a}^2 - \delta_{1a}\ \delta_{2a} + \delta_{2a}^2} = 22{,}800 \text{ lb./sq.in.}$$

Using the completely reversed endurance limit Se = 66,000 lb./sq.in. as determined in case 2 and applying the Von Mises' stresses to the Goodman Fatigue Diagram indicates a safety factor of 1.4. Since this point was considered to be the position of initial failure in the system, significant bending stresses must have been superimposed on these bolts during installation and operation as a result of tolerance and assembly procedure.

In the fatigue analysis the assumptions of case 3 and 4 i.e. that the shaft torque is constant but misalignment relaxes the initial bolt tightening stress to zero each cycle is probably more realistic than those assumptions of case 1 and 2. In all cases, however, several effects were neglected or minimized. For example, no consideration was made for possible fretting of the bolt due to the bearing stress and the resultant effect on surface factors. As well, the 95% survival rate is relatively modest reliability.

## Literature cited


1. Mechanical Engineering Design, J.E. Shigley, McGraw-Hill, 1963.

**Q.** I missed one point. The calculations showed that there was a fatigue life of about 1 million cycles. What was the thing that caused a million cycles over this two-week period?

**NORTH:** The two-week period of operation is a good deal more than 1 million cycles. You attempt to make an estimate of the life you expect. In this case an oscillating tensile stress was developed.

**Q.** What do you feel was the cause of this stress?

**NORTH:** Primarily misalignment and vibrations.

**Q.** Was there a guard around the coupling?

**J.W. MARTIN,** Canadian Industries, Ltd.: A guard fabricated from light gauge steel was installed at the coupling. When the failure occurred the centre spool piece of the coupling was thrown through the guard.

A heavy plate guard was installed following the incident that should retain the coupling spool piece if a failure occurs.

The compressor shafts adjacent to the coupling have integral hubs. When the failure occurred the coupling spool piece was momentarily thrown into a cocked position between the hubs. One of the hubs was damaged to the extent that it could not be re-used.

**Q.** It was noted that the effect of the nylon locking arrangements on the nuts would have to be considered. One thing I've wondered about is that in recent years there have been developed a number of lubricants for these nuts, particularly the kind using molybdenum disulfide. These markedly change the frictional characteristics between metals and I'm wondering to what extent the use of those should be considered in places such as this and how compensation can be taken for the changes in friction when setting up standards for torque?

**NORTH:** This is a difficult problem. The manufacturer will make a specific recommendation for the torque to be applied when tightening the part. Not all manufacturers indicate what additional torque should be used when using a nylon locking system, for example. It may be that you'd have to determine this experimentally.

My own suggestion is that you take it into your shop and tighten the bolts until they fail. And then use at least 75% of that torque in the installation, if you can possibly get it. Unfortunately a lot of these use Allen wrenches and such to tighten up and it is very difficult to develop the correct tightening torque with a system like that.

**E.L. KEMMLER,** Hartford Steam Boiler Inspection and Insurance Co.: We've run into a lot of cases over the years where the practice of tightening flange bolts and coupling bolts by torquing is inadequate. There are entirely too many variables, such as the difference in thread friction, the amount of lubrication used, locking devices, and the ability of the man who is using the torque wrench to read it. The turbine manufacturers learned a long time ago that the best possible way to achieve the desired amount of torque is to stretch the bolt by calculation a certain elongation. If you stretch the bolt a pre-determined amount, you get the torque you want. If the coupling manufacturers and compressor manufacturers would adopt that technique you might have a lot less bolt failures.

**NORTH:** That is a well accepted technique and there are some cases where bolt elongation works well. Where you have very long bolts, measuring the elongation works well. If you have a very short bolt, it doesn't work nearly so well.

**W. JENKINS,** Sinclair Petrochemicals Co.: Some of us had the old cryogenic plants where we had serious leakage problems. The new cryogenic plants have solved the leakage problem by welding at all points. Bolted surfaces inside a cryogenic plant were a very severe subject at these meetings for a number of years. There is only one way to stop leaks at bolted surfaces in a cryogenics plant and that is by measuring elongation of the bolts. We have torqued all kinds of bolts of various lengths and various sizes and we have found that by measuring the actual elongation there is not a direct relation between the elongation and the foot pounds of torque. The only correct way to prevent leakage of flanges is to measure the elongation of the bolts. We've done it on bolts 1¼ in. long and on those 6 in. long and have had very good results.