METAL DIAPHRAGM COUPLING PERFORMANCE

by

M. M. Calistrat

Manager, Power Transmission Development Section

Research & Development Department

Metal Products Division

Koppers Company, Inc.

Baltimore, Maryland

Michael M. Calistrat, a native of Rumania, graduated from the University of Bucharest with an M.S. degree in Mechanical Engineering in 1951.

He worked for many years designing components for oil drilling rigs, including rotary tables, hydrostatic transmissions and sludge pumps.

In 1964 Mr. Calistrat immigrated to the USA and worked for Crane Carrier Co. in Tulsa, Oklahoma, designing truck frames, some for loads exceeding 100 tons. Late in 1966, he moved to Baltimore, Maryland as a senior project engineer for the Metal Products Division of Koppers Company, Inc., in the Research and Development Department, and after one year, was promoted to his present position as manager of the Power Transmission Development Section. Since then, he has worked mainly on flexible couplings, helping in improving the existing product line and in adding new products. He holds twelve U.S. patents.

Since 1975, Mr. Calistrat has been chairman of the Shaft Couplings and Clutches Subcommittee of ASME. He has presented various papers at ASME and ASLE meetings and also authored articles in Hydrocarbon Processing and Mechanical Engineering.

ABSTRACT

The performance of a Metal Diaphragm Coupling is to a large extent determined by its endurance limit. The endurance limit can be estimated by calculations but must be confirmed through testing. The author cites data obtained from a comprehensive test program conducted in the Research and Development facilities of Koppers Company, Inc. in connection with the development of an improved version of the Single Diaphragm Coupling.

Failure of the coupling's disk, or diaphragm, occurs through fatigue when it is stressed above its endurance limit; the mode of failure depends on the angular misalignment and axial displacement. Full scale tests conducted in the laboratory under a variety of extreme conditions confirmed the operating limits and provided a pictorial reference which may be used as a tool to identify probable causes of disk failures.

A special test rig was used to obtain S-N (stress versus number of cycles) curves for diaphragms made of various materials, for various treatments, surface finishes, etc. The large number of tests performed gave a good insight into the influence of various factors on the endurance limit and on the consistency of the endurance limit from disk to disk.

INTRODUCTION

The Metal Diaphragm Coupling is relatively new in turbo-machinery applications. Although the first recorded use of such a coupling dates back to 1922*, the contoured diaphragm did not become widely used until the late 1950's.

Diaphragm couplings accommodate the system misalignment through flexing. Fatigue resistance is the main performance criterion. The life expectancy of a diaphragm coupling *that operates within its design limit* is theoretically infinite. Considering that it is a relatively new product, and considering that machinery users are not accustomed to having products that are designed to last forever, Koppers conducted comprehensive laboratory tests at conditions which were much more demanding than normal field service. The results of these tests are very encouraging.

Figure No. 1 shows a section through a diaphragm coupling. The coupling has only five parts: two rigid hubs, one spool piece, and two alignment rings. These five parts are solidly bolted together and misalignment is accommodated through flexing of the two disks of the spool. The spool piece is made up of three separate parts: two disks and a spacer tube. These parts are welded together through the electron beam (EB) process. EB welding, a relatively new industrial process, is one of the keys to the successful manufacture of diaphragm couplings.

The heart of these couplings is the flexing disk; it is manufactured from vacuum degassed alloy steel, forged with radial

*JUNGSTROM Condensing Steam Turbine Driven Locomotive, Sweden.

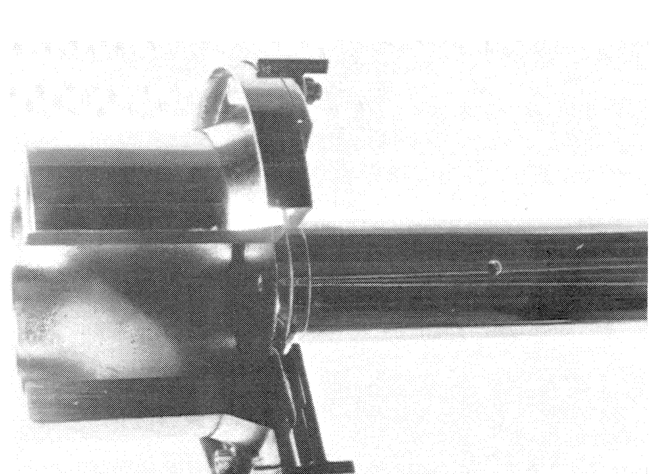

Figure 1. Metal Diaphragm Coupling (one end shown).

grain orientation and has a contoured profile machined on high precision equipment.

WHY CONTOURED PROFILE?

In order to obtain the minimum of force for a given deflection (maximum flexibility) a beam must be uniformly stressed throughout. Under normal operating conditions a diaphragm of a coupling is subjected to uniform and cyclic stresses. The uniform stresses are generated by torque, centrifugal forces and axial deflection. The cyclic stresses are induced by the angular misalignment seen by the disk. A method for calculating these stresses was published for the first time in 1948 by Wolff (1), and was updated by the manufacturers of diaphragm couplings. The following simple analysis of the stresses in a disk is merely intended to demonstrate the need for a contoured profile.

a. Torque transmission (See Figure No. 2)

The tangential force at r_i is T/r_i (1)

Where T is the torque

The section area at r_i is $2\pi r_i t_i$ (2)

The shear stress is $\tau = T/2\pi r_i^2 t_i$ (3)

For constant shear stress

$$r_i^2 \times t_i = \text{Constant or}$$

$$t_i = K_1/r_i^2 \quad (4)$$

b. Centrifugal forces

Due to centrifugal action, a section through the disk is also subjected to a radial force C. Because the disk is solidly bolted to a rigid hub, the calculations of the centrifugal force acting on the disk are rather involved. Standard methods to calculate centrifugal stresses in a rotating disk (2) show that both the tangential and radial stresses increase rapidly with a decrease in the radius. For a uniformly stressed disk we can again write

$$t'_i = K_2/r_i^2 \quad (5)$$

c. Axial deflection (See Figure No. 3)

This type of disk deformation is called "umbrella" by Wolff, but is more appropriately known as "drum head" because the rim diameter remains constant. It can be seen that the disk deformation has two effects: radial stretch (the distance between rim and hub increases) and bending. The stresses imposed by axial deflection (See Figure No. 4) are larger at the hub than at the rim. As will be shown later, these large stresses significantly influence the failure mode of the disks.

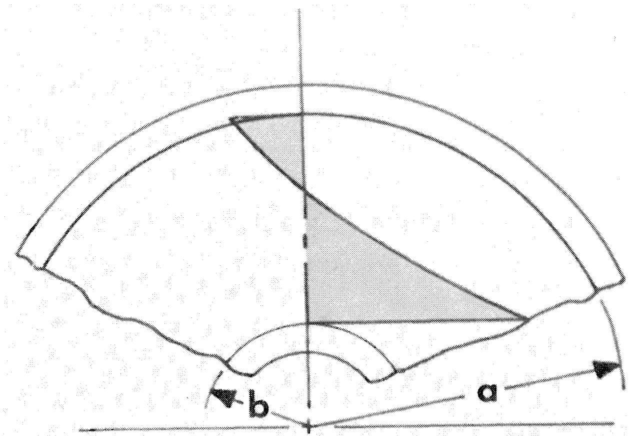

Figure 4. Stress Distribution Under Axial Deflection.

d. Angular misalignment

Figure No. 5, reproduced from Wolff's paper, illustrates how the stresses generated by angular misalignment are influenced by the ratio of outer diameter/inner diameter; the curves are drawn for a disk with uniform thickness and for a profiled disk. As the ratio b/a (See Figure No. 4) for most of the diaphragm couplings is approximately .45, it can be seen that the advantage of profiled disks, *for this type of stress,* is small.

In conclusion, in order to have a uniform stress when all the various forces acting on the disk are at their maximum rated values the disk must have a profiled contour. If this stress exceeds the endurance limit of the material the failure can occur anywhere between the hub and rim area. This statement is no longer true if only one or more of the forces are at their maximum values, as will be shown later by the test results. Failures of early disks occurred mainly in the thinnest section. Even Wolff states in his paper: "the disk . . . should be progressively thickened over the outermost 15 per cent of the annulus to give a rim thickness some 25 per cent greater than the theoretical."

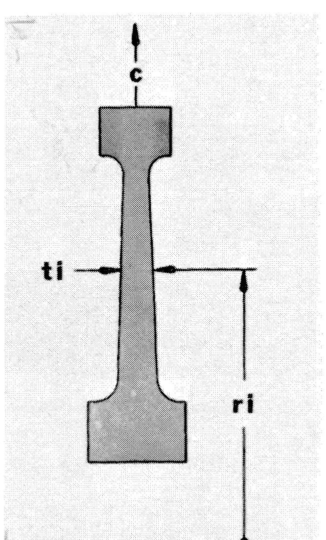

Figure 2. Section Through a Profiled Disk.

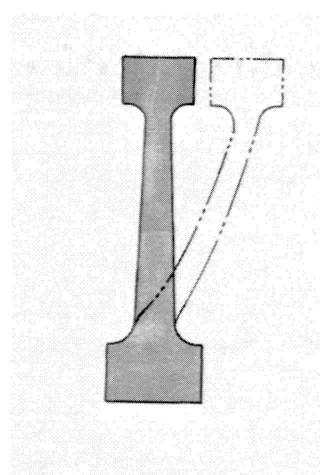

Figure 3. Axial Deflection in a Disk.

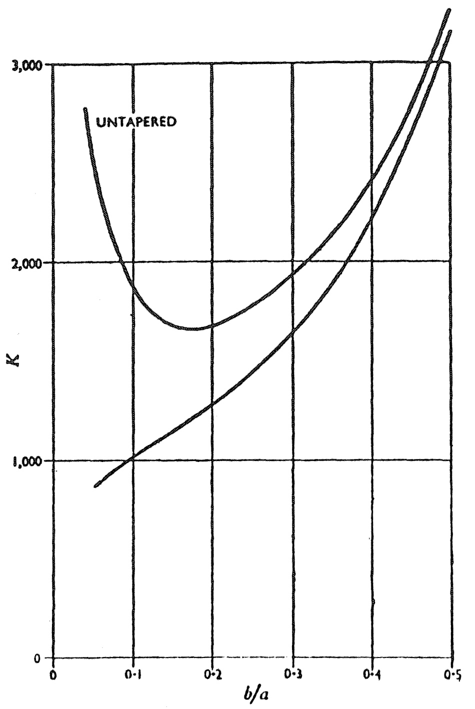

Figure 5. Stress Under Angular Misalignment.

What was not recognized originally was that progressive thickening was not necessary for reducing stresses, rather it is required for reduction of stress *concentr tions*. Both at the hub and at the rim, fillet radii are used to connect the contoured profile to the thick annuli required for making the connections to the other components of the coupling. At the hub, the contoured profile and the fillet radius have the same sign, i.e., both tend to thicken the disk with a decrease in r_i At the rim, however, the contoured profile and the fillet radius have opposite signs, and this fact causes an unacceptable stress *concentration*.

Methods used to obtain the progressive thickening vary somewhat from manufacturer to manufacturer. Optimum profile is obtained through experimentation rather than through theoretical analysis.

EXPERIMENTAL PROGRAM

The experimental program we conducted had two phases: first, S-N (stress versus number of cycles) curves were determined for various materials, heat treatments, manufacturing methods, etc.; second, full scale tests of entire couplings were conducted at gradually increasing loading conditions until failure occurred.

a. S-N tests

Establishing S-N curves with entire couplings is impractical for several reasons: considering the large number of test points required to determine an S-N curve the cost of the test parts would be prohibitive; the energy transmitted through the couplings would be wasted; and the overall variations between the many test parts used for one curve would cause a scatter of the results. It was therefore decided that any single S-N curve would be established using test samples machined from one disk. The disk was cut radially into 10° segments, and the sides were ground in a special jig to insure uniformity from sample to sample.* A special test rig was built in order to simulate, as closely as possible, the flexure of a disk rotating under conditions of angular misalignment. The test rig is illustrated in Figure No. 6.

Although this test rig did not reproduce any of the constant stresses in the disk, the results of the tests helped us in determining the endurance limit, and enabled us to select the best variables for the coupling.

The following graphs illustrate how the endurance limit is influenced by some of the factors studied.

Figure No. 7 represents the results obtained with a disk having all the parameters chosen for production couplings: material was modified 4335, vacuum degassed, radially forged; the disk's surface finish was very fine, and proper heat treatment and shot-peening were applied. Not only is the endurance limit very high (75,000 psi) but the scatter of the test results is minimal. Figure Nos. 8 and 9 represent the results of tests with similar disks having surface finishes 4 and 10 times coarser than the one selected for production. It can be seen that, as the surface quality worsens, the scatter of the test results becomes larger, and beyond a certain point an endurance limit cannot be established.

Although shot-peening generally improves the endurance limit of a component subjected to cyclic stresses, it was found that, for parts as thin as these profiled disks, shot-peening can sometimes be detrimental. Figure No. 10 illustrates this finding. Poor results can be caused either by shot that is too large or by impact velocity that is too high.

*Note that slight variations in the width of the test samples do not affect the stresses.

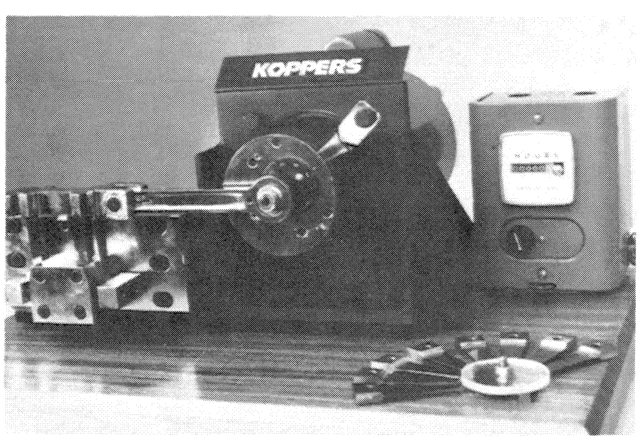

Figure 6. Fatigue Test Rig.

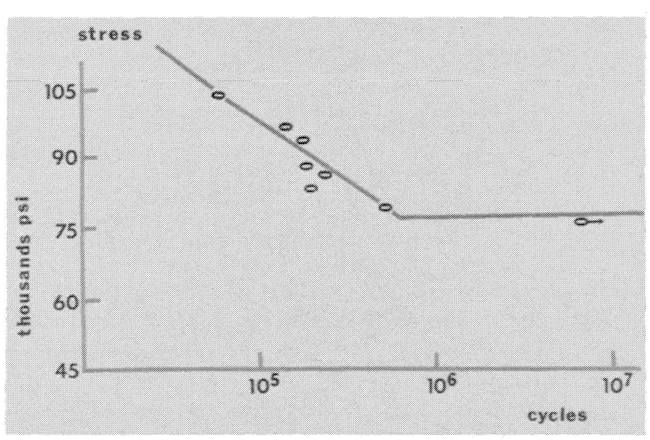

Figure 7. S-N Curve for Production Disk.

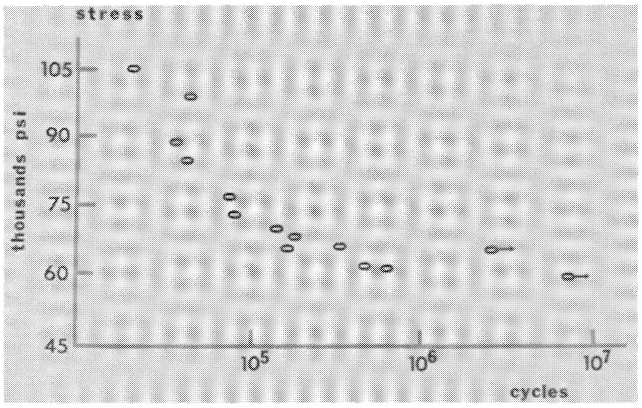

Figure 8. S-N Curve for Disk with Medium Grade Surface Finish.

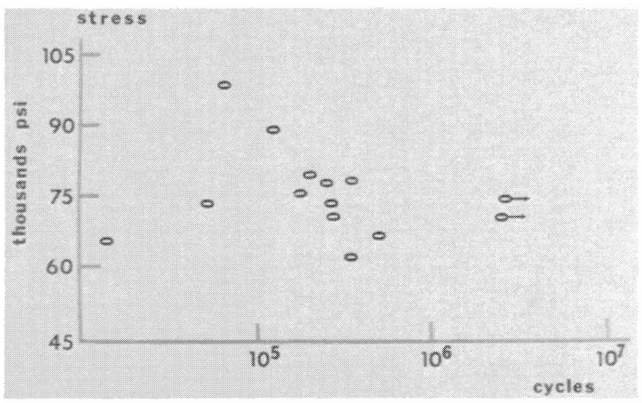

Figure 9. S-N Curve for Disk with Coarse Surface Finish.

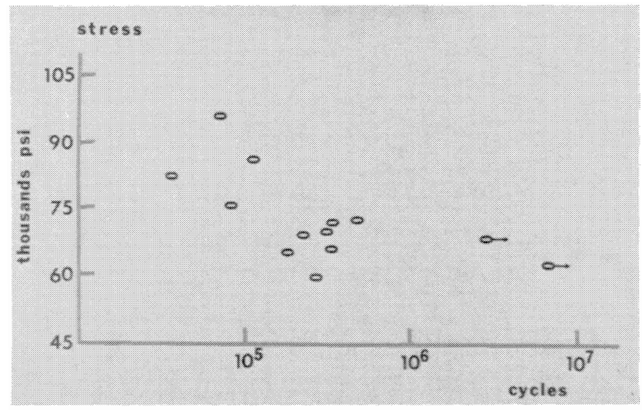

Figure 10. S-N Curve for Disk with Improper Shot-Peening.

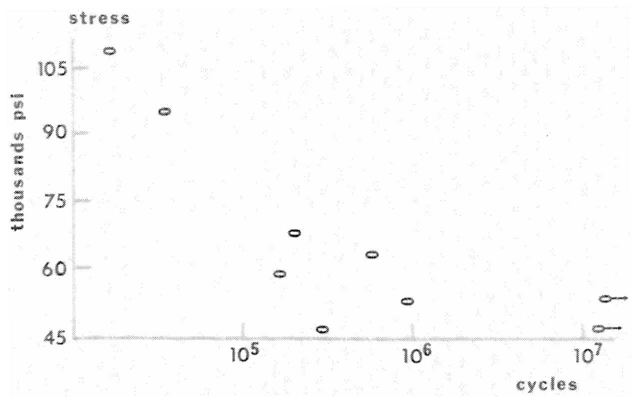

Figure 11. S-N Curve for Disk Manufactured Through ECM.

Rather surprising results were obtained with disks manufactured by "electro-chemical-milling" (ECM) process and with those which were "electro polished." Figure No. 11 represents the S-N graph of such a disk. Although the ECM process yields parts which are geometrically perfect, it was found under electron scanning microscope observation that small cavities sometimes develop on the surface of the metal . . . cavities which apparently generate stress concentration.

The results shown here are only a small part of the data obtained in our laboratory fatigue tests. They help us to understand some of the problems which prevented successful production of this type of coupling until sometime in the 1950's.

b. Full scale testing

Through careful stress calculation, and by knowing the endurance limit of the material selected, a designer can calculate the conditions under which fatigue failures of contoured diaphragms will occur. By applying a reasonable safety factor, he can then establish ratings.

But verification of the work, experimental and theoretical, can be done only through testing the couplings to failure.

Testing in a laboratory, as compared with field experience, has two major advantages: first, it can be done under well controlled conditions, and second, the testing can be stopped shortly after failure occurs.

Testing in a laboratory also has disadvantages; it wastes energy, and the size of couplings that can be tested is limited. To offset these disadvantages, testing of contoured diaphragm couplings was performed in a "four-square" rig, described in the Appendix.

In order to stop the test rig shortly after the failures occurred, the rig was equipped with sensors that measured torque, speed, and vibrations in the gear boxes, and with proximity gauges placed in the vicinity of the coupling tubes. Other sensors measured temperatures at various points, oil pressures, etc. In case of a malfunction, the rig's annunciator automatically stopped the rig and indicated which of the sensors triggered the test interruption. This instrumentation was successful in stopping the rig before major damage was done to the failed disks, and thus enabled us to analyze the failures and learn much about the failure modes.

When a coupling is used below its ratings, one expects the coupling: to transmit the required torque, to withstand the operating speed, and to operate for a reasonable length of time For a machine element subjected to fatigue, the third condition, time, is usually the most demanding. Considering that a disk operating at high rotating speed accumulates an impressive number of cycles every day, the only acceptable "reasonable length of time" is infinity. Fortunately, it is known (3) that, for steels, infinite life is obtained if the parts can successfully operate for more than ten million cycles. Our tests were conducted until failure, or until a minimum of 10^7 cycles were accumulated. All the failures occurred before a disk could accumulate more than four million cycles.

c. Test Procedure

To obtain meaningful results, the tests had to be conducted at various levels of constant and cyclic stresses. Any of the three constant stresses (torque, centrifugal force and axial displacement) could have been varied. Both theoretical analysis and experimental work showed that the axial displacement imposes the largest stress of the three. Also, in field service, it is more likely that a diaphragm coupling can be overstressed through improper axial spacing than through overtorquing or overspeeding. It was hence decided that various levels of *constant* stresses would be applied by gradually increasing the shaft spacing while the torque and speed were kept constant.

A graph was established, having as coordinates the axial displacement and the angular misalignment per disk, as shown in Figure No. 12. It was known from stress analysis that a curve exists on this graph . . . a curve which separates the infinite life zone from the failure zone.

The object of our tests was to experimentally determine this curve. Tests started zero axial displacement and a safe amount of angular misalignment. After 10^7 cycles the misalignment was slightly increased and another 10^7 cycles were accumulated. The process was repeated until failure of a disk occurred. The failure defined the point "A" on Figure No. 12. The failed coupling was replaced with a special gear coupling, and testing was continued until the second diaphragm coupling in the "four-square" rig failed. The second failure gave us the point "A" for the second coupling size.

New diaphragm couplings were then installed in the test rig. A certain amount of axial displacement was introduced at each coupling, and testing was resumed at a safe amount of angular misalignment. The procedure described above was repeated and points "B" were obtained for the two coupling sizes. After obtaining a sufficient number of points to define the failure curve many "repeat" tests were performed to verify the consistency that could be obtained with this type of coupling.

d. Test Results

It was found that: a. As expected, the failure curve is similar in nature to a typical Goodman diagram; and b. The repeatability of failure points was excellent, in fact, many of the tests resulted in the failure of both disks of a single coupling.

Figure No. 13 gives a typical* example of a failure curve determined experimentally, and compares it with the coupling's rating. Very similar results were obtained with other size couplings.

RATING METHOD

Any product must incorporate a safety factor. A good safety factor is a compromise between safe operation and a reasonable cost. Once a safety factor is selected, the ratings of a contoured diaphragm coupling can be established by tracing a curve roughly parallel to the failure curve at a distance representing the safety factor (See Figure No. 14).

Such a rating has two disadvantages: a. It can cause serious misapplications at both extremes, and b. It is relatively awkward to use. Let us analyze these disadvantages: a. Short of laboratory conditions, installations having either no axial displacement or no angular misalignment are highly improbable. To safeguard both the manufacturer and the user, it is better to limit the maximum axial displacement and the angular misalignment as shown in Figure No. 15.

b. Such a rating curve is awkward to use because it requires a graph. A simpler method is to specify a maximum axial displacement and a maximum angular misalignment, as shown in Figure No. 16. The drawback of that rating method is that the safety factor varies widely, and in extreme conditions can be smaller than the one originally selected.

*Note: The coupling with the smallest safety factor was chosen.

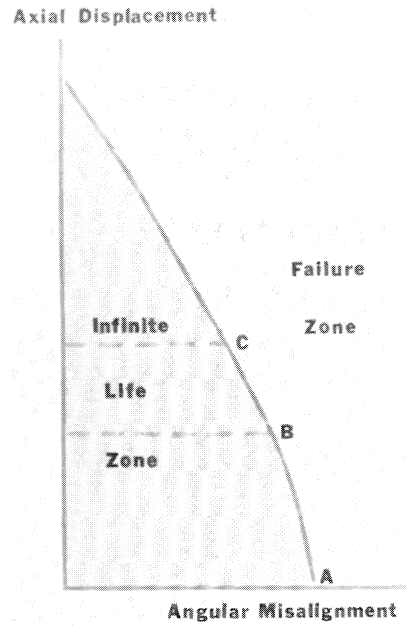

Figure 12. Typical Failure Diagram.

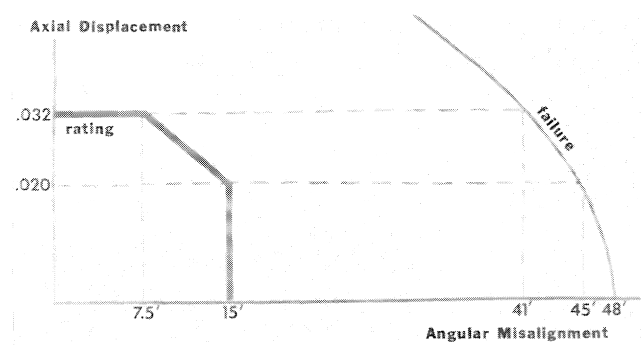

Figure 13. Failure Curve of a Koppers Size 1MDH Coupling Operating at 1,800 HP and 14,000 RPM.

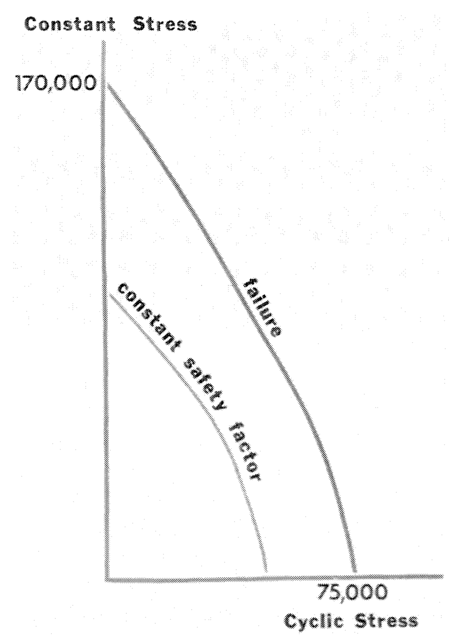

Figure 14. Constant Safety Factor Rating Method.

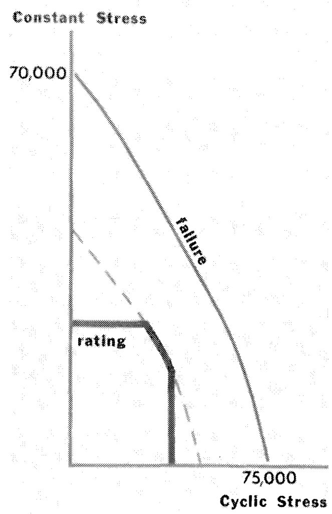

Figure 15. Simplified Rating Method A.

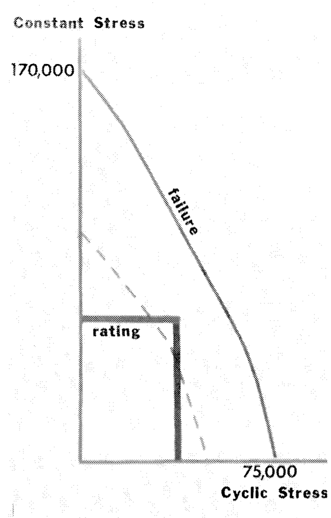

Figure 16. Simplified Rating Method B.

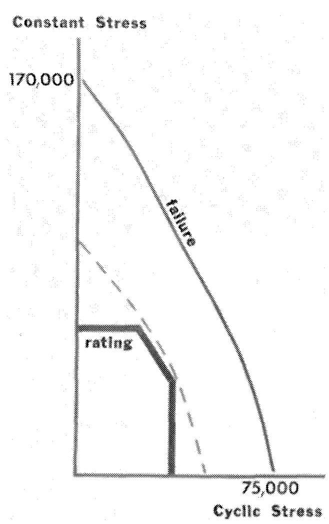

Figure 17. Simplified Rating Method C.

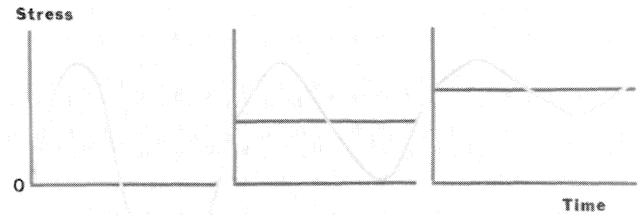

Figure 18. Stress Variation Under Three Conditions.

Figure 19. Failure Under Zero Axial Displacement Condition.

Another method, which is a compromise of the previous two, is to replace the constant safety factor portion of the rating with a straight line, as shown in Figure No. 17.

FAILURE MODES

Figure No. 18 represents the stress variations in three hypothetical cases.

While in the case of reversing stresses any particle of a disk is subjected alternately to compression and tension, in the case of high constant stresses any particle is subjected only to various degrees of tension. It is to be expected that the failure mode of a disk will be influenced by the amount of stress reversal. Two distinct modes of failure were found, one at zero axial displacement and the other at large axial displacement.

Figure No. 19 illustrates the failure of a disk that had zero axial displacement. Three aspects characterize this type of failure: a. The crack line is circular and goes through the thinnest portion of the disk; b. The crack is relatively smooth; and c. There is no, or very little, buckling in the disk.

Figure Nos. 20 and 21 illustrate the failure of a disk that had a large amount of axial displacement. The three aspects that characterize this type of failure are the opposite of the previous ones: a. The crack line has a random path going from the thinnest to the thickest portions of the disk; b. The crack line is very irregular; and c. There is severe buckling of the unfailed part of the disk.

The transition of these characteristics from zero axial displacement to large axial displacement is gradual only for aspects a. and c; a smooth crack line is obtained only when the axial displacement is zero.

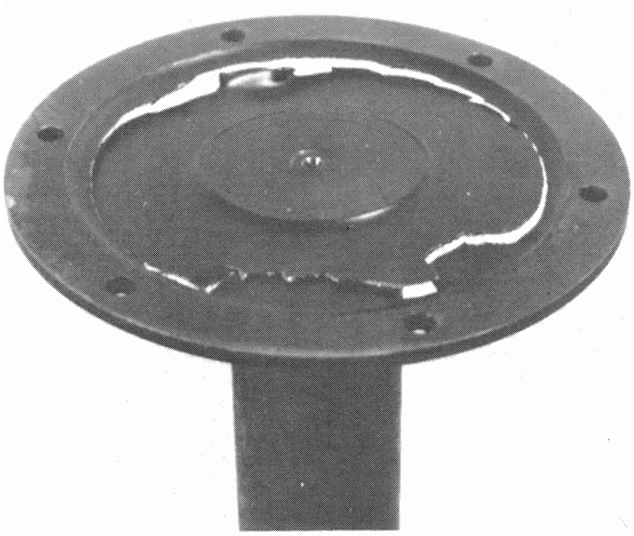

Figure 20. Failure Under Large Axial Displacement and Angular Misalignment Condition.

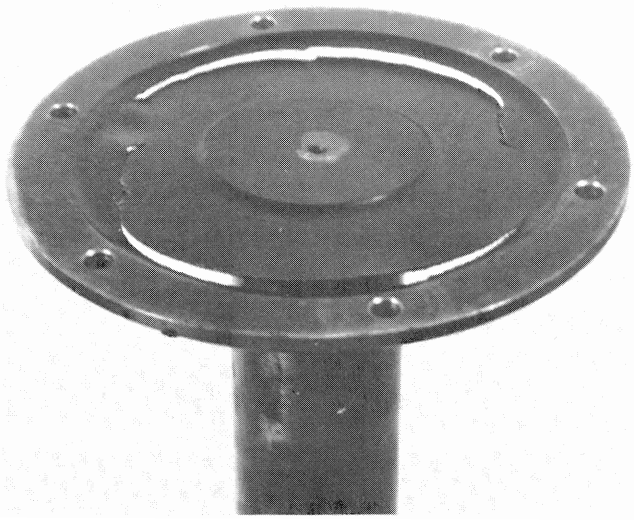

Figure 21. Double Failure Under Large Axial Displacement and Angular Misalignment Condition.

Another aspect of the failure mode is significant. As shown in Figure No. 22, the crack line propagates more than 270° before buckling of the disk takes place. As it was anticipated theoretically, this condition indicates that the torque load makes only a small contribution to the total stress in the disk.

CONCLUSIONS

Metal Diaphragm Couplings of the type tested are highly reliable pieces of machinery when operated within their rated conditions. Angular misalignments had to be increased at least three times beyond ratings before failure could be induced.

In its ability to transmit torque the coupling is even safer . . . a disk can transmit the rated torque even after more than half of it is broken.

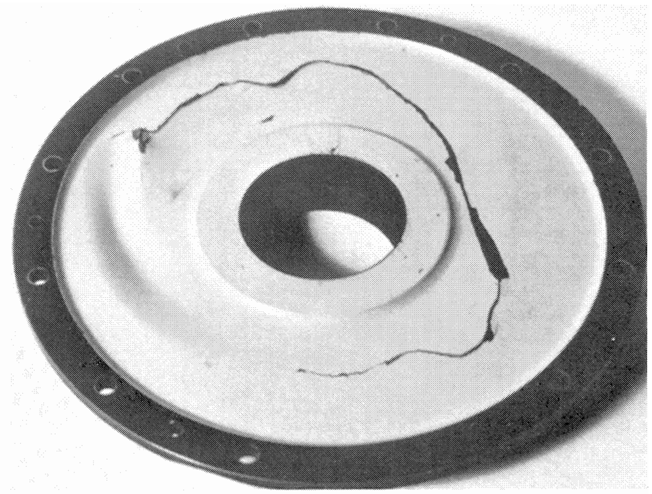

Figure 22. Failure and Buckling Under Large Axial Displacement and Angular Misalignment Condition.

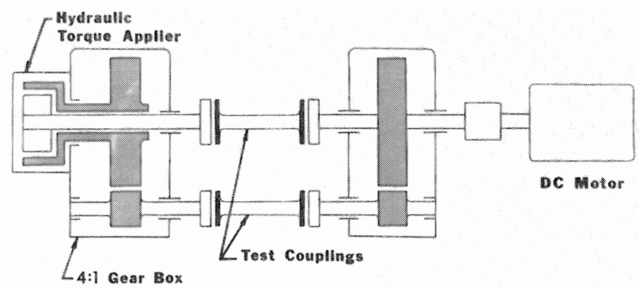

Figure 23. "Four-square" Test Rig.

By observing a failed disk one can easily identify the cause of failure, whether there was excessive angular misalignment, axial displacement or torque.

APPENDIX A

The "Four-square" Test Rig

This type of test rig has the advantage that the energy is recirculated within the rig, hence only the energy lost in friction must be supplied externally. Schematically, the rig is shown in Figure No. 23.

The test rig on which the metal diaphragm couplings were tested can develop 4,000 HP, and the maximum speed is 20,000 RPM. The heat generated through friction is transferred by the lubricating oil to a water circuit.

REFERENCES

1. Wolff, P. H. W., "The Design of Flexible Disk Misalignment Couplings"; Proceedings of Institute of Mechanical Engineering; London, 1948 (revised 1950).
2. Timoshenko, S., "Strength of Materials," Part II, page 205; Van Nostrand Co., 1958.
3. Battelle Memorial Institute, "Prevention of the Failure of Metals under Repeated Stress," page 108; Wiley, 1941.