

MAINTENANCE TECHNIQUES FOR TURBOMACHINERY

by

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ABSTRACT

This paper covers turbomachinery repair techniques that have evolved from experiences in a large gasoline refinery. Engineering design problems and upgrading of all types of compressors, drivers, and accessories are discussed. Particular attention is given to barrel-type compressors and shop repairs.

INTRODUCTION

Maintenance of turbomachinery can best be described as a series of upgrading projects to restore the machinery to a reliable, efficient operating condition. Restoration to a like new condition is not acceptable. Frequently, each repair must be to a better than new standard. The heart of good maintenance is developing these improvements through the efforts of knowledgeable and dedicated engineers, supervisors, and craftsmen.

Amoco's Texas City refinery has a refining capacity of over 400,000 barrels per day of crude with emphasis on processes that increase the gasoline yield of that crude. In addition, the refinery has a large ammonia-producing complex and generates much of its own power.

There are 59 centrifugal compressor cases in our refinery. Drivers include 26 steam turbines, 4 gas turbine-steam turbine combinations, 11 electric motors, and 15 gear boxes, giving a total of 115 items that fall under the term of turbomachinery as used in this symposium. This paper will discuss some of the techniques and the improvements used by Amoco to maintain this equipment. Maintenance of reciprocating compressors, large pumps, and turbine generators will not be covered.

BARREL-TYPE COMPRESSORS

Barrel-type compressors are being utilized in the process industry to an increased extent because the barrel design con-

finer gases more effectively than horizontally split cases. This becomes a critical consideration in two areas — high pressure and low molecular weight gas compression. API-617, "Centrifugal Compressors for General Refinery Services," requires a barrel design based on the molecular percent of hydrogen contained in the process gas and the discharge pressure. Figure 1 details those requirements. The barrel design is essentially a compressor placed inside a pressure vessel. For higher pressures some manufacturers have merely "beefed up" lower pressure barrel designs, while others have perfected unique designs such as the "shear ring" head design. All of these designs make extensive use of elastomer O-rings as sealing devices.

There are several inherent maintenance problems with barrel-type compressors as experiences with our eighteen barrels (30% of the total) have illustrated. Special techniques are necessary to cope with these problems.

1. *Handling* — Barrel-type machines must be removed from their foundations for total maintenance. Since we have barrel machines weighing up to 30 tons, the han-

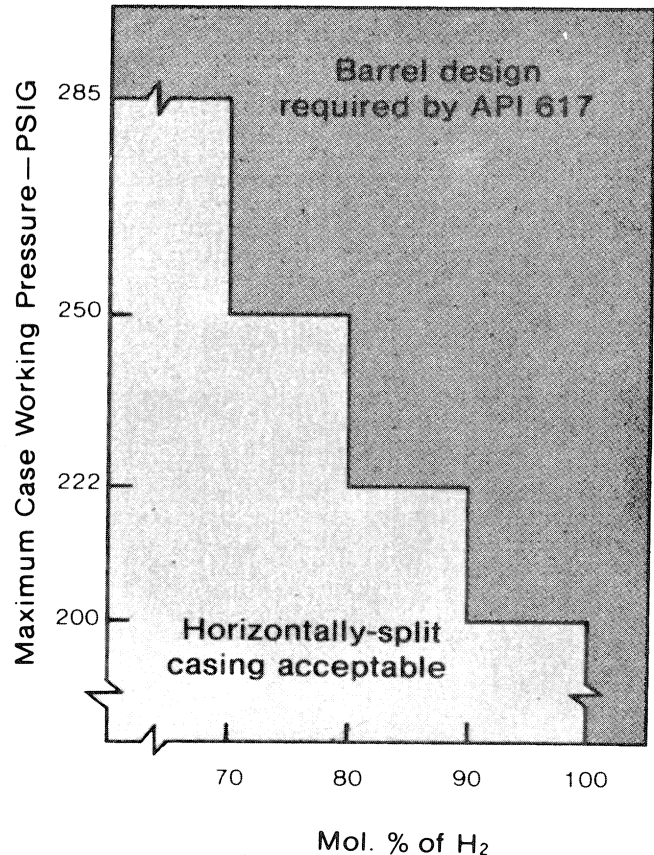


Figure 1. Compressor Casing Selection — API-617 [1].

dling problems become formidable. Our shop is equipped with a 30-ton bridge crane, actually two 15-ton hoists six feet apart, to facilitate easy rigging. Hold-downs are provided in the floor for pulling the bundle from the barrel, as shown in Figure 2.

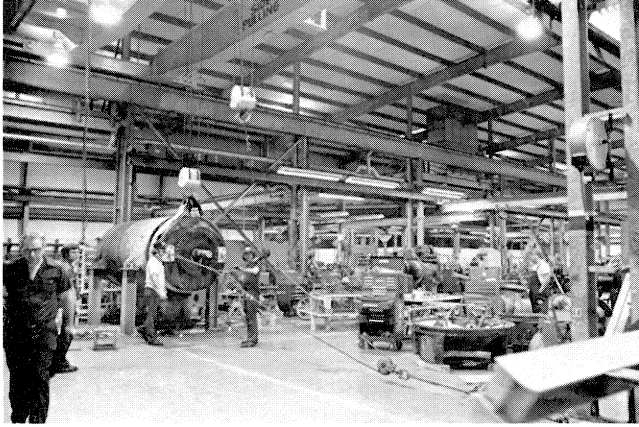


Figure 2. Shop Facilities for Barrel Compressors.

2. *Inner Casing Alignment* — Since this type compressor consists of a bundle contained within the pressure walls of the barrel, alignment and positive positioning is often very poor and the bundle is free to move to a certain extent. Bundle length is critical. Interstage leakage may occur if the bundle length is not correct as shown in Figure 3. The sketch also shows bundle

lengths supplied by the manufacturer *after* severe problems were encountered. Note the very critical tolerances. Assembly errors can be cumulative, particularly in the case of a stacked diaphragm design, and care must be exercised to maintain proper impeller-diaphragm positioning. Since the bundle is subjected to discharge pressure on one end and suction pressure on the other, a force builds up that is transmitted from diaphragm to diaphragm, causing high loading on the inlet wall. The bundle lengths must be maintained carefully.

3. *Internal Leakage* — The discharge and suction compartments of the inner bundle on a straight-through flow design are normally separated by a *single* O-ring. Compressors with side nozzles can have several bundles of O-rings. Excessive bundle-to-barrel clearance may cause leakage past the O-ring(s). In addition, the O-ring(s) is frequently pinched and cut as it passes across the suction nozzle opening in the barrel, a condition that is hard to prevent and doubly hard to detect if it occurs (point "A" in Figure 3).

Pressure differentials in excess of the 400-500 pounds of good design practice can cause extrusion and failure of the O-rings. Amoco has been forced to add back-up rings to the O-rings of seven machines to prevent such failures. Grooves with O-ring ribbons have been added to the horizontal joints of the bundles of almost all of the machines to prevent interstage leakage.

4. *Bearing Bracket Alignment* — In contrast to horizontally split compressors where the bearing brackets are normally an integral part of the lower case half, barrel machine bearing brackets are bolted to the barrel

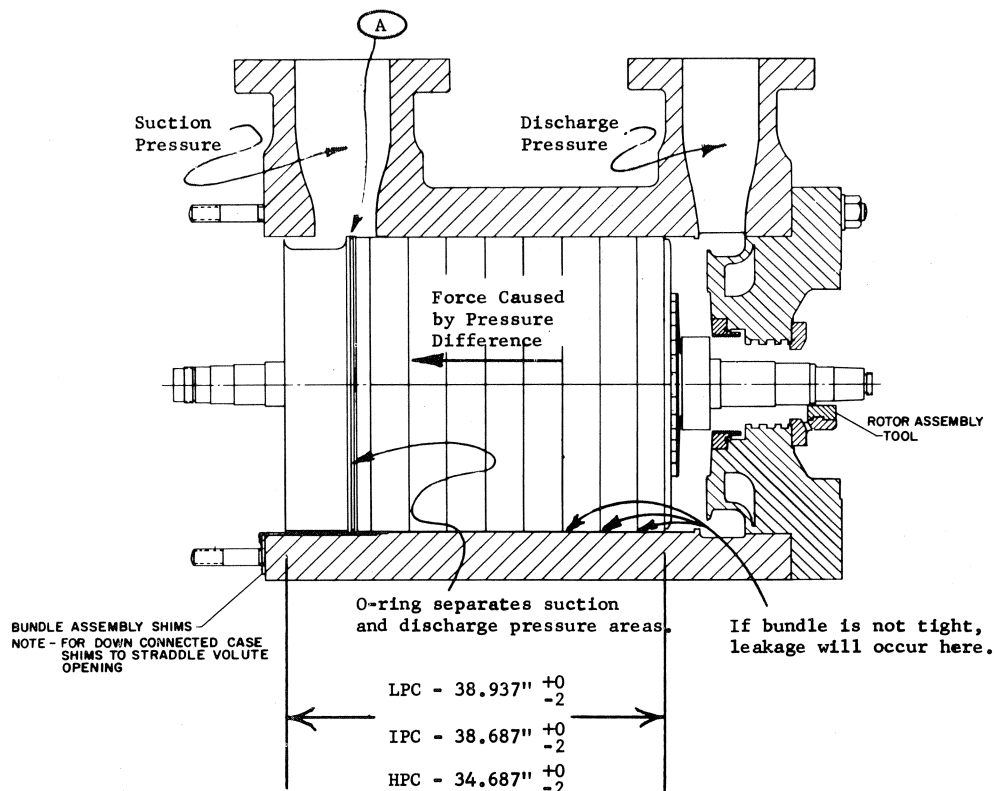


Figure 3. Bundle Assembly Installation.

heads. Both the bearing bracket *and* the head are removed during the disassembly operation, thus requiring *all* internal alignment to be re-established each time maintenance work is performed. This is a time-consuming and exacting procedure that is not spelled out well in maintenance manuals. This procedure, commonly called “setting the lift,” must be done largely by “feeling” the rotor in the bundle. Experience on the part of the mechanic is very essential in this step. After the bundle and rotor are in place, the heads must be made up carefully to properly position the bundle and to align the bearing brackets. For bolt-on heads the following procedures have proven to be satisfactory:

- a. Install a new head gasket and tape in place with “Scotch Tape.” Carefully align head to shaft and case studs. Push into bore.
 - b. Start and firmly drive up four diametrically opposite stud nuts. Remove hoisting equipment from the head. Run up remainder of the stud nuts. Sledge up on diametrically opposite nuts until all are equally tight and check out as follows:
 - (1) With a 0.0015-inch thickness gauge, check at least eight equidistant points between the head and the case. The gauge should “no-go” at all of the points checked.
 - (2) Check the same points with a 0.0010-inch thickness gauge. The gauge should “go.” If necessary, sledge up on the stud nuts to achieve this clearance.
 - c. At this point there should be .010” to .015” clearance between the intake head and the inlet wall. If not, then the intake head must be machined on the “fingers” or the gasket surface to achieve this clearance. If there is too much clearance, the bundle will “leak” at each joint, reducing efficiency. Too little clearance can cause damage to the diaphragm sections.
5. *Material Problems* — In order to limit the physical size of the case or pressure vessel, the rotor bearing span, and to maximize the number of stages within the heavy barrel, the gas path of a barrel is “squeezed” to a greater extent than in a horizontally split machine. This means the diaphragms and inlet guide vanes are intricate shapes with very small openings. Plain gray cast iron is normally used for these shapes because of casting ease and other economic reasons. The gray iron is not strong enough in many instances to withstand the pressure differentials imposed on them, resulting in failures. Inlet guide vanes have been especially troublesome. On several occasions inlet guide vanes have been fabricated from wrought stainless and carbon steel materials (see Figure 4). Replacement diaphragms and inlet guide vanes cast of nodular iron have also been used to alleviate some of these material problems.

COMPRESSORS IN GENERAL

In addition to those problems inherent in barrel-type compressors, there are others that are common to all types of compressors. Some of our troubles and improved maintenance approaches have occurred in these areas.

1. *Rotor Thrust Calculations* — Thrust loads in compressors due to aerodynamic forces are affected by impeller geometry, pressure rise through the compressor, and



Figure 4. Fabricated Inlet Guide Vanes.

internal leakage due to labyrinth clearances. The impeller thrust is calculated, using correction factors to account for internal leakage, and a balance piston size selected to compensate for the impeller thrust load. The common assumptions made in the calculations are:

- a. The radial pressure distribution along the outside of disc cover is essentially balanced.
- b. Only the “eye” area is effective in producing thrust.
- c. The pressure differential applied to “eye” area is equal to the difference between the static pressure at the impeller tip, corrected for the “pumping action” of the disc, and the total pressure at inlet.

These “common assumptions” are grossly erroneous and can be disastrous when applied to high-pressure, barrel-type compressors where a large part of the impeller-generated thrust is compensated by a balance piston. The actual thrust is about 50% more than the calculations indicate. The error is less when the thrust is compensated by opposed impellers, because the mistaken assumptions offset each other.

The magnitude of the thrust is considerably affected by leakage at the impeller labyrinth seals. Increased leakage here produces increased thrust independent of balancing piston labyrinth seal clearance or leakage. A very good discussion of thrust action is found in reference [2].

The thrust errors are further compounded in design of the balancing piston, labyrinths, and line. API-617, “Centrifugal Compressors” [3], specifies that a separate pressure tap connection shall be provided to indicate the pressure in the balance chamber. It also specifies that the balance line shall be sized to handle balance piston labyrinth gas leakage at twice the initial clearance without exceeding the load ratings of the thrust bearing, and that thrust bearings for compressors should be selected at no more than 50% of the bearing manufacturer’s rating.

Many compressor manufacturers design for a balancing piston leakage rate of about 1½-2% of the total compressor flow. Amoco and others feel that the average barrel-type compressor, regardless of vendor, has a leakage rate of 3-4% of the total flow and the balance line must be sized accordingly. This design philosophy would dictate a larger balance line to take care of the increased flow than is normally provided. The balancing chamber in some machines is extremely small and

probably highly susceptible to educting-type action inside the chamber which can increase leakage and increase thrust action. The labyrinth's leakage should not be permitted to exceed a velocity of 10 feet per second across the drum. The short balancing piston design of many designs results in a very high leakage velocity rate.

These problems have been handled at Amoco by retrofitting 28 centrifugal compressors (48% of the total) with improved bearing designs. Most of the emphasis has been toward increased thrust capacity via adoption of a Kingsbury-type design, but journal bearings are always upgraded as part of the package. Design features include spray-lubed thrust bearings (about a dozen cases), copper alloy shoes, ball and socket tilting pad journals, and many other advanced state of the art concepts. In addition to the compressors, 40 steam and gas turbines, large pumps and gear boxes have been fitted with new design bearings. These designs have been the subject of technical papers at previous symposiums [4]. We have found that a good indicator of thrust bearing load and wear is the combined use of thermocouples and position probes.

Some of the balancing piston leakage problems have been solved by the use of honeycomb labyrinths. The use of honeycomb labyrinths offers better control of leakage rates (up to 60% reduction of a straight pass-type labyrinth). Honeycomb seals operate at approximately one-half the radial clearance of conventional labyrinth seals. The honeycomb structure is composed of stainless steel foil about 10 mils thick. Hexagonal-shaped cells make a reinforced structure that provide a larger number of effective throttling points (see Figure 5). In addition, stainless steel honeycomb retains its strength at temperature and pressure levels which would cause weakening of an aluminum labyrinth.

2. *Thrust Collar Designs* — The thrust collar design of many compressors presents some problems. The minimum thrust capacity of a standard 8" (32.0 square inches) Kingsbury-type bearing with flooded lubrication at 10,000 rpm is well in excess of 6½ tons. The thrust collar and its attachment method must be designed to accommodate this load. In most designs the inboard bearing has a solid base ring and the thrust collar must be installed after this thrust bearing is in place. The collar can be checked by revolving the assembled rotor in a lathe. The collar is removed subsequently for seal installation and it must be checked for true; i.e., the face is normal to the axis of the bearing housing again after it is finally fitted to the shaft.

The thrust bearing housing also generates a collar fit-up problem. In most cases a heavy puller cannot be attached to the collar outside diameter to remove it nor can heat be utilized because of space limitations. This limits the fit of the collar to about ½ to 1 mil loose.

The poor thrust collar to shaft fit, the use of a very small and weak collar key, and the use of a flimsy collar spacer and clamping sleeve arrangement all contribute to failures. In one failure we experienced, the collar was broken in three pieces; in another, two pieces. In both instances a corner of the keyway was involved in a fracture line. There have been several instances of thrust collars loosening on the shaft. These failures illustrate the reasons API-670 suggests two thrust posi-

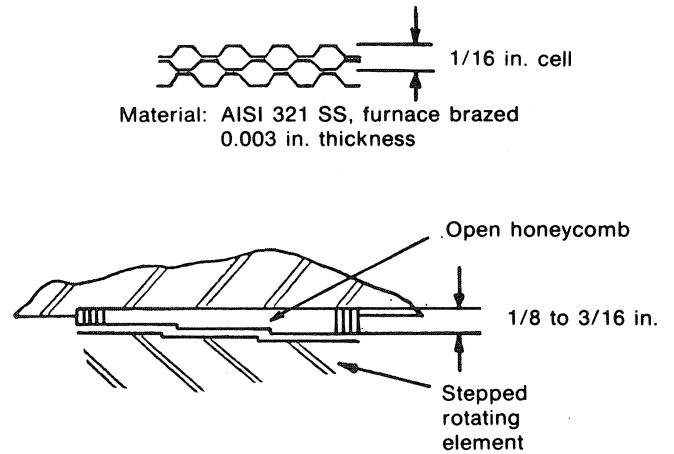


Figure 5. Schematic of Honeycomb Labyrinth Seal Used to Control Leakage Rates.

tion probes: one looking at the thrust collar and one at the shaft or both looking at the end of the shaft. As a solution, we frequently redesign the collar. A favorite design is an "ell" cross section that permits increasing the fit area and thus collar rigidity. This is part of the bearing upgrading.

3. *Thrust Bearing Maintenance* — Tilting pad-type thrust bearings are used in most major pieces of rotating equipment under the general term "Kingsbury." This type of bearing consists of pivoted segments or pads (usually six) against which the thrust collar revolves, forming a wedge-shaped oil film. This film plus minute misalignment of the thrust collar and the bearing pads causes movement and wear of the various bearing parts. The erroneous thrust calculations discussed earlier cause the bearing to be loaded heavier than desired. This accelerates the wear problem. There are seven wear points in the bearing as shown in Figure 6.
 - a. The soft babbitted shoe face.
 - b. The hardened steel shoe insert face (about Rockwell C 30-35 hardness).
 - c. The face of the hardened steel upper leveling plate (about Rockwell C 47-50 hardness).
 - d. The outer edge of the upper leveling plate.
 - e. The upper edge of the lower leveling plate (about Rockwell C 47-50 hardness).
 - f. The pivot point of the lower leveling plate (about Rockwell C 47-50 hardness).
 - g. The inner face of the base ring (about Rockwell C 25-27 hardness).

Note: Hardness numbers are for a Kingsbury bearing.

All of these points must be checked for wear. The leveling plates are normally surface hardened a few mils deep. Because the base ring is the softer component, it is likely to show the most wear. Also, a flat surface is more easily evaluated. Our experience indicates that wear of about 6 mils here will cause "lock-up" of the leveling plates; therefore, we correct, by replacement of parts or a carefully supervised reworking of the entire bearing, wear in excess of 1½-2 mils at this point. All shoe or pad thickness at the pivot point

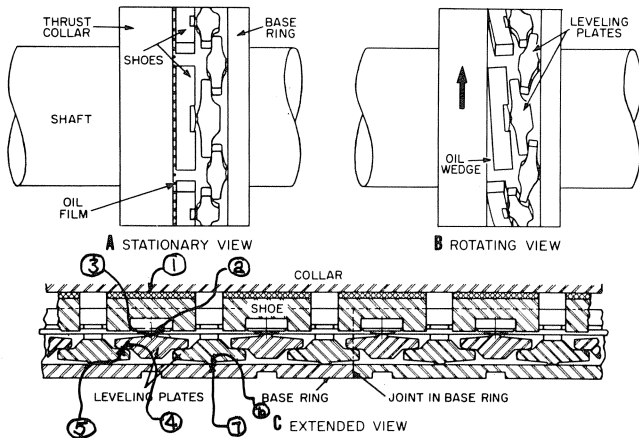


Figure 6. Diagrammatic Arrangement of Tilting Pad Thrust Bearing.

should be within .0005" of the same thickness. The leveling plates wear is difficult to evaluate. Recently we found accumulated wear in excess of 25 mils (each bearing) in an opposed impeller, parallel flow catalytic cracking air blower. The normal thought is that thrust action is zero in this type machine, yet wear was a problem. Because of the tremendous forces that are imposed on a thrust bearing, it must be in good shape. A thorough inspection may prevent machinery failure.

4. *Impeller Design* — The high speed rotation of the impeller of a centrifugal compressor imparts the vital dynamic velocity to the flow within the gas path. The buffeting effects of the gas flow can cause fatigue failures in the conventional fabricated shrouded impeller due to vibration-induced alternating stresses. These may be of the following types:

- a. Resonant vibration in a principal mode.
- b. Forced-undamped vibration, associated with aerodynamic buffeting or high acoustic energy levels.

The vibratory mode most frequently encountered is of the plate-type and involves either the shroud or the disc. Fatigue failure generally originates at the impeller outside diameter, adjacent to a vane. The fatigue crack propagates inward along the nodal line, and finally a section of the shroud or disc tears out (see Figure 7).

To eliminate failures of the plate type, impellers operating at high density levels are frequently scalloped between vanes at the outside diameter. The consequent reduction in disc friction also causes a small increase in stage efficiency [5].

Several rotors have been salvaged by scalloping the wheels after a partial failure has occurred. The eight-stage, 6,520-hp, 10,225 rpm compressor in low molecular weight service, shown in Figure 8, had 54 scallops done to each wheel during an emergency shutdown. To accomplish this, the rotor was unstacked. Each wheel was set up in the milling machine and scalloped. Then each wheel was individually balanced on a mandril. The rotor was restacked and the machine returned to service in slightly over a week's time in our shops.

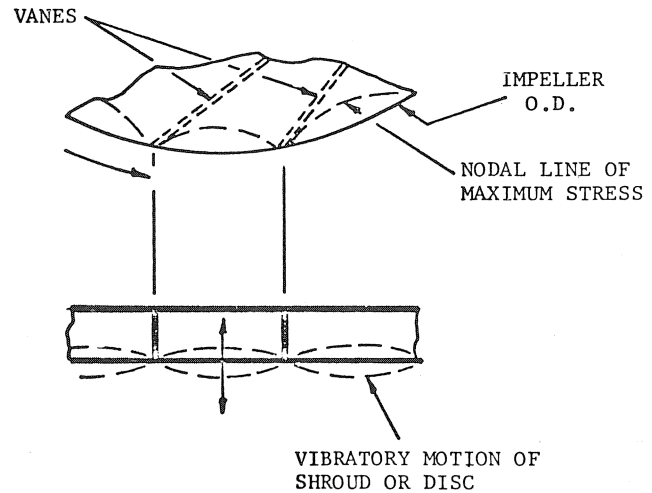


Figure 7. Impeller Vibration in the Plate Mode.

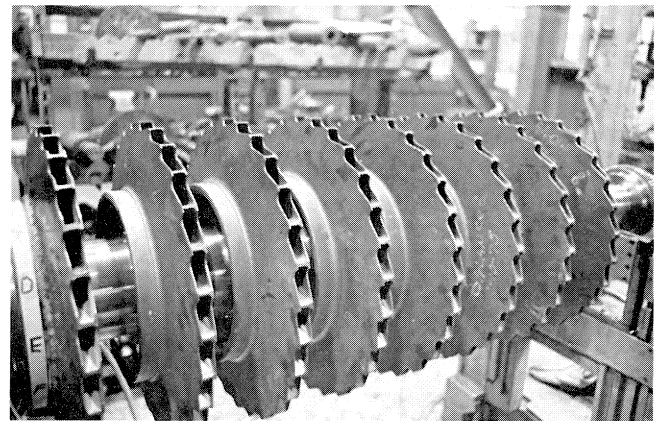


Figure 8. Scalloping of Impellers to Eliminate Plate Mode Vibration Failures.

COMPRESSOR SEALS

The extent of the leakage past the seals where the shaft comes through the casing frequently limits the running time of the compressor, yet the seals and the seal systems are not given adequate treatment in the maintenance manuals or the operating instructions furnished by the compressor manufacturer.

Shaft seals are divided into the following categories by API Standard 617, "Centrifugal Compressors for General Refinery Services":

- Labyrinth
- Restrictive carbon rings
- Mechanical (contact) type
- Liquid film or floating bushing type
- Liquid film-type with pump bushings

The first two seal categories are usually operated dry, and the last three categories require seal oil consoles either separately or as part of the lube system. While each of these seal designs have their own characteristic maintenance difficulties, a discussion of the liquid film bushing-type will illustrate some of them. First, a review of how the seal functions is in order. A liquid film or bushing seal is simply a close-clearance sleeve

surrounding the shaft. Sealing is accomplished by two sleeves which normally are pressurized at a midpoint with 2-4 gpm of seal oil, about 5-15 psi greater than the process gas chamber pressure. Seal oil flows in two directions — through the inboard sleeve to a high-pressure drain area and out through a trap (sour oil), and through an outboard sleeve to an atmospheric drain (sweet oil). The inboard seal oil will absorb some process gas in the drain area; therefore, the inboard oil flow must be kept down to a few gallons per day if a separate oil system is not used. The bulk of the oil circulation passes through the outer sleeve which is somewhat longer than the inner sleeve. This breaks down the greater pressure differential. Since all the oil returned to the sweet oil system passes through the sleeve, the oil temperature will be much higher than a bearing because of extreme shearing of the oil film.

The sleeves are usually made of babbitt-lined steel with 0.002-0.006-inch diametral shaft clearance. The sleeves must float radially; that is, they must follow the shaft movements. Therefore, the sealing face between the end of the sleeve and its housing is very important. The entire assembly, sleeves and housing, is sealed into the casing with suitable O-rings and gaskets to avoid gas leakage and to conduct seal oil to proper compartments.

These concepts, oil flows, and critical clearances are not spelled out well in either the operating instructions or maintenance manuals. Because of this, our experiences with some 46 bushing seals have indicated that several maintenance technique improvements are needed.

1. *Radial Clearances* — Radial clearance between the bushing and the shaft and the length of the bushing must be selected to obtain minimum leakage without exceeding fluid temperature limitations. The clearances recommended by some equipment manufacturers result in exceeding good design temperature levels. We generally run with greater clearances than those recommended by the manufacturer in order to provide good lubrication and cooling for the bushings.
2. *Quality Control* — The flatness, parallelism, and surface finish of the mating sleeve faces must also be carefully controlled to obtain maximum seal effectiveness. Poor quality control by the manufacturer over these parts requires that each part be carefully checked and frequently remanufactured. Such simple things as an incorrect O-ring groove depth can cause malfunctioning of the total seal.
3. *Axial Clearances* — Axial clearance between the bushing or sleeve and the housing is critical, but is completely ignored by most manufacturers. We have found that there should be 12-15 mils clearance per bushing between the bushing or sleeve and the housing. Where the sleeves are mounted back-to-back, there will be 25 to 30 mils clearance total for the seal.
4. *Seal Design* — In higher pressure seals, more than one outboard (i.e., high differential) sleeve may be used. Generally, it is desirable to use a single sleeve because the inboard sleeve operates with up to 80% of the total pressure drop across it. The outer sleeve with the lower differential causes lubrication and cooling problems that can shorten the life of one or both sleeves. In some cases we have to alter the inner bushing so as to allow more oil flow to the outer sleeve.
5. *Training for Seal Maintenance* — In order to get the concepts of seals across to our machinists, we have developed a training film. We have also developed guidelines that show oil flow rates and the interaction

of various components. The oil flow rates, vital to the operation of the seal, are usually buried in a single drawing (the combined lube and seal layout) in the manuals supplied with the machines. The use of simplified diagrams, such as the one in Figure 9, and a training film has aided understanding.

6. *Rules of Thumb* — There are a few rules of thumb that help in understanding seal operation *and* maintenance.
 - a. The oil flow rate will vary
 - (1) *Directly* with the differential pressure and the wetted perimeter of the sleeve.
 - (2) With the *cube* of the radial clearance.
 - (3) With the *square* of the eccentricity of the sleeve and shaft.
 - (4) *Inversely* with oil viscosity, temperature, and length of the sleeve.
 - b. Shear work done on the sealing fluid during its passage through the sleeve raises its temperature to a much higher level than may be expected.

ROTOR REPAIRS

Repair work on the rotating elements of compressors and turbines has traditionally been the field of the original equipment manufacturer, not the equipment owner. Our experiences have pushed us into rotor repair in order to improve reliability and availability of the rotors and the machinery. With these capabilities, a 14,500 hp, 5,920 rpm, eight-stage spare compressor rotor was restored to operating condition less than 30 days after it was damaged. Returning it to a vendor for repairs could have meant a 12-15 month period without a backup rotor in a critical process.

Our shop has a five-man "rotor crew" that works almost exclusively on the repair of the 100 spare rotors owned by Amoco. This group and their supervisor make detailed inspections and evaluations of the condition of a given rotor. In some

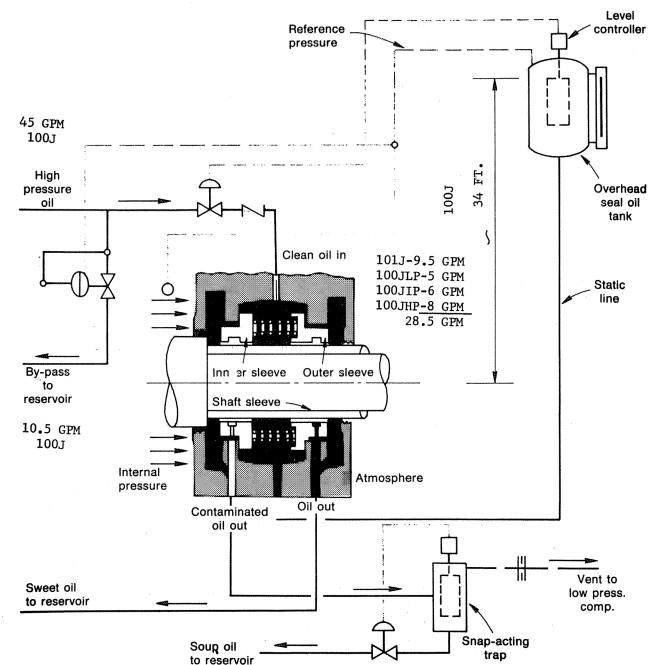


Figure 9. Liquid Film Type Seal System for High Pressure Compressors [6].

instances the decision is made to send the rotor to the original equipment manufacturer (OEM); in some instances it is sent to a local machine shop with rotor repair capabilities, either independent or affiliated with an OEM; or, as in the majority of the cases, the repair work is done in our own shop.

1. *Repairs in Factory* — When the rotor is returned to a vendor's factory, a consultant is often hired to be our on-site representative. His services are normally shared with other companies. This has been a successful approach as it gives us considerable expertise available at the factory over long periods of time without depleting experienced manpower from our refinery activities. A detailed repair list is part of the purchase order and reports are provided by both the consultant and the vendor.
2. *Repairs in Local Shops* — When the rotor is sent to a local shop, the rotor crew foreman makes periodic visits to analyze the work progress and the quality of the repairs. When needed, he can call upon the services of several consultants within the company as well as his supervisors to aid in the decision-making. Many local shops have excellent capabilities for rotor repairs. Impellers have been spin-tested in local facilities to prove their integrity. Local high-speed balancing facilities have also been utilized to check rotor dynamics characteristics in troublesome cases.
3. *Repairs in Amoco's Shops* — The repair work at Amoco is done in a specially equipped area of our shop. The most prominent feature of the area is a stacking pit about eight feet deep with a hydraulic service station lift in the bottom. Use of this lift keeps the working height of the rotor about 36 inches off the floor at all times during the restacking operation. The balancing machine is nearby. A small drill press and a special fitting storage area are in the vicinity.

Training of the rotor crew is given a lot of attention. The younger members are rotated out of the crew every six to eight months to increase the "know-how" distribution. Describing all of this extra effort leads to a question: What do we get out of our in-plant rotor repair program? There are several advantages:

- a. Reduced cost of repairs because more judgment factors can be exercised by the owner than at a distant factory. In addition, overhead costs are reduced.
- b. Quicker repairs result in shorter outage times.
- c. The rotor can be re-engineered to remove problem areas readily; for example, a redesigned thrust collar can be installed or an additional balance plane added.
- d. In-plant skills are developed for emergency use.

As an example of the latter, a process unit upset resulted in damaging the cast bearing housing and bracket on the thrust end of one of our catalytic cracking unit wet gas compressors. The entire horizontally split compressor, weighing nearly 30 tons, was removed to the shop. The "raw material" for the bearing bracket was made from heavy plate forged into cylinder halves, flanges welded on, and the entire assembly stress-relieved. The bracket was then machined to shape as shown in Figure 10. The bearing housing was machined out of solid stock. When the two pieces were fitted together, a perfect match resulted. The reproduced components were so faithful that a machining error made 17 years ago by the manufacturer was inad-

vertently reproduced. The "correcting" shim originally used by the manufacturer had to be reused. We fell short of reaching the expression "everything including the kitchen sink" but we did include the kitchen sink drain as a conduit for the thrust bearings lube oil expulsion. This item, costing \$3.69 at a local hardware store, replaced an expensive (and unavailable) fabricated part. The ingenuity displayed in restoring the machine to service was worth many days of production time.

4. *Other Repairs* — We also reblade steam turbines routinely. The rotor shown in Figure 11 is the driver for the 14,500 hp, 5,920 rpm compressor discussed earlier. The first three stages of blading on both the operating and the spare rotors have been replaced in the last year by Amoco personnel. The expertise for this type work was developed gradually. Several small turbines were rebladed first. A lot of talk preceded the reblading of the first major turbine as several persons contributed their thoughts and experiences. Trips were made to observe vendor shops and their reblading techniques. As self-confidence developed, we began to do selected jobs. The 14,500 hp turbine is our largest reblade job to date.

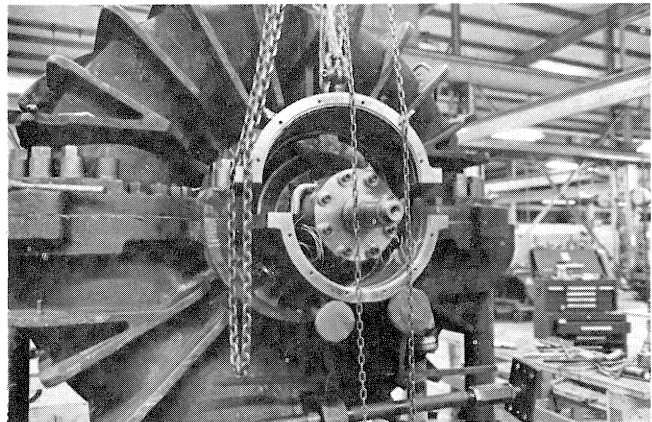


Figure 10. Fabricated Thrust Bearing Bracket and Housing (Note Sink Drain).

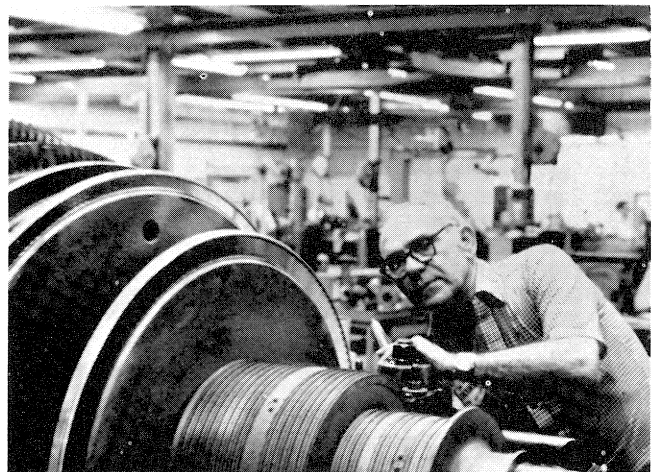


Figure 11. Reblading a 14,500 HP Steam Turbine.

HYDRAULIC FIT COUPLINGS

Hydraulic fit or keyless assembly is becoming an increasingly popular method of mounting couplings, but many maintenance personnel view it with skepticism. The basic concept provides for expanding of the hub bore by pumping oil under high pressure (30,000-35,000 psig) into the clearance space between the shaft and hub. Two O-rings with backup rings are used to prevent the high pressure oil from escaping (see Figure 12). The hub is forced back axially as the bore is expanded by tightening of the bolts on an adapter nut at the forward end. Some manufacturers recommend a hydraulic piston as an advancing tool. When the hub has been advanced onto the taper sufficiently far to produce the correct interference, the oil pressure is relieved and the bore shrinks to produce a very tight fit.

1. *Advantages* — At Amoco hydraulic fit couplings have greatly improved reliability of machinery, reduced vibration levels, lowered coupling wear, and reduced shaft damage due to "fretting" of the coupling hub fit area. The sources of these advantages are:

- a. *Greater Torque Transmission* — More torque (up to 25%) can be transmitted for a given coupling size, since the removal of stress concentrations from keys, splines, or holes allows a greater unit loading capability. We actually utilize this increase as an increase in service factor, not as an increase in capacity.
- b. *Greater Concentricity* — High-centering a hub on a key that is radially too thick is eliminated. Such a condition causes a severe mass eccentricity overhung imbalance. Blue check fitting of keys to prevent binding or shallow keys is eliminated. Since overhung imbalance has a great impact on rotor dynamics, this can be very important.
- c. *Wear* — Since there is no relaxation of stresses at the keyways, uniformity exists under load; i.e., no elliptical runout in the tooth pitch circle. This results in lower gear mesh sliding velocities in gear-type and less flexing in the diaphragm-type couplings — vital factors in wear.
- d. *Shaft Damage* — One of the nice things about hydraulic fitting is ease and simplicity of tooling. No heat is required (no torches) and no gouging from nut-drawing a hub on a shaft. The increased interference fit greatly reduces "fretting" between the hub and the shaft.

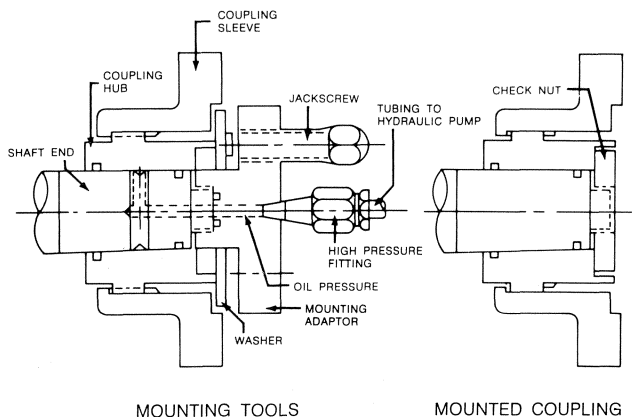


Figure 12. Hydraulic Fit Coupling [10].

e. *Balancing* — Since the hub and shaft are symmetrical, balancing is easier and harder to disturb in service.

2. *Disadvantages* — Hazards — Removal of hydraulic hubs can be hazardous, if improperly attempted. Care must be exercised.
3. *Amount of Interference* — There are no uniformly accepted design practices governing the fit-up interference of coupling hubs on equipment shafts. Experimentation by coupling manufacturers has shown that for steels both the mounting pressure and holding force continue to increase up to a limiting fit value of 0.003 inch per inch of shaft diameter. The hub will remain elastic until the maximum equivalent tensile stress becomes greater than the yield strength of the hub material in tension. In other words, steel hubs of any outside diameter will exhibit a maximum equivalent tensile stress of 30,000 psi if the interference fit is 0.001 inch per inch, 60,000 psi if the fit is 0.002 inch per inch, etc. [7]. The required axial advance is calculated on the basis of about 2-2.7 mils interference per inch of shaft diameter by most manufacturers for a stress level of 60,000 to 82,000 psi.
4. *Hub Fit Guidelines* — The following general guidelines are essential to hydraulic mounting:
 - a. Thoroughly clean the shaft and the hub. Use Prussian blue for checking the contact (minimum 80%). Finish should be no rougher than 63 micro-inch.
 - b. If lapping is required, *DO NOT* match lap the hub on the shaft; improper installation will result. This correction is properly done using plug and ring gauges as masters and a cast iron ring lapping block to lightly dress the entire taper for proper contact. The plug tool (not the master gauge) would be used to dress the bore of the coupling hub. Both these tools must be longer than the fit area by about $\frac{3}{8}$ " in both directions. Naturally, this lapping operation should be minimal, as it will alter the coupling hub position and affect the coupling "float" and its internal clearance. For a dry membrane coupling this may require a spacer shim.
 - c. Push hub back onto the shaft as far as it will go by hand. Record the position of the component relative to a fixed axial position on the shaft (nut shoulder, shaft end, etc.).
 - d. Refer to the appropriate rotor assembly drawing for required interference and the axial advance necessary to achieve the correct interference. This is the distance the hub must be moved beginning from the base position established in paragraph c, above. The advance may also be calculated. Several coupling manufacturers have handy calculation guides to assist you in determining the correct amount [8].
 - e. Install a spacer ring or stop on the shaft, if possible.
 - f. Use new O-rings, Viton, 90 durometer hardness only (standard rings are 70). The O-ring inside diameter must be carefully selected. It should not be stretched, nor should it be crumpled in the groove. The back-up rings should be outboard; the O-rings should face each other.
 - g. Put the coupling hub onto the shaft end and push back by hand as far as it will go. Line up the match marks on the hub and the shaft. This is important to balance the entire rotor.

- h. Tighten jacking bolts to 25 ft.-lbs. Apply hydraulic pressure up to an initial value of 8,000-10,000 psi. Tighten the jacking bolts *uniformly* to 75 ft.-lbs. and observe the motion of the component. If no motion occurs, tighten the jack screws to 90 ft.-lbs. If there is still no motion, raise the pressure slowly in increments of 3,000-5,000 psi, pausing 10-15 minutes between each plateau [9].
 - i. Alternate pumping and tightening the jacking bolts until the hub is seated to the required dimension. Tighten the jack bolts to 90 ft.-lbs. Release the hydraulic pressure, but *do not remove jack bolts for two hours*. This permits oil to flow completely from the clearance space. Failure to do so will require reassembly and could possibly damage the hub or the shaft.
 - j. *Maximum Limits* — Never exceed a maximum of 35,000 psi hydraulic pressure or damage to the components or personnel may occur. *Note that the pressure increases as the jacking bolts are tightened*. The maximum torque applied to the jacking bolts is 90 ft.-lbs.
5. *Training for Coupling Installation* — To aid in training our personnel, a dummy stub shaft and coupling hub have been fabricated. The machinists install and remove the coupling hub until they have a good feel for the techniques. O-ring failures and other problems are introduced and handling methods demonstrated. After several years of experience on ten compressor cases with hydraulically-mounted hubs, we feel this method is far superior to the key-type with mechanical draw.

COUPLING LUBRICATION

One sadly neglected area of coupling maintenance is lubrication. The lubrication requirements of damless marine style and other variations of the gear-type coupling are not always fully appreciated by the equipment vendor or user personnel. Adequate oil levels may not be maintained in the mesh area in the many variations. In order to determine the proper oil flows *through the coupling*, several factors must be considered:

1. Size
2. Rotational speed
3. Basic type of coupling; i.e., marine, dam-type, damless, etc. The damless types require greatly increased oil flows.
4. Method of oil flow through the coupling; i.e., straight through, "U" bend, antisludge holes, etc.

Some recent work by the Koppers Company indicates that accepted lube oil flows in continuous lubed models of a few years ago may only be 50% or less of that required for long life. Also, such simple things as a 15-degree change in mesh spray nozzle orientation can cause a 25-30% reduction in actual oil flow to the coupling. Details are available from Koppers' papers listed in the references [11]. Recently developed greases also permit better lubrication and operation at higher speeds for grease-packed versions of couplings; however, sealing is a major problem.

TURBINE GOVERNORS

Another turbomachinery area we felt in need of improvement at Amoco was speed control. The inadequate capabilities of existing governors, the high maintenance and poor reliability

of governor gear drives and other mechanical components, and the lack of flexibility in existing equipment forced Amoco to design and build improved speed control equipment for our turbomachinery.

Success of this effort can be seen in the conversion of some 100 machines amounting to over 500,000 hp at Texas City and other Amoco locations. Installations up to 43,000 hp with extraction control have been converted to electronic governors of our design. This includes several large gas turbine-steam turbine combination drives.

We did not set out to be competitive speed control designers but were rather forced into this position for self-preservation in machinery maintenance. This equipment subsequently has been licensed for manufacture and is commercially available. Several major turbine manufacturers are now shipping some of their machines with these controls as OEM equipment.

INSTRUCTION BOOKS

Manufacturers' instruction books are often inadequate for maintenance needs. We have resorted to writing slide-tape maintenance guides for the mechanic on such subjects as mechanical seals, vertical pumps, hot-tapping machines, gas turbine overhaul, compressor seals, trip throttle valves, and many others. To increase the impact of the presentations, 35mm slide illustrations showing certain details are used. In most cases, these are professionally made slide and cassette tape training aids. These can be shown quickly before a major turnaround is begun to refresh and review the mechanics on critical details and concepts. Hopefully, the requirements of the new API specifications for "as built" critical dimensions and other additional information will move toward correction of this problem area.

GENERAL UPDATE TRAINING

In order to pass on knowledge and to update the mechanic, unusual experiences or special techniques are written up and distributed to all machinists frequently. The ground rules of this "newspaper" limit it to two typewritten pages, and the use of reduced size sketches and/or drawings is encouraged. These *Machinist Grapevines* are chronicles of our problems and now fill two large loose-leaf binders. Frequent one-hour discussion sessions with supervisory, engineering, and selected hourly personnel are held to improve our "know-how."

IDENTIFICATION OF ELASTOMER MATERIALS

Elastomer materials are widely used in turbomachinery as O-rings. Many different compounds are available, each of which may be a disastrous misfit in the wrong application. Many of the compounds are black or dark gray and look alike. All the careful selection process is for naught if O-ring compounds are accidentally mixed in the maintenance shop! We have found a simple burn test invaluable as a maintenance tool. The burn test requires that a sample of the unknown compound be held in a match flame and that the ash, odor, behavior, and initial appearance be observed for inputs to an identification chart. Of course, the odors created by the test are quite difficult to describe verbally, but once they are smelled when burning a known compound, they are readily identified thereafter. A *Parker Seals* trade publication details this burn technique [12]. A copy should be available in every maintenance shop.

COMPOUNDS USED ON CASING JOINTS

The selection of a joint sealing compound to be used on horizontally split casing joints of compressors and steam turbines is a complex subject and is not spelled out in maintenance instructions. Operating temperature is the most important consideration in selecting the compound to be used. Joint flange condition is the next consideration. The following compounds have proven to be very successful in our experience.

1. *Bodied Linseed Oil*
 - a. Trade name, "Alinco," manufactured by Wheeler Paints.
 - b. Temperature limit — 900°F.
 - c. Description — A thick viscous material that is brushed or troweled on flange.
2. *Silicone Rubber*
 - a. Trade name, "RTV-60," manufactured by General Electric.
 - b. Catalyzed compound — up to 400°F — recommended by the U.S. Navy for saturated steam.
 - c. Uncatalyzed compound — up to 900°F — recommended by the U.S. Navy for superheated steam installations.
 - d. Description — RTV-60 is a two-component silicone rubber compound, red in color, that is superior to the other RTV's such as No. 106.

	RTV-106 (Tube Type)	RTV-60 (Two-Component Type)
Hardness, Shore A Durometer	33	60
Tensile Strength, psi	350	800
Shelf Life, Months	12	6

RTV-60 produces a thinner but stronger film thickness than the one tube material. Catalyzed, it has a pot life of 3-5 hours. It becomes firm within 24 hours. This compound is available in a one-pound packaged form with a premeasured amount of catalyst. The most favorable results are obtained by thinning the compound with an equal volume of mineral spirits and applying with an ordinary paint brush. For the uncatalyzed application, the same instructions apply except no catalyst is added. Although the cure time is not critical for the uncatalyzed, the joint should be closed as soon as the surface of the joint is properly coated in order that foreign material will not contaminate the compound. It takes up to 72 hours for the material to firmly cure under these conditions.

CONCLUSION

Good turbomachinery repair techniques are derived from understanding of design concepts, attention to details, and teamwork by owner, vendor, and selected shops. Training of personnel and providing adequate support facilities are key factors in achieving success. The techniques outlined here, coupled with the careful preoverhaul planning and record systems discussed John Houghton's paper [13] given at the Seventh Turbomachinery Symposium, can result in more reliable efficient machinery overhauls.

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