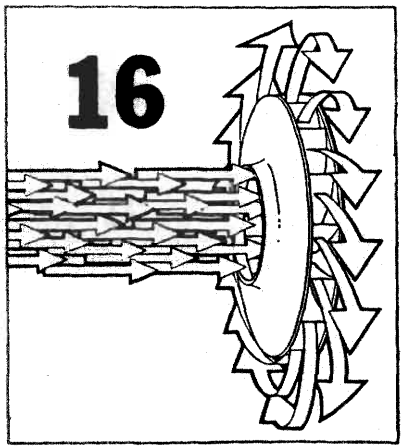
Part 3

Mechanical Design

This page intentionally left blank



Shaft Design and Axial Thrust

Shaft Design

The pump rotor assembly consists of the shaft, impellers, sleeves, bearing or bearing surfaces, and other components such as balancing disks, shaft nuts, and seals that rotate as a unit. The primary component of the rotor assembly is the shaft. The pump shaft transmits driver energy to impellers and consequently to the pumped fluid. This section will be concerned primarily with the sizing of the pump shaft.

The pump shaft is subject to the combined effects of tension, compression, bending, and torsion. As a result of the cyclic nature of the load, when shaft failures occur they are almost exclusively fatigue-type failures. Therefore, the first consideration in sizing the shaft is to limit stresses to a level that will result in a satisfactory fatigue life for the pump. The degree of detail involved in the stress analysis will be dependent upon the intended application of the pump. The analysis can be a simple evaluation of torsional shear stress at the smallest diameter of the shaft or a comprehensive fatigue evaluation taking into consideration the combined loads, number of cycles and stress concentration factors.

Sizing the shaft based on stress is not the only consideration. Shaft deflection, key stresses, fits for mounted components, and rotor dynamics must be evaluated by the designer. The analytic tools available range from simple hand calculations to sophisticated finite element computer programs. The following sections are intended to present the fundamental considerations with which the designer can begin the design of the pump shaft. In some situations, satisfying these fundamental requirements can be considered adequate for a complete shaft design. In other, more critical services, further analysis is required before finalizing the design.

Shaft Sizing Based on Peak Torsional Stress

The stress produced in the shaft as a result of transmitting driver energy to the impellers is torsional. A simple technique for sizing pump shafts is based on limiting the maximum torsional stress to a semi-empirical value. The limiting-stress value is based on the shaft material, operating temperature, and certain design controls on keyway geometry, diameter transitions, and type of application. Since only one stress value is calculated due to one type of load, the limiting stress is obviously kept low. Typical values range from 4,000 psi to 8,500 psi. With this method of shaft sizing, no attempt is made to calculate the effects of stress concentration factors, combined stresses resulting from radial and axial loads, or stresses due to start-up and off-design conditions.

The peak torsional stress is equal to the following:

$$\tau = \frac{Tr}{J} \tag{16-1}$$

The torque is calculated from the maximum anticipated operating horsepower. Special attention should be given to pumps operating with products having a low specific gravity. Shop performance testing will generally be conducted with water as the fluid, and overloading the shaft may occur; hence, performance testing at reduced speed may be required. The shaft diameter used for calculating the stress should be the smallest diameter of the shaft that carries torsional load. For most centrifugal pumps the shaft diameters gradually increase toward the center of the shaft span. This is necessary to facilitate mounting the impellers. As a result the coupling diameter tends to be the smallest diameter carrying torsional load. It is a good design practice to ensure that all reliefs and grooves are not less than the coupling diameter. On some designs, such as single-stage overhung pumps, the smallest shaft diameter is under the impeller, in which case, this diameter shall be used for calculating shaft stress. Reliefs and groove should not be less than this diameter.

Example

Determine the minimum shaft diameter at the coupling for a 4-stage pump operating at 3,560 RPM where the maximum horsepower at the end of the curve is 850 bhp. Use 4140 shaft material with a limiting stress value of 6,500 psi.

Solution

$$\tau = \frac{Tr}{J}$$

$$J = \frac{\pi D^4}{32}$$

$$P = T\omega$$

$$\text{Hence} \bullet \frac{Tr}{J} = \frac{Pr}{\omega J} = \frac{Pr}{\frac{\omega \pi d^4}{32}} = \frac{P16}{\omega \pi D^3}$$

Solving for D:

$$D = \left(\frac{P \times 16}{\omega \pi \tau}\right)^{.333}$$

$$D = \left(\frac{850 \text{ bhp} \times 550 \frac{\text{ft-lb}}{\text{hp-sec}} \times \frac{12 \text{ in.}}{1 \text{ ft}} \times 16}{3{,}560 \frac{\text{Rev}}{\text{min}} \times \frac{2}{\text{Rev}} \times \frac{1 \text{ min}}{60 \text{ sec}} \times \pi^2 \times 6{,}500 \frac{\text{lb}}{\text{in.}^2}}\right)^{.333}$$

$$D = \left(\frac{850 \text{ bhp} \times 321{,}000}{3{,}560 \text{ RPM} \times 6{,}500 \text{ psi}}\right)^{.333} = 2.276 \text{ in.}$$

For design round up to nearest 1/8-in. increment: D = 2.375.

Example

A 2-stage pump has been designed with a 2⅝-in. shaft diameter at the coupling. The maximum horsepower at 3,560 RPM is 900. What is the maximum operating speed for a limiting stress of 7,000 psi?

Solution

Using the pump affinity laws described in Chapter 2:

$$\frac{bhp_2}{bhp_1} = \left(\frac{N_2}{N_1}\right)^3$$

or

$$bhp_2 = bhp_1 \left(\frac{N_2}{N_1}\right)^3$$

From the previous example:

$$\tau = \left(\frac{bhp \times 321{,}000}{N \times D^3}\right)$$

Substituting and solving for N:

$$N_2 = \left(\frac{\tau \times N_1^3 \times D^3}{bhp_1 \times 321{,}000}\right)^{.5}$$

$$N_2 = \left(\frac{7{,}000 \times 3{,}560^3 \times 2.625^3}{900 \times 321{,}000}\right)^{.5}$$

$$N_2 = 4{,}450 \text{ RPM}$$

Shaft Sizing Based on Fatigue Evaluation

Pump shafts are subjected to reversing or fluctuating stresses and can fail even though the actual maximum stresses are much less than the yield strength of the material. A pump shaft is subject to alternating or varying stresses as a result of the static weight and radial load of impellers, pressure pulses as impeller vanes pass diffuser vanes or cutwater lips, driver start-stop cycles, flow anomalies due to pump/driver/system interaction, driver torque variations, and other factors. In order to perform a fatigue analysis, it is first necessary to quantify the various alternating and steady-state loads and establish the number of cycles for the design life. In most cases, the design life is for an infinite number of cycles; however, in the case of start-stop cycles, the design life might be 500 or 1,000 cycles depending on the application.

Once the loads have been defined and the stresses have been calculated, it is necessary to establish what the acceptable stress values are. The use of the maximum-shear-stress theory of failure in conjunction with the Soderberg diagram provides one of the easier methods of determining the acceptable stress level for infinite life (Peterson; Shigley; Roark). Since the primary loads on pump shafts are generally torsion and bending loads, the equation for acceptable loading becomes:

$$\left(\frac{\tau a}{.5Se} + \frac{\tau m}{.5Sy}\right)^2 + \left(\frac{\sigma a}{Se} + \frac{\sigma m}{Sy}\right)^2 \leq 1 \qquad (16\text{-}2)$$

Where τa and σa = alternating stress components
τm and σm = mean stress components
Se = fatigue endurance limit for the shaft material corrected for the effects of temperature, size, surface roughness, and stress concentration factors
Sy = yield strength for the material at the operating temperature

A safety factor is generally applied to Se and Sy to account for unanticipated loads. Equation 16-2 is applied at the location(s) where stresses are the highest.

There are circumstances where it is not necessary to have infinite-life for certain loads. The designer must review all the operating modes and possible upset conditions before a load is classified as a finite-life load. Loads that might be placed in this category are start-stop cycles, and off-design flow, speed, or temperature transients. If the event has an anticipated occurrence of less than 1,000 cycles, it can be considered as a static load with no effect on fatigue life, providing the stresses are less than the material yield strength. For loading conditions of more than 1,000 cycles, but less than 10^7 cycles, the designer has the option to perform a cumulative fatigue damage analysis.

Example

It has been determined that the maximum stresses occur at an impeller-locating ring groove shown in Figure 16-1. The steady state torque is 28,000 in.-lb. The bending moment due to radial hydraulic load is 10,700 in.-lb. Due to axial thrust, there is a tensile force in the shaft of 20,000 lb. Are the stresses at the locating ring acceptable?

Solution

The steady-state loads are the torque and axial load. The alternating bending stress is:

$$\sigma = \frac{Mr}{I}$$

$$I = \frac{\pi D^4}{64} = 3.330 \text{ in.}^4$$

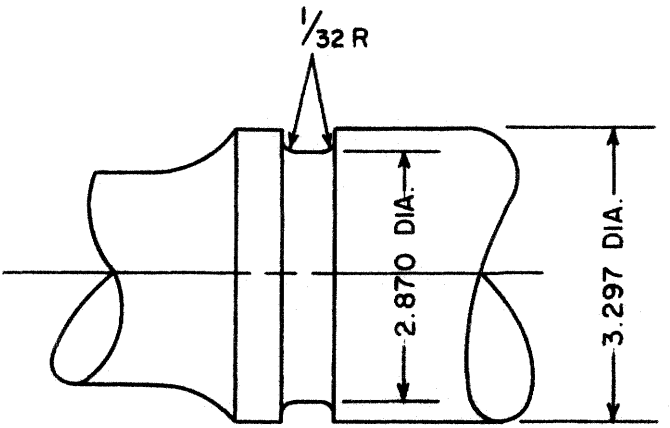


Figure 16-1. Shaft section.

$$\sigma = \frac{10{,}700}{3.330} \text{ in.-lb.} \times \frac{2.870}{2} = 4{,}610 \text{ psi}$$

Yield strength of 4140 steel is 80,000 psi.
The endurance limit for 4140 steel is 52,500 psi. This limit must be adjusted to account for the service condition (Shigley).

$$Se = K_a K_c K_e S'e$$

For 99% reliability use $K_c = .814$. For ground surface finish use $K_a = .9$. For stress concentration due to locating ring groove radius:

$$K_e \cong \frac{1}{K_t}$$

where K_t is the stress concentration factor from Peterson.

$$K_t = 3.75$$

$$K_e = \frac{1}{3.75} = .267$$

$$S'e = 52{,}500 \text{ psi}$$

$$Se = .9 \times .814 \times .267 \times 52{,}500 = 10{,}300 \text{ psi}$$

Torsional stress:

$$\tau = \frac{Tr}{J}$$

$$J = \frac{\pi D^4}{32} = 6.661 \text{ in.}^4$$

$$\tau = \frac{28{,}000 \text{ in.-lb}}{6.661 \text{ in.}^4} \times \frac{2.870}{2} \text{ in.} = 6{,}030 \text{ psi}$$

Axial stress:

$$\sigma = \frac{P}{A}$$

$$A = \frac{\pi D^2}{4} = 6.469 \text{ in.}^2$$

$$\sigma = \frac{20{,}000 \text{ lb}}{6.469 \text{ in.}^2} = 3{,}090 \text{ psi}$$

Hence from Equation 16-2:

$$\left(\frac{6{,}030}{.5 \times 80{,}000}\right)^2 + \left(\frac{4{,}610}{10{,}300} + \frac{3{,}090}{80{,}000}\right)^2$$

$$= .0227 + .237 = .260 < 1$$

Stresses are satisfactory.

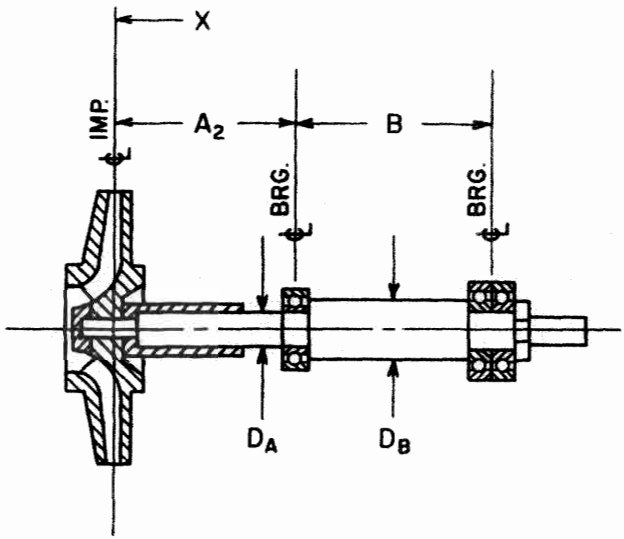


Figure 16-2. Schematic representation of horizontal overhung pump.

Shaft Deflection

Shaft deflection calculations are usually performed on single-stage overhung pumps to establish a relative measure of shaft stiffness. Deflection calculations are also performed on horizontal multi-stage pumps when the potential of galling at wear ring or sleeves exists during start-up or coastdown. The calculation for single-stage overhung pumps assumes that the loading on the shaft consists of weight (for a horizontal pump) and the dynamic radial thrust due to the pump hydraulics.

Figure 16-2 schematically represents a typical horizontal overhung pump (Figure 16-3). The deflection at any point along the shaft between the impeller and the radial bearing can be calculated from the following equation.

$$Y = \frac{W}{3E}\left(\frac{A_2^3}{I_A} + \frac{A_2^2 B}{I_B}\right) - \frac{W}{3E}\left[\times\left(\frac{3\,A_2^2}{2\,I_A} + \frac{A_2 B}{I_B}\right) - \frac{x^3}{2\,I_A}\right]$$

This equation does not account for any support the shaft might receive from the hydrostatic stiffness at the impeller wear rings and shaft throttle

bushing. The deflection calculated by this formula is conservative and should be used for relative comparisons; the actual value will be less.

The calculation of shaft deflection for a horizontal multi-stage pump is complicated by the fact that the distributed weight of the shaft as well as the concentrated weight of the impellers must be accounted for. A method of approximating the shaft deflection can be found in Stepanoff.

Key Stress

A simple but important part of shaft design is calculating key stress. In most centrifugal pumps, driver torque is transmitted across the major

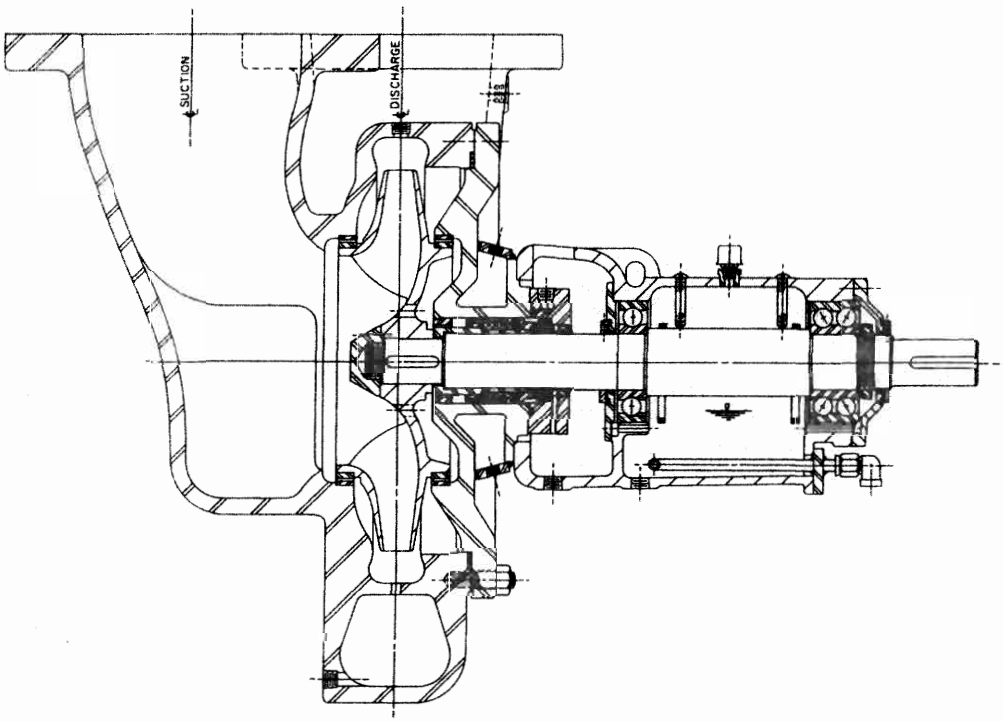


Figure 16-3. Single-stage overhung pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

component interfaces by means of a shear force applied to a key (e.g. driver shaft to driver half coupling hub; pump half coupling hub to pump shaft; and pump shaft to impeller). The general practice is to use a square key which has a height equal to one-fourth of the shaft diameter. (See ANSI-B17.1). The length of the key is determined by the strength requirements and the length of the hub. The strength of the key is evaluated by calculating the bearing and shear stress, assuming that the forces are uniformly distributed along the length of the key.

Example

For a pump with a 2⅞-in. diameter shaft at the coupling and a maximum horsepower of 1,580 at 3,560 RPM, determine the coupling key size. Shaft material is 4140 and the coupling key is 316 stainless steel.

Solution

The mechanical properties of the key material are lower than that of the shaft, so it will be adequate to only examine the key.

The torque to be transmitted from the driver is:

$$T = \frac{\text{bhp} \times 5{,}250}{N} \text{ ft-lb}$$

$$T = \frac{1{,}580 \text{ bhp} \times 5{,}250}{3{,}560 \text{ RPM}} \frac{\text{ft-lb RPM}}{\text{bhp}}$$

$$T = 2{,}330 \text{ ft-lb} = 28{,}000 \text{ in.-lb}$$

The force on the key is:

$$F = \frac{T}{r} = \frac{28{,}000 \text{ in.-lb}}{(2.875/2) \text{ in.}} = 19{,}478 \text{ lb}$$

Key section:

Using diameter divided by 4 as a rule of thumb, a 3/4-in. square key will be used.

Shear stress:

- 316 stainless steel.
- Yield Strength = 30,000 psi.
- For shear use .5 × 30,000 = 15,000 psi.
- With a safety factor of 1.5, the design stress should be 10,000 psi.
- Key shear area = Width × Length.

Shear stress is:

$$\tau_1 = \frac{F}{A_1}$$

Hence

Shaft Design, and Axial Thrust

$$L = \frac{F}{\tau_1 \times W_1} = \frac{19{,}478}{10{,}000 \times .75} = 2.597$$

Use 2 5/8

See Figure 16-4 for a graphic illustration of the solution to this example.

Axial Thrust

In a centrifugal pump, axial thrust loads result from internal pressures acting on the exposed areas of the rotating element. These pressures can be calculated; however, such values should only be considered approximate, as they are affected by many variables. These include location of the impeller relative to the stationary walls, impeller shroud symmetry, surface roughness of the walls, wear ring clearance, and balance hole geometry. Calculated axial thrust is therefore based on a number of assumptions and should only be considered an approximation.

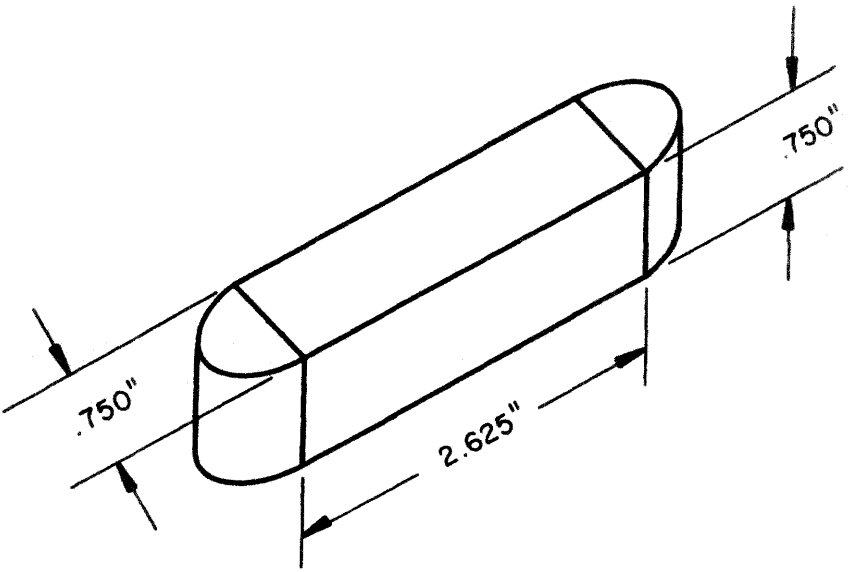


Figure 16-4. Coupling key.

In the methods of calculation described in this section, it is assumed that single- or multi-stage pumps will have 1/4-in. end play in either direction and that the impeller centerline will be located in the center of the volute, within reasonable limits (say ± 1/32 in.). It is also assumed that impeller profile is symmetrical and that the pressure on front and back shroud is equal and constant from the impeller outside diameter to the ring diameter. For simplification, parabolic reduction in pressure is ignored and the axial thrust calculations shown assume an average pressure acting on the impeller shrouds of 3/4 P_D.

On critical pump applications, it is recommended that performance testing include a thrust test. A thrust testing device mounted on the existing bearing housing will measure magnitude and direction of thrust by means of an inboard and outboard load cell. In addition, pressure sensing holes should be drilled through case and cover to record accurate pressure readings acting on the various parts of the rotor. Should the measured thrust be unacceptable, internal diameter changes can be analyzed and modified to reduce thrust to an acceptable level. This procedure is also useful for field observations on pumps experiencing thrust bearing problems and removes any doubt if in fact the problem is a result of excessive thrust.

Approximate methods of calculation for various pump configurations are now described.

Double-Suction Single-Stage Pumps

Theoretically double-suction pumps with double-suction impellers are in axial balance due to the symmetrical mechanical arrangement. If the flow approach to the impeller is also symmetrical, there is no need for thrust bearings. However, as minor mechanical or casting variations will cause axial unbalance, small capacity thrust bearings are normally supplied.

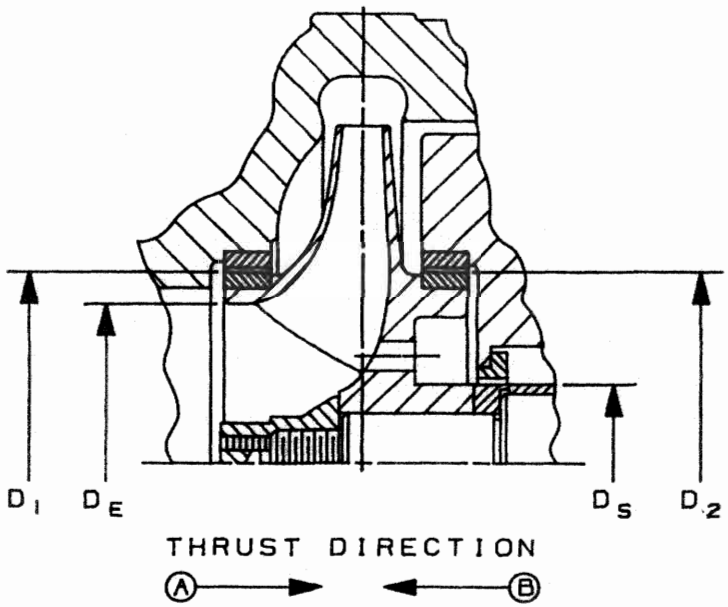
If symmetry of this type of pump is violated by relocating the impeller away from volute center, the developed pressure above the impeller rings on either side of the impeller will not be the same and substantial axial thrust will be developed. The amount of thrust will vary directly with axial displacement.

Single-Suction Single-Stage Overhung Pumps

Single-suction closed or open impellers have different geometries on either side of the impeller and thus develop substantial axial thrust. Calculating this thrust requires analysis of the different design variations normally used in commercial pumps.

Closed impeller with rings on both sides and balancing holes in back shroud. Balance holes through the impeller allow liquid leaking across the wear ring clearance to flow back to the suction side of the impeller. This aids in reducing thrust as the pressure behind the balance holes will be less than the pressure above the wear ring but greater than the suction pressure. Many simplified axial thrust calculations assume the pressure behind the balance holes to be the same as suction pressure; however, testing on small process pumps has shown this pressure to range from 2% to 10% pump differential pressure, depending on balance hole geometry. The calculation method shown in Figure 16-5 assumes liberal clearance holes with the hole area being a minimum of eight times the area between the wear rings. In the event that jetting action of leakage behind the back ring clearance creates a wear pattern on the back cover, it is often necessary to install a hardened ring on the pump cover.

Closed impeller with front ring only. All things being symmetrical, this impeller is axially balanced from the impeller outside diameter down to the suction ring diameter. Below this, there is an unbalance force acting on the area between the suction ring and the shaft sleeve. For the calculation method see Figure 16-6.



WHERE $D_1 = D_2$

NET THRUST $T = A_S P_S - [.03 P_D (A_1 - A_S)] + F_X$

WHERE $D_1 \neq D_2$

THRUST DIRECTION (A) $T_A = A_S P_S - [.03 P_D (A_1 - A_S)] + F_X$

THRUST DIRECTION (B) $T_B = 3/4\ P_D (A_1 - A_2)$

NET THRUST $T = T_A - T_B$

Figure 16-5. Axial thrust on single-suction closed impellers with balance holes.

Closed impeller with front ring and back balancing ribs. Where it is not desirable to design the impeller with a back ring, axial thrust can be balanced by designing integrally cast ribs on the back of the impeller shroud. A gap between rib and cover of 0.015 in. maximum, permits the liquid to rotate at approximately full impeller angular velocity, thus reducing the pressure on the impeller back shroud over the area of the rib. This method of balance is not very accurate or positive. If the gap is increased, the pressure acting on the back shroud increases, resulting in higher axial loads. Experience has shown that this type of impeller is less efficient than one with a back ring. Axial thrust can be estimated as shown in Figure 16-7.

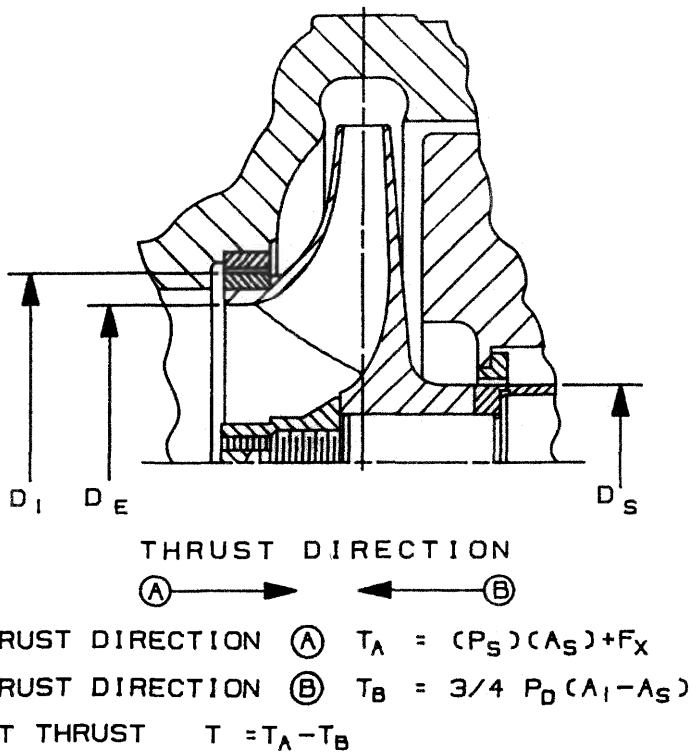
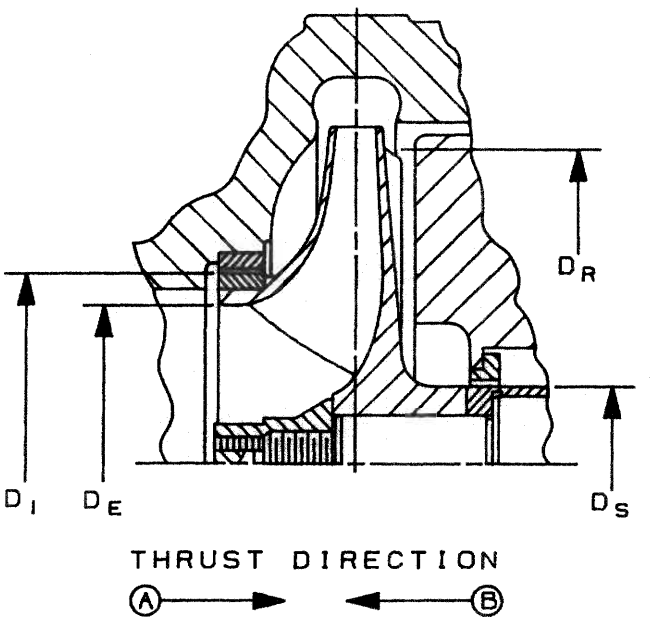


Figure 16-6. Axial thrust on single-suction closed impellers with no rings on back side.

Semi-open impellers. Depending on level of thrust, semi-open impellers can be designed with or without back ribs. Pumps of this type are quite often troublesome and inefficient. The gap between the open vane and liner must be carefully controlled at assembly, as any increase in this gap will affect the head developed by the pump. The problem is compounded when the back rib is added, as gaps on each side of the impeller must be controlled. Methods for calculating axial thrust are shown in Figures 16-8 and 16-9.

Multi-Stage Pumps

Axial thrust becomes more important with multi-stage pumps, because of the higher developed pressure and combined thrust of several stages. Impeller arrangement will normally be back to back for volute-type



THRUST DIRECTION Ⓐ

$$T_A = (A_S)(P_S) + [(.00253)(A_R - A_S)(U_R^2 - U_S^2)(SP.GR.)] + F_X$$

THRUST DIRECTION Ⓑ

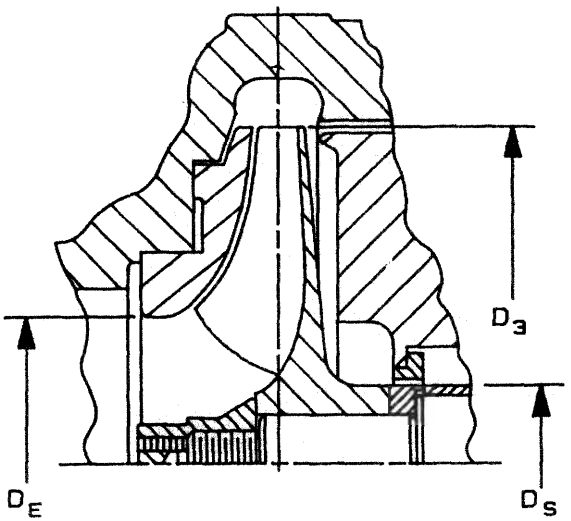
$$T_B = 3/4\ P_D (A_I - A_S)$$

NET THRUST $T = T_A - T_B$

Figure 16-7. Axial thrust on single-suction closed impeller with ribs on back side.

pumps and stacked (all impellers facing one direction) for diffuser pumps.

Back-to-back impellers. This arrangement simplifies axial balance for an odd or even number of stages. In this design there are two locations on the rotating element that break down half the total pump head. One is the center bushing between the center back-to-back impellers, and the other is the throttle bushing in the high-pressure stuffing box. Sleeve diameters at these locations can be sized to provide axial balance for any number of stages.



THRUST DIRECTION

Ⓐ ⟶ ⟵ Ⓑ

THRUST DIRECTION Ⓐ

$$T_A = (A_S)(P_S) + \left[(A_3 - A_E)\frac{P_D}{2}\right] + F_X$$

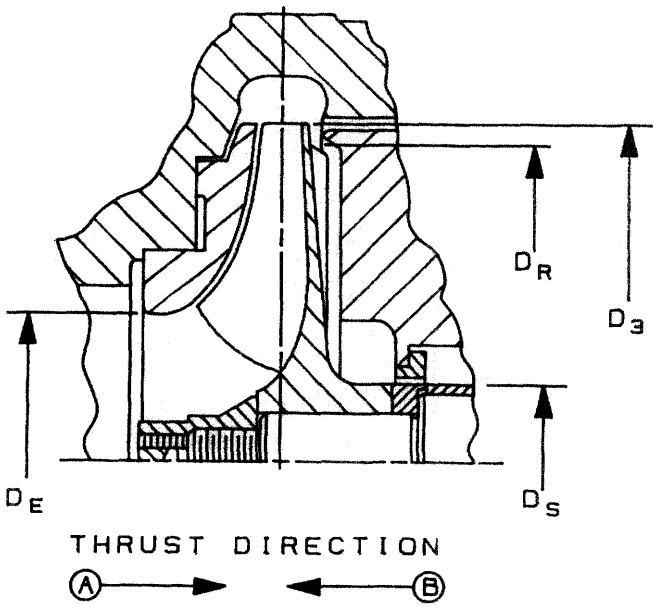
THRUST DIRECTION Ⓑ

$$T_B = 3/4\ P_D\ (A_3 - A_S)$$

NET THRUST $T = T_A - T_B$

Figure 16-8. Axial thrust on single-suction open impellers without ribs on back side.

It is recommended that the element maintain a small unbalance (minimum 100 lb) in one direction to lightly load the bearing and prevent floating of the rotor. Normal wear of the original ring or bushing clearance will not affect balance. It is recommended that all impeller rings be the same size and on even-stage pumps that the center sleeve diameter be the same as the sleeve diameters in each stuffing box. To balance an odd number of stages, the simplest solution is to change only the diameter of the center sleeve.



THRUST DIRECTION Ⓐ

$$T_A = (A_S)(P_S) + [(.00253)(A_R - A_S)(U_R^2 - U_S^2)(SP.GR.)] + [(A_3 - A_E)\frac{P_D}{2}] + F_X$$

THRUST DIRECTION Ⓑ

$$T_B = 3/4\ P_D\ (A_3 - A_S)$$

NET THRUST $T = T_A - T_B$

Figure 16-9. Axial thrust on single-suction open impeller with ribs on back side.

Stacked impellers. With all impellers facing in one direction, the area between the last impeller and the high-pressure stuffing box is under full discharge pressure. Pressure is broken down across a balance drum, with the diameter of the drum sized for axial balance. Wear on the balance drum will affect axial balance, therefore bearing rating should be sized accordingly. On critical applications, particularly where a high rate of erosion wear is anticipated, the permanent installation of load cells should be considered.

The low-pressure side of the balance drum is piped back to suction, as is the low-pressure side of the throttle bushing in the back-to-back ar-

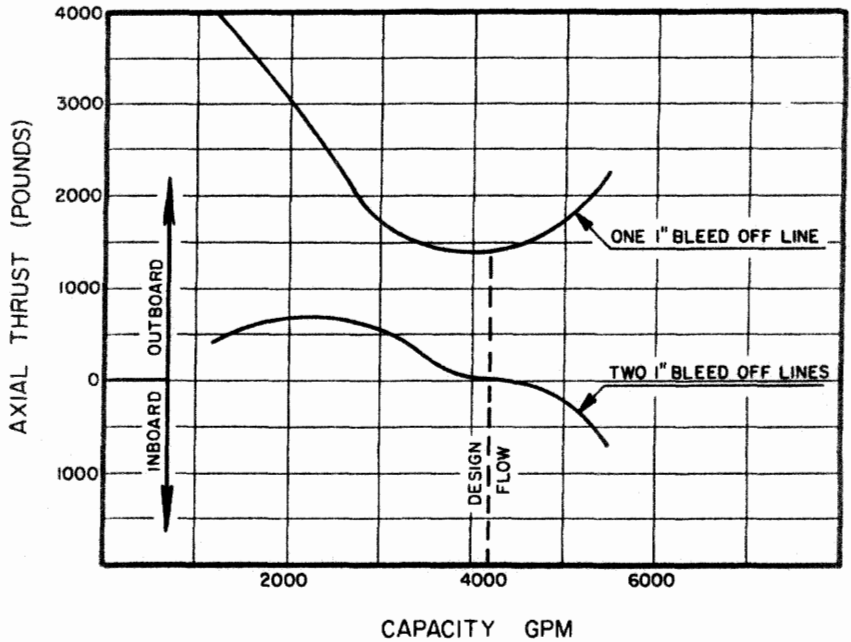


Figure 16-10. Increasing number of bleedoff lines reduces thrust.

rangement. This aids axial balance and subjects the inboard and outboard mechanical seal to only suction pressure. It is essential that the bleed-off line be adequately sized to transfer leakage back to suction without permitting a pressure build-up in the stuffing box. Any such build-up of pressure will have an adverse effect on axial thrust. If pressure build-up is detected, or suspected, the bleed-off line should be increased in size or additional lines added. This principle also applies to overhung process pumps with balance holes, where the back pressure can be reduced by increasing the size of the holes. The effect on thrust of a two-stage pump with one and two balance lines is illustrated in Figure 16-10. Note, that the thrust level and direction of thrust is not constant across the capacity range of the pump. This tendency is typical of most pumps.

Notation

Shaft Design

τ_1 Shear stress (psi)
τ Torsional stress (psi)

T	Torque (lb-in.)
r	Radius of shaft (in.)
J	Polar area moment of inertia (in.^{4})
P	Power (ft-lb/sec)
ω	Angular velocity (rad./sec)
D (with or without subscript)	Diameter of shaft as indicated (in.)
N	Speed (RPM)
M	Bending moment due to radial hydraulic load (in.-lb)
I (with or without subscript)	Moment of inertia of shaft as indicated (in.^{4}) $= \frac{\pi D^4}{64}$
Se	Endurance limit of mechanical element (psi)
S′e	Endurance limit of rotating beam specimen (psi)
Ka	Surface finish factor
Kc	Reliability factor
Ke	Stress concentration factor
A	Area of shaft cross section (sq in.)
W	Impeller weight plus radial thrust (lb)
E	Modulus of elasticity of shaft (psi)
A_2	Distance from centerline of impeller to centerline of inboard bearing (in.)
B	Bearing span centerline to centerline (in.)
X	Distance from centerline of impeller to location on shaft where deflection is to be calculated (in.)
bhp	Brake horsepower
A_1	Key shear area (sq in.)
Y	Deflection at any point on shaft (in.)
F	Force (lb)
L	Useful key length (in.)
W_1	Key width (in.)

Axial Thrust

A (with subscript)	Area of cross section as indicated by subscript (sq in.) $= \pi D^2/4$.

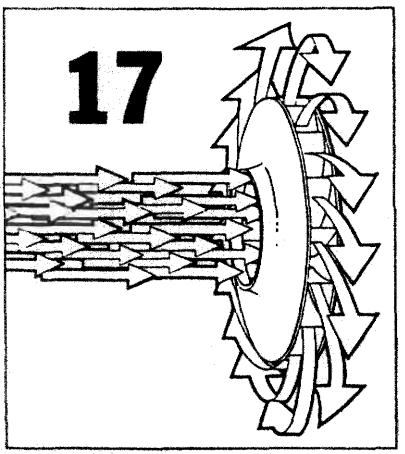
D (with subscript)	Diameter of cross section as indicated by subscript (in.).
P_D	Differential pressure (psi).
P_S	Suction pressure (psi).
T	Net axial thrust (lb).
T (with subscript)	Axial thrust in direction as indicated by subscript. (lb)
U (with subscript)	Peripheral velocity at diameter indicated by subscript (ft/sec) = D × RPM/229.
sp gr	Specific gravity.
g	32.2 ft/sec^2 (constant used in calculation for Figures 16-14 and 16-16)
γ	Fluid density of water lb/ft^3 (constant used in calculation for Figures 16-14 and 16-16.
F_X	Thrust due to momentum change (lb) = $(Q^2 \times \text{sp gr})/A_E \times 722$
Q	GPM

References

Peterson, R. E., *Stress Concentration Factors,* John Wiley and Sons, Inc., New York, 1974.

Roark, R. J., and Young, W. C., *Formula For Stress and Strain,* 5th edition, McGraw-Hill, Inc., 1975.

Shigley, J. E. *Mechanical Engineering Design,* 3rd edition, McGraw-Hill, Inc., 1977.



Mechanical Seals

by **James P. Netzel**
John Crane Inc.

The most common way to seal a centrifugal pump is with a mechanical seal. The basic components of a mechanical seal are the primary and mating rings. When in contact they form the dynamic sealing surfaces that are perpendicular to the shaft. The primary ring is flexibly mounted in the seal head assembly, which usually rotates with the shaft. The mating ring with a static seal, forms another assembly that is usually fixed to the pump gland plate (Figure 17-1). Each of the sealing planes on the primary and mating rings is lapped flat to eliminate any visible leakage. There are three points of sealing common to all mechanical seal installations:

- At the mating surfaces of the primary and mating rings.
- Between the rotating component and the shaft or sleeve.
- Between the stationary component and the gland plate.

When a seal is installed on a sleeve, there is an additional point of sealing between the shaft and sleeve. Certain mating ring designs may also require an additional seal between the gland plate and stuffing box.

Normally the mating surfaces of the seal are made of dissimilar materials and held in contact with a spring. Preload from the spring pressure holds the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

The secondary seal between the shaft or sleeve must be partially dynamic. As the seal faces wear, the primary ring must move slightly forward. Because of vibration from the machinery, shaft runout, and thermal expansion of the shaft to the pump casing, the secondary seal must

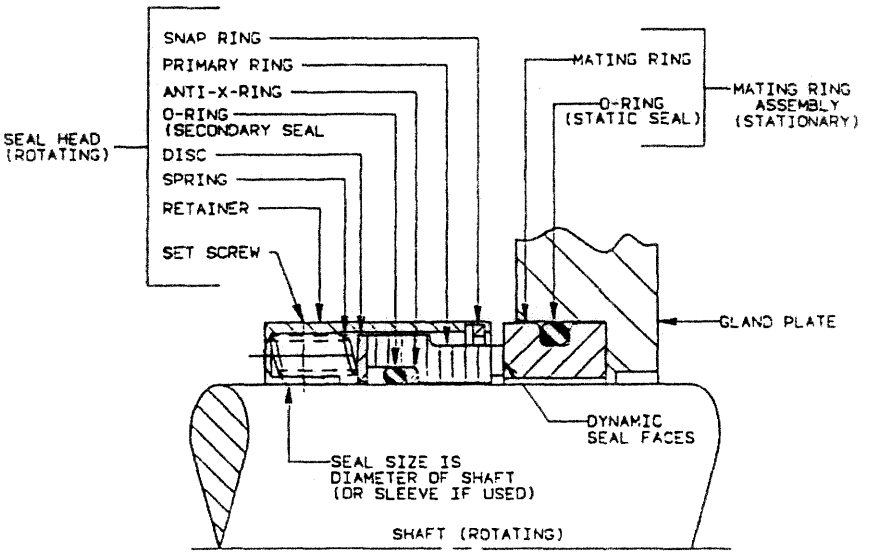


Figure 17-1. Mechanical seal.

move along the shaft. This is not a static seal in the assembly. Flexibility in sealing is achieved from such secondary seal forms as a bellows, O-ring, wedge, or V-ring. Most seal designs fix the seal head to the sleeve and provide for a positive drive to the primary ring.

Although mechanical seals may differ in various physical aspects, they are fundamentally the same in principle. The wide variation in design is a result of the many methods used to provide flexibility, ease of installation, and economy.

Theory of Operation

Successful operation of a seal depends upon developing a lubricating film and controlling the frictional heat developed at the seal faces. Whenever relative motion occurs between the primary and mating rings, frictional heat is generated. Normally, cooling of the seal faces is accomplished by a seal flush. Many theories have been developed to explain the formation of the lubricating film. These theories include the types of motion transferred to the seal, thermal distortion of the sealing plane, and surface waviness of both the primary and mating rings (Figure 17-2). The generation of a lubricating film is particularly important if the seal is run for a long period of time without exhibiting any appreciable wear at

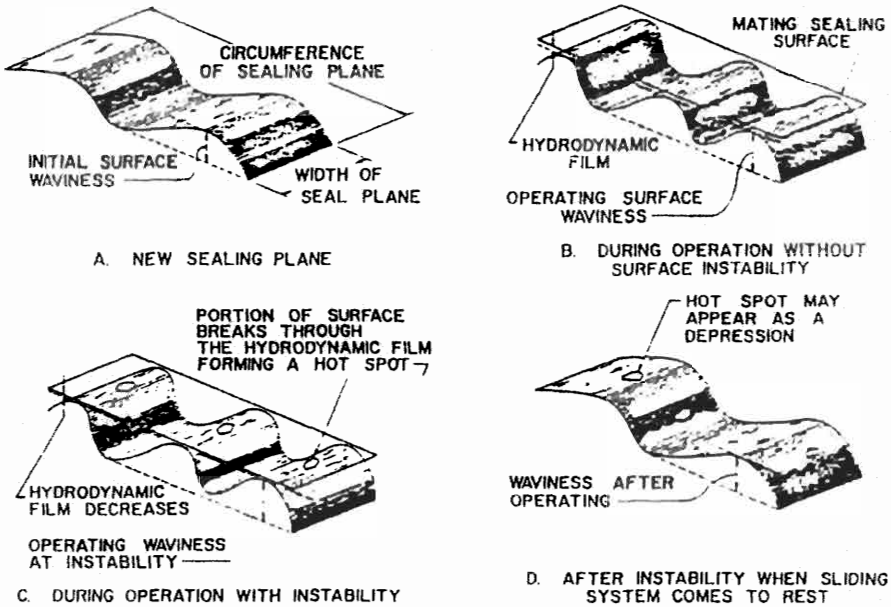


Figure 17-2. Mechanical seal waviness for different conditions of operation.

the faces or any seal leakage. Surface waviness is believed to be the primary reason for the generation of a lubricating film. Despite the fact that both seal faces are lapped flat, a very small amount of initial waviness remains (Figure 17-2A). For a hard seal face like tungsten carbide, the initial surface waviness will be 2 to 8 micro-inches. Softer seal face materials like carbon graphite would be within the range of 10 to 16 micro-inches. As sliding contact occurs between the primary and mating rings of the seal, frictional heat develops that increases the surface waviness of each seal face from its initial value to some operating waviness. One of the consequences of waviness is that the changes to the sealing surfaces are not uniform. However, continuous operation without solid contact between the seal faces is the result of an ideal wave formation for the operating conditions of the seal. This also results in the development of a lubricating film and a long life for the seal installation (Figure 17-2B).

Under certain conditions non-uniform heating of the seal faces may lead to even more concentration of friction. Portions of the sealing surfaces will break through the lubricating film, resulting in solid contact and increased frictional heat in localized areas. The formation of highly stressed hot spots on the seal face will reduce the effects of the lubricat-

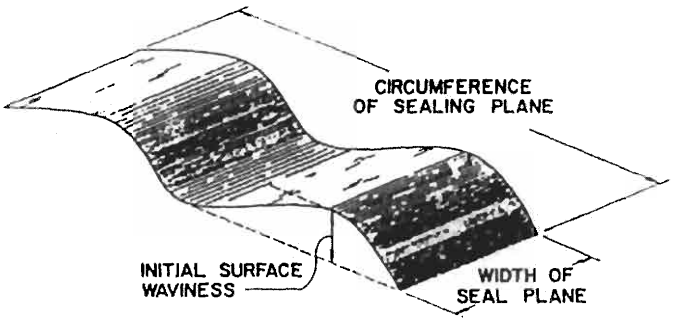


Figure 17-2A. New sealing plane.

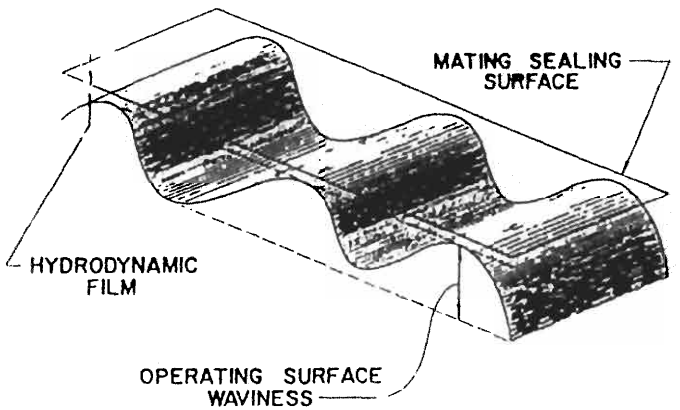


Figure 17-2B. During operation without surface instability.

ing film and these spots will grow in intensity, generating still more heat. This intense heat may cause the liquid being sealed to flash or vaporize, resulting in unstable operation of the seal. The transition from a normally flat condition to a highly deformed surface is called thermoelastic instability (Figure 17-2C).

During the mid-seventies, many experiments were conducted to determine if this could be present in mechanical seals. Burton et al. (1980) demonstrated this phenomenon by running a seal configuration against a glass plate. Hot patches or spots appeared on the surface of the metal primary ring. These spots moved through the entire circumference of the seal. Movement of these spots is believed to be the result of wear. Confirmation of hot patches or spots on conventional materials used in mechanical seals was determined by Kennedy (1984). Localized hot spots from

thermoelastic instability has been observed under both dry and liquid lubricated conditions.

Surface distress at a hot spot occurs because of rapid heating in operation, followed by rapid cooling. When a liquid film at the seal faces flashes, the seal opens for an instant, allowing cool liquid into the seal faces. This then heats up only to flash again until the surface cracks. These thermo cracks are attributed directly to the high thermal stresses near the small hot spot. When the seal faces are removed from the equipment, the location of the distressed hot spots may not have any particular relationship to the final surface waviness (Figure 17-2D).

Stable operation of a seal is achieved through control on surface waviness and thermoelastic instability. This is accomplished through cooling and seal face design. Figure 17-3 illustrates waviness traces for a set of 3.94-inch mechanical seal faces run in liquid ethane for 3,056 hours without any visible leakage. The pressure and shaft speed were 820 lbs $in.^{-2}$ and 3,600 revolutions min^{-1}. The temperature was 48°-60°F. The surface waviness of the carbon ring is 75 micro-inches peak to peak, while the tungsten carbide surface has a waviness of approximately 12 micro-inches. These traces are typical of a good running seal.

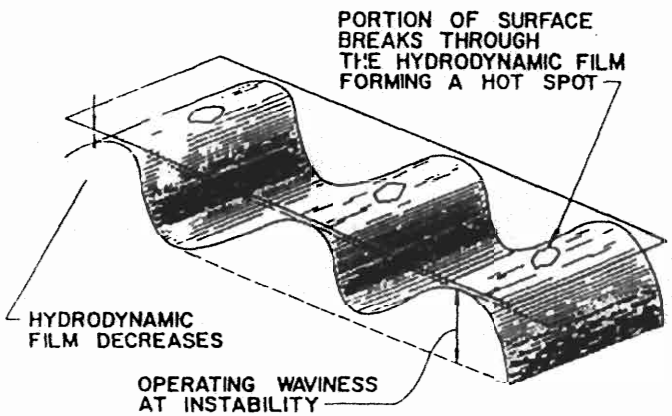


Figure 17-2C. During operation with instability.

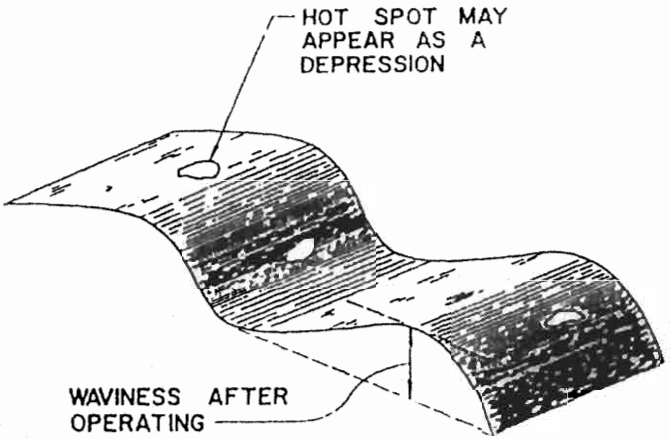


Figure 17-2D. After instability when sliding system comes to rest.

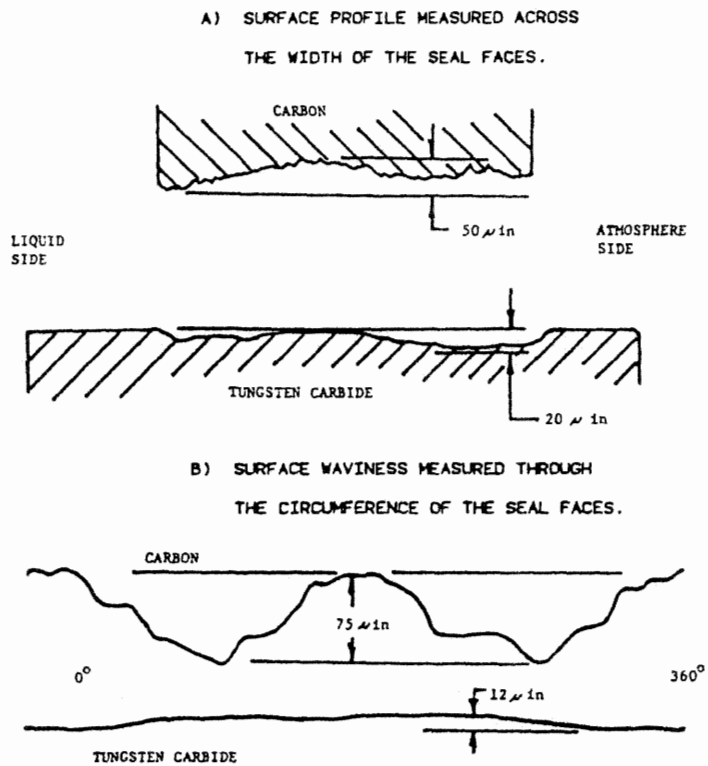


Figure 17-3. Wear of mechanical seal faces after 3056 hours of operation.

Design Fundamentals

Seal Balance. This is the ratio of hydraulic closing area to seal face area:

$$b = \frac{a_c}{a_o}$$

where b = seal balance
a_c = hydraulic closing area in.^{2}
a_o = hydraulic opening area (seal face area), in.^{2}

Seal balance is used to reduce power loss at seal faces from sliding contact. If the load at the seal faces is too high, the liquid film between the seal rings could be squeezed out or vaporized. An unstable condition from thermoelastic instability could result in a high wear rate of the sealing surfaces. Seal face materials also have a bearing limit that should not be exceeded. Seal balancing can avoid these conditions and lead to a more efficient installation.

The pressure in any stuffing box acts equally in all directions and forces the primary ring against the mating ring. Pressure acts only on the annular area a_c (Figure 17-4A) so that the closing force in pounds on the seal face is:

$$F_c = pa_c$$

where p = stuffing box pressure, lb/in.^{2}
F_c = hydraulic closing force, pounds

The pressure in pounds per square inch between the primary and mating rings is:

$$\frac{F_c}{a_o} = \frac{pa_c}{a_o}$$

To relieve the pressure at the seal faces, the relationship between the opening and closing force can be controlled. If a_o is held constant and a_c decreased by a shoulder on a sleeve or on the seal hardware, the seal face pressure can be lowered (Figure 17-4B). This is called seal balancing. A seal without a shoulder in the design is an unbalanced seal. A balanced seal is designed to operate with a shoulder. Only a metal bellows seal is a balanced seal that does not require a shoulder. The balance diameter is the mean area diameter of the bellows.

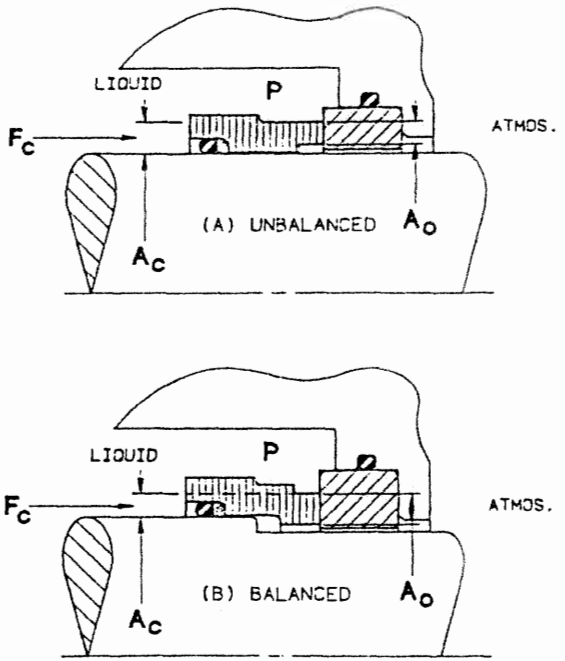


Figure 17-4. Hydraulic pressure acting on the primary ring.

Seals can be balanced for pressures at the outside diameter of the seal faces as shown in Figure 17-4B. This is typical for a seal mounted inside the stuffing box. Seals mounted outside the stuffing box can be balanced for pressure at the inside diameter of the seal faces. In special cases, seals can be double balanced for pressure at both the outside and inside diameters of the seal. Seal balance can range from 0.65 to 1.35, depending on operating conditions.

Face Pressure. This is an important factor in the success or failure of a mechanical seal. Hydraulic pressure develops within the seal faces that tend to separate the primary and mating rings. This pressure distribution is referred to as a pressure wedge (Figure 17-5). For most calculations it may be considered as linear. The actual face pressure P_f in pounds per square inch is the sum of the hydraulic pressure P_h and the spring pressure P_{sp} designed into the mechanical seal. The face pressure P_f is a further refinement of P which does not take into account the liquid film pressure or the load of the mechanical seal:

$$P_f = P_h + P_{sp}$$

where $P_h = \Delta P(b - k)$, lb/in^2
P = pressure differential across seal face, lb/in^2
b = seal balance
k = pressure gradient factor

The pressure gradient factor is normally taken as 0.5 for a given installation. The mechanical pressure for a seal is:

$$P_{sp} = \frac{F_{sp}}{a_o} \text{ lb/in.}^2$$

where F_{sp} = seal spring load, lb
a_o = seal face area in.^{2}

then the actual face pressure can be expressed as:

$$P_f = \Delta P(b - k) + P_{sp}$$

The actual face pressure is used in the estimate of the operating pressure-velocity for a given seal installation.

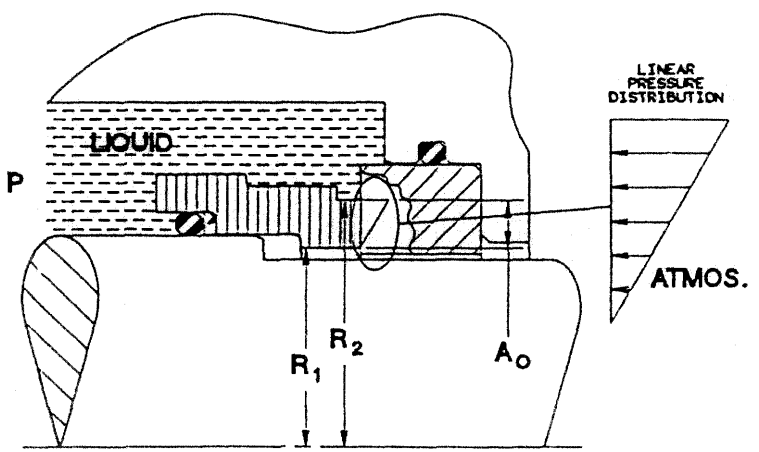


Figure 17-5. Pressure wedge opening force at the seal faces.

Pressure-Velocity. The value for a seal installation may be compared with values developed by seal manufacturers as a measure of adhesive wear. As the primary and mating rings move relative to each other, they are affected by the actual face pressure and rotational speed. The product of the two, pressure times velocity, is referred to as PV and is defined as the power N per unit area with a coefficient of friction of unity:

$$PV = \frac{N_f}{a_o}$$

For seals, the equation for PV can be written as follows:

$$PV = P\ V_m = [\Delta P(b - k) + P_{sp}]V_m$$

where V_m = velocity at the mean face diameter d_m, ft/min

Power Consumption. The power consumption of a seal system can be estimated using the PV value and the following equation:

$$N_f = (PV)\ f \cdot a_o \text{ ft. lb/min}$$

where f is the coefficient of friction.

As a general rule, the power to start a seal is five times the running value.

The coefficients of friction for various common seal face materials are given in Table 17-1. These coefficients were developed with water as a lubricant at an operating PV value of 100,000 lb/in.^{2} ft/min. Values in oil would be slightly higher as a result of viscous shear of the fluid film at the seal faces.

Example:

A pump having a 3-inch diameter sleeve at the stuffing box is fitted with a balanced seal of the same size and mean diameter. The seal operates in water at 400 lb/in.^{2}, 3,600 rpm and ambient temperatures. The material of construction for the primary and mating rings are carbon and tungsten carbide. Determine the PV value and power loss of the seal.

Given: $P = 400$ lb/in.^{2}
$b = 0.75$
$k = 0.5$
$d_m = 3$ in.
$P_{sp} = 25$ lb/in.^{2}

$$V_m = \frac{\pi}{12} \times 3 \times 3{,}600 = 2{,}826 \text{ ft/min}$$

$a_o = 1.88$ in.^{2}
$f = 0.07$

$$PV = [400\ (.75 - 0.5) + 25]\ (2826) = 353{,}250\ \frac{\text{lb.}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$$

$$N = (353{,}250)\ (0.07)\ (1.88) = 46487\ \frac{\text{ft. lb}}{\text{min}} = 1.4 \text{ hp.}$$

Temperature Control. At the seal faces, temperature control is desirable because wear is a function of temperature. Heat at the seal faces will cause the liquid film to vaporize, generating still more heat. Thermal distortion may also result, which will contribute to seal leakage. Therefore, many applications will require some type of cooling.

Table 17-1
Coefficient of Friction for
Various Seal Face Materials
(Water Lubrication, PV—100,000 Lb/In. W × Ft/min. John Crane Inc.)

SLIDING MATERIALS		COEFFICIENT OF FRICTION
PRIMARY RING	MATING RING	
CARBON-GRAPHITE (RESIN FILLED)	CAST IRON	0.07
	CERAMIC	0.07
	TUNGSTEN CARBIDE	0.07
	SILICON CARBIDE	0.02
	SILICONIZED CARBON	0.015
SILICON CARBIDE	TUNGSTEN CARBIDE	0.02
	SILICONIZED CARBON	0.05
	SILICON CARBIDE	0.02
	TUNGSTEN CARBIDE	0.08

The temperature of the seal faces is a function of the heat generated by the seal plus the heat gained or lost to the pumpage. The heat generated at the faces from sliding contact is the mechanical power consumption of the seal being transferred into heat. Therefore,

$$Q_s = C_1 N_f = C_1 (PV_f a_o)$$

where Q_s = heat input from the seal Btu/h
$C_1 = 0.077$

If the heat is removed at the same rate it is produced, the temperature will not increase. If the amount of heat removed is less than that generated, the seal face temperature will increase to a point where damage will occur. Estimated values for heat input are given in Figure 17-6 and Figure 17-7.

Heat removal from a single seal is generally accomplished by a seal flush. The seal flush is usually a bypass from the discharge line on the pump or an injection from an external source. The flow rate for cooling can be found by:

$$Gpm = \frac{Q_s}{C_2 \text{ (sp.ht.) (SpGr) } \Delta T}$$

where Q_s = seal heat, Btu/h(W)
$C_2 = 500$
SpHt = specific heat of coolant, Btu/lb°F
SpGr = specific gravity of coolant
ΔT = temperature rise, °F

When handling liquids at elevated temperatures, the heat input from the process must be considered in the calculation of coolant flow. Then

$$Q_{net} = Q_p + Q_s$$

The heat load Q_p from the process can be determined from Figure 17-8.

Example:

Determine the net heat input for a 3-inch diameter balanced seal in water at 3600 rpm. Pressure and temperature are 400 lb/in^2 and 170°F. From Figure 17-7.

$$Q_s = (2750 \text{ Btu/h}/1000 \text{ rpm}) \, 3600 = 9900 \text{ Btu/h}$$

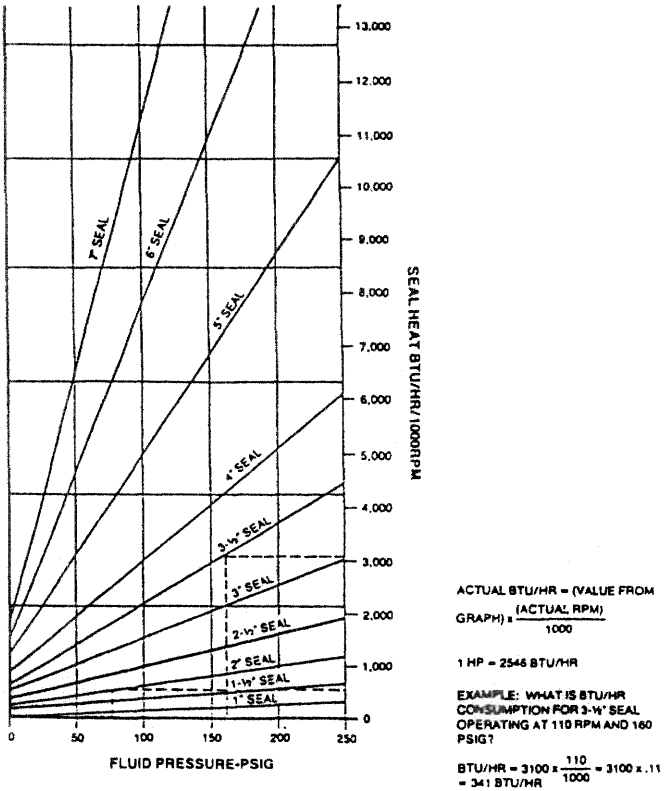


Figure 17-6. Unbalanced seal heat generation (courtesy of John Crane).

From Figure 17-8 assuming that the stuffing box will be cooled to 70°F and that the temperature difference between the stuffing box and pumpage is 100°F

$$Q_p = 255 \text{ Btu/h}$$
$$Q_{net} = 10{,}155 \text{ Btu/hr}$$

The total heat input is used to estimate the required flow of coolant to the seal. For single seal installations, the flow of coolant or seal flush may be achieved through the use of a bypass line from pump discharge. Some installations may require an injection from an external source of liquid. A bypass line can be built internal to the pump or may be constructed externally on the unit. External piping will allow for the installa-

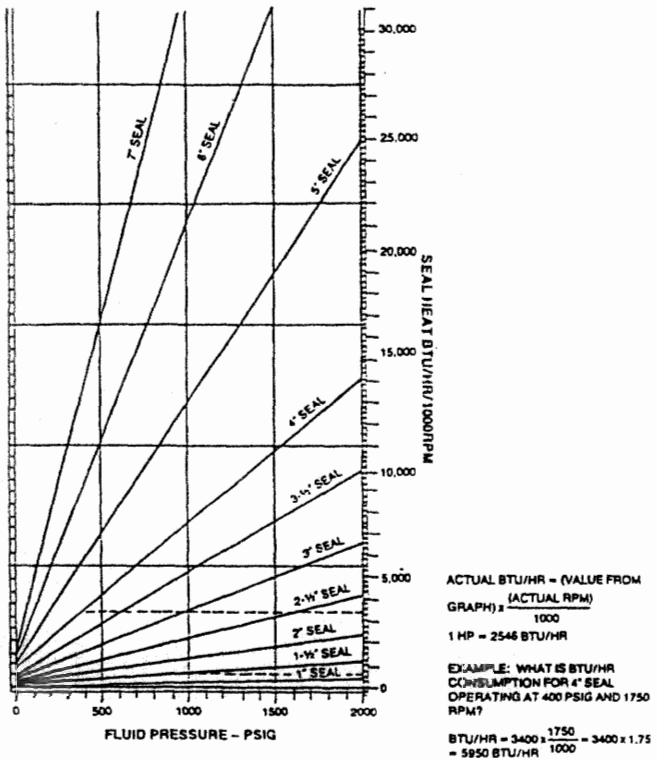


Figure 17-7. Balanced seal heat generation (courtesy of John Crane).

tion of a heat exchanger or abrasive separator when required (Figure 17-9). Another method of supplying additional cooling to a stuffing box is shown in Figure 17-10. This seal is equipped with a pumping ring and heat exchanger. The pumping ring acts like a pump, causing liquid to flow through the stuffing box. Liquid is moved out the top of the stuffing box and through the heat exchanger, returning directly to the faces at an inlet at the bottom of the end plate. As liquid is circulated, heat is removed at the seal and stuffing box. This is an extremely efficient method of removing heat because the coolant circulated is only in the stuffing box and does not reduce the temperature of the liquid in the pump. A closed loop system such as this is commonly used on hot water pumps.

For double seal installations, the heat load for each seal must be considered, as well as the heat soak from the process. The flow of coolant can be achieved through the use of an external circuit with a small pump.

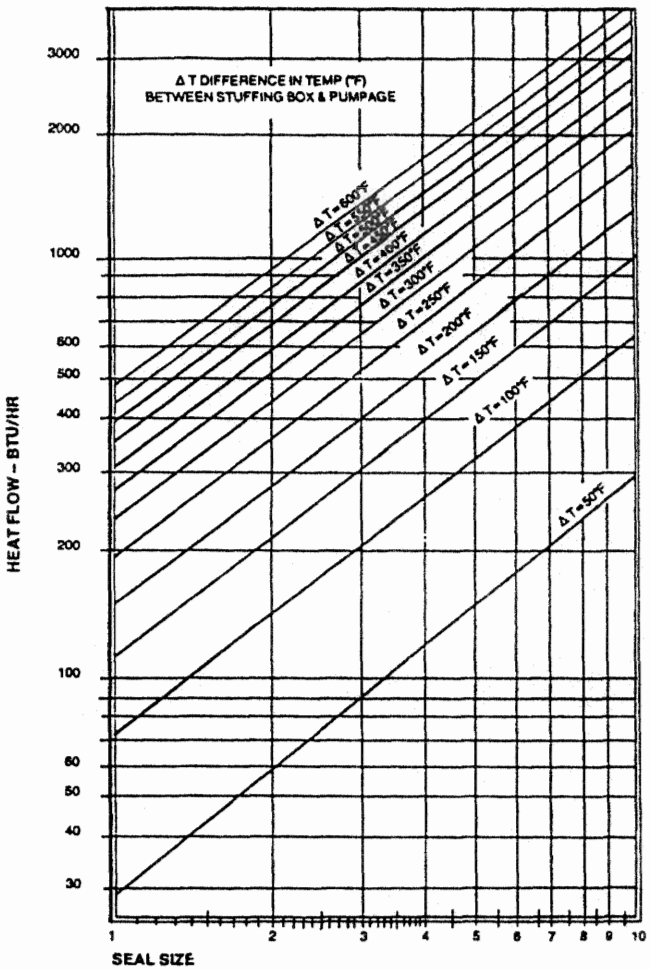


Figure 17-8. Heat soak from the process fluid (courtesy of John Crane).

Pressures in the stuffing box should be kept above the pressure at the inboard seal. As a rule of thumb, the stuffing box is normally 10% or a minimum of 25 psig above the product pressure against the inboard seal. For low pressure installations, a pressurized loop and axial flow pumping ring shown in Figure 17-11 can be considered. Here, as the shaft rotates, the pumping ring moves liquid to a reservoir tank where heat is removed from the system. Liquid returns to the bottom of the stuffing box and continues to remove heat from the seal area.

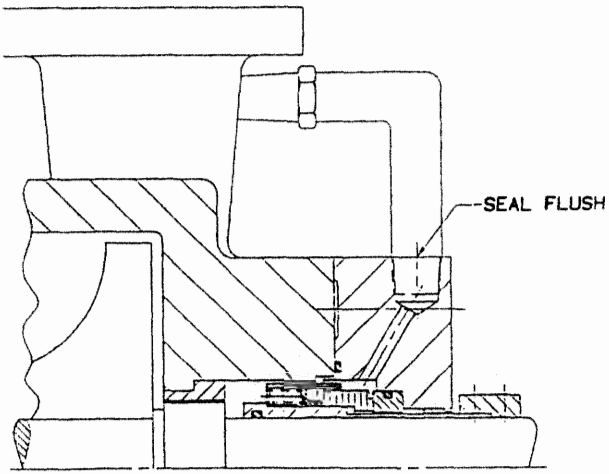


Figure 17-9. Package mechanical seal installation with external piping.

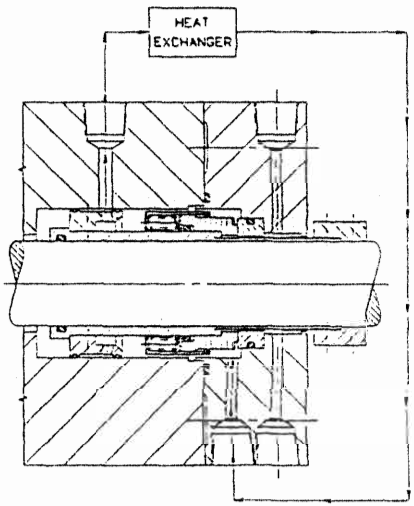


Figure 17-10. Package mechanical seal with pumping ring and heat exchanger closed looped for cooling (courtesy of John Crane).

On tandem seal installations, the inboard seal may be treated as a single seal installation for cooling. However, the outboard seal is normally a closed circuit with a pumping ring (Figure 17-16).

Positive methods for providing clean cool liquid to the seal must be considered at all times. The use of a thermal syphon system for removing heat on a pump application should be avoided.

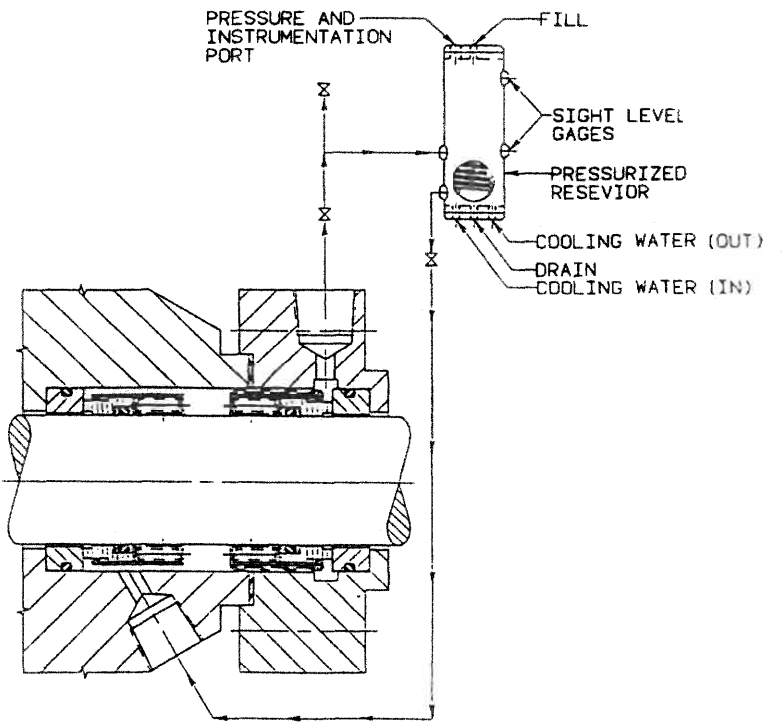


Figure 17-11. Double mechanical seal installation closed loop system with axial flow pumping ring (courtesy of John Crane).

Some special applications after periods of shutdown may require that the process liquid be heated to maintain the product being pumped as a liquid. In these instances, the stuffing box must be preheated prior to starting the pump to avoid damage to the seal.

Seal Leakage

An estimate for seal leakage in cubic centimeters per minute can be made from the following equation:

$$Q = -C_3 \frac{h^3 (P_2 - P_1)}{\mu \ln (R_2/R_1)}$$

where $C_3 = 532$
h = face gap, in
P_2 = pressure at face ID, lb/in.^{2}
P_1 = pressure at face OD, lb/in.^{2}
μ = dynamic viscosity, CP
R_2 = outer face radius, in.
R_1 = inner face radius, in.

Negative flow indicates flow from the face outer diameter to the inner diameter. The effect of centrifugal force from one of the rotating sealing planes is very small and can be neglected in normal pump applications. The gap between the seal faces is a function of the materials of construction, flatness, and the liquid being sealed. The face gap can range from 20×10^{-6} to 50×10^{-6} inches. Leakage from a seal is also affected by the parallelism of the sealing planes, angular misalignment, coning (negative face rotation), thermal distortion (positive face rotation), shaft runout, axial vibration, and fluctuating pressure. Further discussions of these problem areas will be covered later in the chapter on seal installation and troubleshooting.

Seal Wear

The types of wear found in mechanical seals are adhesive, abrasive, corrosive, pitting or fatigue, blistering, impact, and fretting. Adhesive wear is the dominant type of wear in well designed seals. The other types of wear are related to problems with the entire sealing application and will be discussed later in seal installation and troubleshooting.

Even though the successful operation of a seal depends on the development of a lubricating film, solid contact and wear occur during startup, shutdown, and during periods of changing service conditions for the equipment. For most applications, seals are designed with a carbon graphite primary ring considered to be the wearing part in the assembly. The primary ring runs against the mating ring that is made of a harder material and wears to a lesser extent. In mechanical seal designs, face loads are sufficiently low so that only a mild adhesive wear process occurs. If the application is extremely abrasive, hard seal faces for both the primary and mating rings must be considered. Abrasive applications include sand type slurries, paint, and abrasives generated at elevated temperatures from carbonized products.

The PV criterion used in the design of seals is also used in the expression of the limit of mild adhesive wear. Table 17-2 gives the PV limitations for frequently used seal face materials. Each limiting value has been

Table 17-2
Frequently Used Seal Face Materials and Their PV Limitations

SLIDING MATERIALS		PV LIMIT LB/IN2 FT.MIN	COMMENTS
PRIMARY RING	MATING RING		
CARBON GRAPHITE	NI-RESIST	100,000	BETTER THERMAL SHOCK RESISTANCE THAN CERAMIC
	CERAMIC (85% AL203)	100,000	POOR THERMAL SHOCK RESISTANCE. BETTER CORROSION RESISTANCE THAN NI-RESIST
	CERAMIC (99% AL203)	100,000	BETTER CORROSION RESISTANCE THAN 85% (AL203) CERAMIC
	TUNGSTEN CARBIDE (6% CO)	500,000	WITH BRONZE FILLED CARBON GRAPHITE. PV IS UP TO 800,000
	TUNGSTEN CARBIDE (6% NI)	500,000	NI BINDER FOR BETER CORROSION RESISTANCE
	SILICONIZED CARBON	500,000	GOOD WEAR RESISTANCE THIN LAYER OF SI-C MAKES RELAPPING QUESTIONABLE
	SILICON CARBIDE (SOLID)	500,000	BETTER CORROSION RESISTANCE THAN TUNGSTEN CARBIDE
CARBON GRAPHITE		50,000	LOW PV, BUT VERY GOOD AGAINST FACE BLISTERING
CERAMIC		10,000	GOOD SERVICE ON SEALING PAINT PIGMENTS
TUNGSTEN CARBIDE		120,000	PV IS UP TO 185,000 WITH GRADES THAT HAVE DIFFERENT BINDERS
SILICONIZED CARBON		500,000	EXCELLENT ABRASION RESISTANCE, MORE ECONOMICAL THAN SOLID SILICON CARBIDE
SILICON CARBIDE		500,000	EXCELLENT ABRASION AND GOOD CORROSION RESISTANCE. MODERATE THERMAL SHOCK RESISTANCE

developed for a wear rate that provides an equivalent seal life of two years. A PV value for a given application may be compared with the limiting value to determine satisfactory service. These values apply to aqueous solutions at 120°F. For lubricating liquids such as oil, values 60% or higher can be used.

Three separate tests are performed by seal manufacturers to establish performance and acceptability of seal face materials. PV testing is a measure of adhesive wear. Abrasive testing establishes the relative ranking of materials in a controlled environment. Operating temperatures have a significant influence on wear. A hot water test is used to evaluate the behavior of seal face materials at temperatures above the atmospheric boiling point of the liquid.

Classification of Seals by Arrangement

Seal arrangement is used to describe the design of a particular seal installation and number of seals used on a pump. Sealing arrangements may be classified into two groups:

- Single seal installations
  - Internally mounted
  - Externally mounted
- Multiple seal installations
  - Double seals
    - Back to back
    - Opposed
  - Tandem

Single seals are commonly used on most applications. This is the simplest seal arrangement with the least number of parts. An installation mounted inside the stuffing box chamber is referred to as an inside mounted seal (Figure 17-12). Here, liquid in the stuffing box and under pressure acts with the spring load to keep the faces in contact.

An outside mounted seal refers to a seal mounted outside the stuffing box (Figure 17-13). Here, if the seal is not balanced for pressure at the inside diameter of the seal face, the pressure will try to open the seal. Outside mounted seals are considered only for low pressure applications. The purpose of an external seal installation is to minimize the effects of corrosion that might occur if the metal parts of the seal were directly exposed to the liquid being sealed.

Multiple seals are used in applications requiring:

- A neutral liquid for lubrication
- Improved corrosion resistance
- A buffered area for plant safety

Double seals consist of two seal heads mounted back to back with the carbon primary rings facing in opposite directions in the stuffing box chamber (Figure 17-14). A neutral liquid with good lubricating properties, at higher pressures than the pumpage, is used to cool and lubricate the seal faces. The inboard seal keeps the liquid being pumped from entering the stuffing box. Both the inboard and outboard seals prevent the loss of the neutral lubricating liquid. Circulation for cooling is normally achieved with an external circulation system and smaller pump. An axial flow pumping ring and closed system may be considered on low pressure systems.

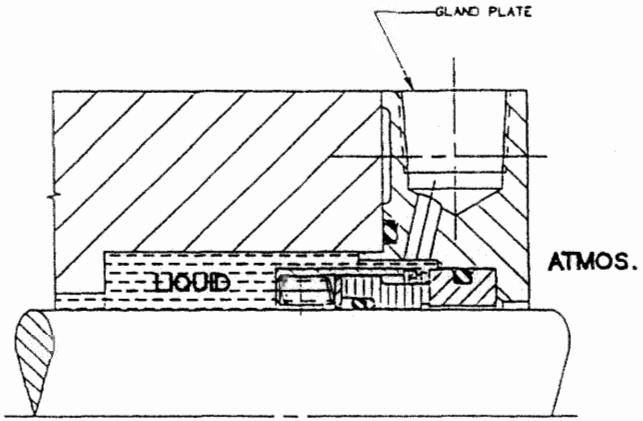


Figure 17-12. Single seal mounted inside the stuffing box.

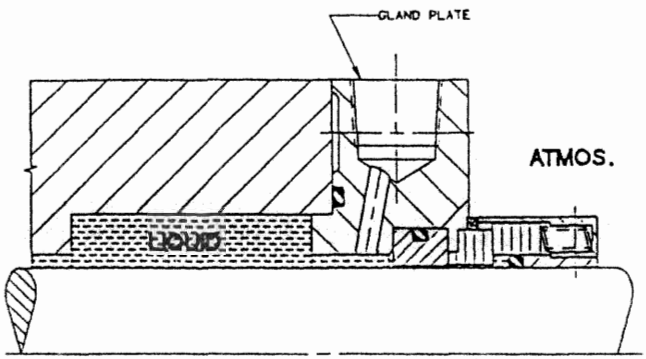


Figure 17-13. Single seal mounted outside the stuffing box. For low pressure service.

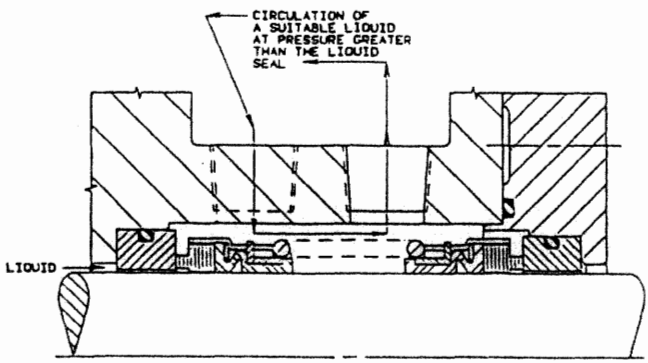


Figure 17-14. Double seal installation.

Double seals may be used in an opposed arrangement (Figure 17-15). Two seals are mounted face to face with the primary sealing rings running on mating rings supported by a common end plate. The neutral liquid is circulated between the seals at the inside diameter of the seal faces. The circulation pressure is normally less than the process liquid being sealed. The inboard seal is similar to a single inside mounted seal and carries the full differential pressure of the pump stuffing box to the neutral liquid. The outboard seal carries only the pressure of the neutral liquid to atmosphere. Both inboard and outboard primary rings are balanced to handle pressure at either the outside or inside diameter of the seal faces without opening. The purpose of this arrangement is to fit a stuffing box having a smaller confined space than what is possible with back-to-back double seals. Double seals are normally applied to toxic liquids for plant safety.

Tandem seals are an arrangement of two single seals mounted in the same direction (Figure 17-16). The inboard seal carries the full pressure differential of the process liquid to atmosphere. The outboard seal contains a neutral liquid and creates a buffered zone between the inboard seal and plant atmosphere. Normally the neutral lubricating liquid is maintained at atmospheric pressure. Developed heat at the inboard seal is removed with a seal flush similar to a single seal installation. The liquid in the outboard seal chamber should be circulated with a pumping ring to remove unwanted seal heat. Tandem seals are used on toxic or flammable liquids that require a buffered area or safety zone in the seal installation.

Package seals are an extension of other seal arrangements. A package seal requires no special measurements prior to seal installation. A package seal assembly consists of a seal head assembly, mating ring assembly, and adaptive hardware. Adaptive hardware are those structural parts that fit a specific seal to a pump. These hardware items are a gland plate, sleeve, and drive collar. A spacer or spacer clips are provided on most package seals to properly set the seal faces. The spacer or spacer clips are removed after the drive collar has been locked to the shaft and the gland plate bolted to the pump. Examples of package seals are shown in Figures 17-9, 17-10, and 17-15.

Classification of Seals by Design

Certain design features are considered important and may be used to describe a seal. These descriptions also form four classification groups. A seal may be referred to as:

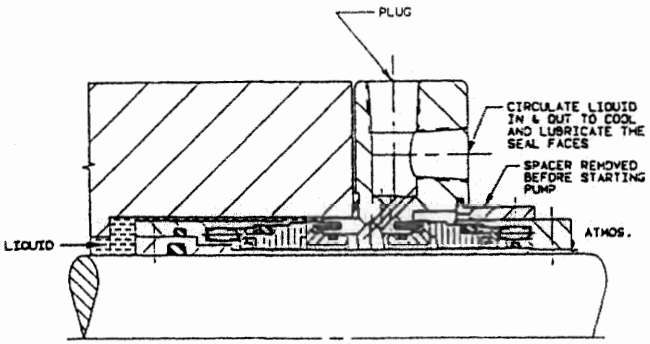


Figure 17-15. Package opposed double seals.

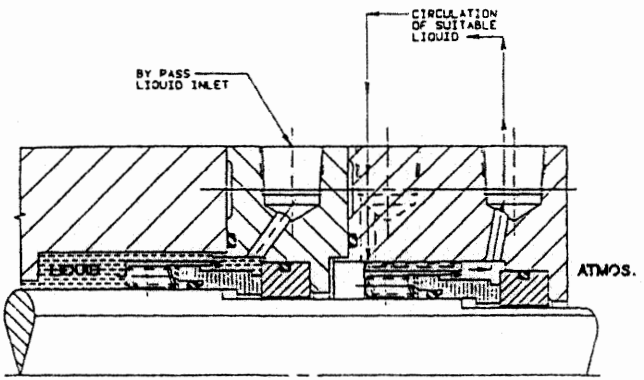


Figure 17-16. Tandem seal installation.

- Unbalanced or balanced
- Rotating or stationary seal head
- Single spring or multiple spring construction
- Pusher or non-pusher secondary seal design

The selection of an unbalanced or balanced seal is determined by the pressure in the pump stuffing box and the type of liquid to be sealed. Balance is a way of controlling the contact pressure between the seal faces and power loss at the seal. When the percentage of balance b (ratio of hydraulic closing area to seal face area) is 100 or greater, the seal is referred to as unbalanced. When the percentage of balance for a seal is

less than 100, the seal is referred to as balanced. Figure 17-17 illustrates common balanced and unbalanced seals.

The selection of a rotating or stationary seal is determined by the speed of the pump shaft. A seal that rotates with a pump with the shaft is a rotating seal assembly. Examples are shown in Figures 17-15 and 17-16. When the mating ring rotates with the shaft the seal is stationary (Figure 17-18). Rotating seal heads are common in industry for normal pump shaft speeds and where stuffing box space is limited. As a rule of thumb, when the shaft speed exceeds 5,000 ft/min, stationary seals are required. Higher speed applications require a rotating mating ring to keep unbalanced forces that may result in seal vibration to a minimum. Also, for certain applications like vacuum tower bottom pumps, a stationary seal allows a steam quench to be applied to the entire inside diameter of the seal to prevent hangup. Stationary seals require a large cross section stuffing box.

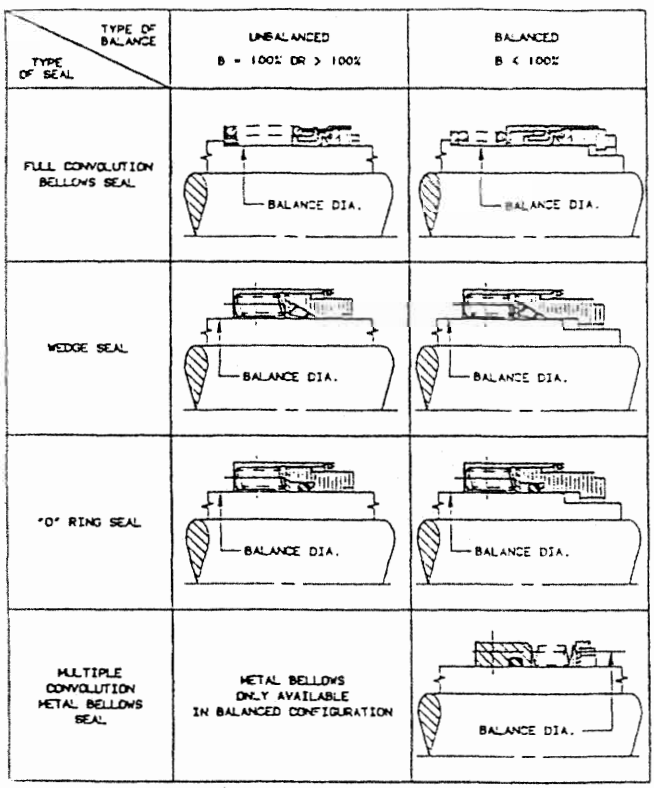


Figure 17-17. Common unbalanced and balanced seals.

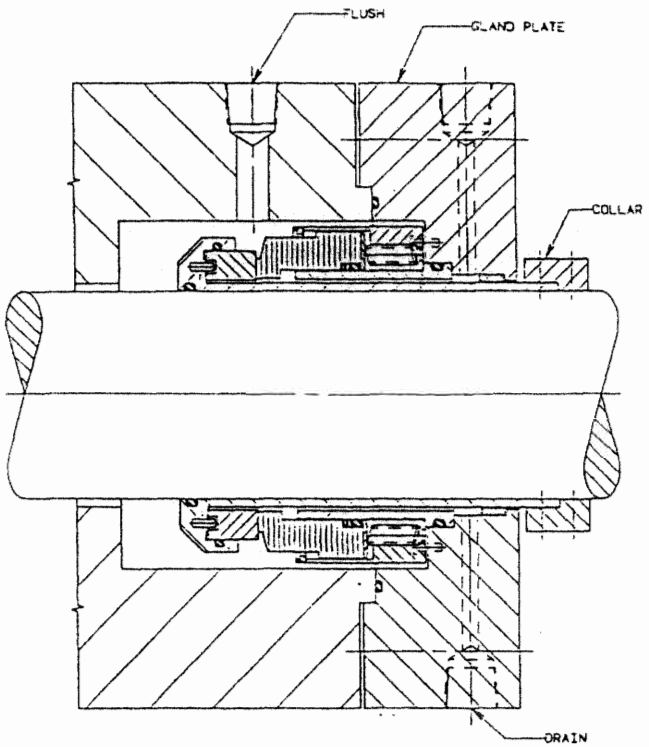


Figure 17-18. Stationary seal head rotating mating ring assembly (courtesy of John Crane).

The selection of a single spring or multiple seal head construction is determined by the space limits available and the liquid sealed. Single spring construction is most often used with elastomeric bellows seals to load the seal faces (Figure 17-19). The advantage of this type of construction is that the openness of the design makes the spring a non-clogging component of the seal assembly. The coils are made of a large diameter spring wire and, as a result, can withstand a great degree of corrosion.

Multiple spring seals require a shorter axial space. Face loading is accomplished by a combination of springs placed about the circumference of the shaft (Figure 17-12). Most multiple spring designs are used with assemblies having O-rings or wedges as secondary seals.

Pusher-type seals are defined as seal assemblies in which the secondary seal is moved along the shaft by the mechanical load of the seal and the hydraulic pressure in the stuffing box. This designation applies to

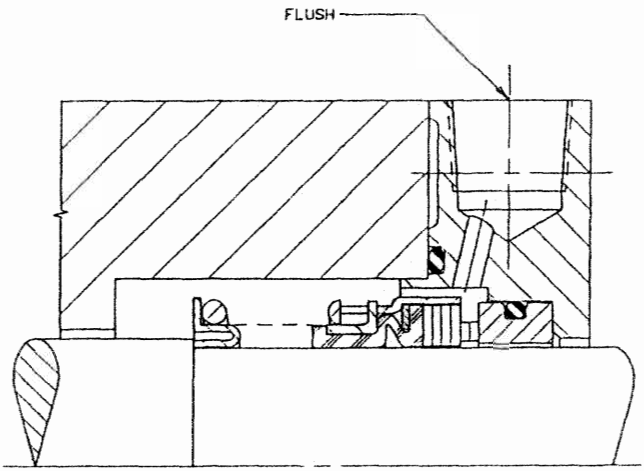


Figure 17-19. Full convolution elastomeric bellows with single spring construction (courtesy of John Crane).

seals that use an O-ring, wedge, or V-ring. Typical construction is illustrated in Figure 17-12.

Non-pusher seals are defined as seal assemblies in which the secondary seal is not forced along the shaft by the mechanical load or hydraulic pressure in the stuffing box. Instead, all movement is taken up by the bellows convolution. A non-pusher design is shown in Figure 17-19. This definition applies to those seals that use half, full, and multiple convolution bellows as a secondary seal.

Materials of Construction

The selection of materials of construction must be based on the operating environment for the seal. The effects of corrosion, temperature, deformation from pressure, and wear from sliding contact must be considered for good life. Each seal must be broken down into component parts for material selection. The effects of corrosion must be known for the secondary seal, primary and mating rings, as well as hardware items. The NACE (National Association of Corrosion Engineers) corrosion handbook is an excellent source of information for corrosion rates of various materials in a wide variety of liquids and gases. When hardware items of a seal are exposed to a liquid and the corrosion rate is greater than 2 mils per year, a double seal arrangement should be considered. The metal parts in this seal arrangement can be kept out of the corrosive

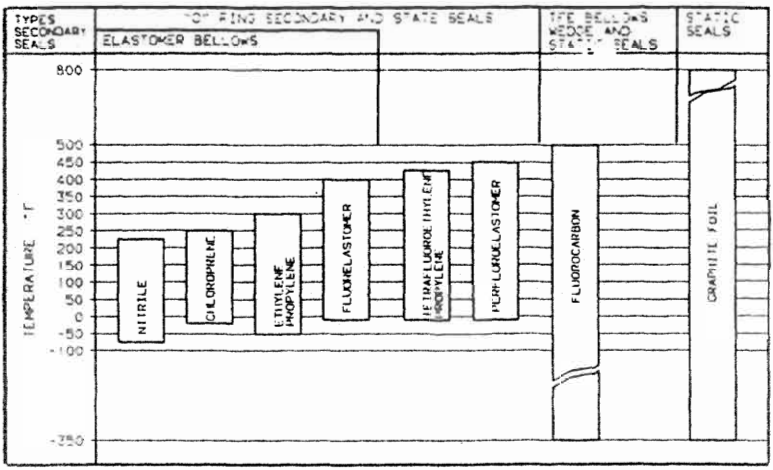
liquid. Then only the seal faces (primary and mating rings) and secondary seal come in contact with the pumpage and can be made of corrosion resistant materials such as Teflon, ceramic, and carbon. Some of the more common seal materials are shown in Table 17-3.

Temperature is another consideration in seal material selection. Both secondary and static seals must remain flexible throughout the entire life of the seal. If seal flexibility is lost by hardening of the elastomer from high temperature, the seal will lose the degree of freedom necessary to follow the mating ring. The temperature limits for common secondary and static seal materials is given in Table 17-4. Temperature is not an issue for a metal bellows seal. This type of secondary seal is specifically designed for high temperature and cryogenic applications beyond the limits of common elastomeric materials. However, this type of seal is limited in terms of corrosion resistance and most high pressure applications due to the thin construction of the bellows.

Table 17-3
Most Common Materials of Construction for Mechanical Seals

COMPONENT	MATERIALS OF CONSTRUCTION
SECONDARY SEALS	
O-RING	NITRILE, ETHYLENE PROPYLENE, FLUOROELASTOMER, PERFLUOROELASTOMER, CHLOROPRENE
BELLOWS	NITRILE, ETHYLENE PROPYLENE, FLUOROELASTOMER, CHLOROPRENE
WEDGE	FLUOROCARBON RESIN, GRAPHITE FOIL
METAL BELLOWS	STAINLESS STEEL, NICKEL BASE ALLOY
PRIMARY RING	CARBON, METAL FILLED CARBON, TUNGSTEN CARBIDE, SILICON CARBIDE, SILICONIZED CARBON
HARDWARE (RETAINER, DISC, SNAP RING, SET SCREWS, SPRINGS)	18-8 STAINLESS STEEL, 316 STAINLESS STEEL, NICKEL BASE ALLOYS, TITANIUM
MATING RING	CERAMIC, CAST IRON, TUNGSTEN CARBIDE, SILICON CARBIDE

Table 17-4
Temperature Limitations for Common Secondary and Static Seals



An additional consideration in the selection of the primary and mating rings is their PV limitations. As discussed in the section on seal wear, a PV value can be determined for each application and compared to those given in Table 17-2. A value of less than the value given in the table will result in a seal life greater than two years. Also shown in Table 17-5 are the physical properties of common seal face materials.

Mating Ring Designs

A complete seal installation consists of the seal head assembly and a mating ring assembly. The mating ring assembly consists of the mating ring and a static seal. There are five common mating ring types: a) groove O-ring, b) square section, c) cup mounted, d) floating, and e) clamped in (Figure 17-20). Other designs may exist but they are all variations of these common types. The purpose of a mating ring is to provide a hard surface for the softer member of the seal face to run against. Because wear will occur at the seal faces, the mating ring must be designed as a replaceable part in the seal installation. This is accomplished through the use of a static seal, as shown. The material of construction of the static seal will determine the temperature limit of the assembly.

Table 17-5
Properties of Common Seal Face Materials

PROPERTY	CAST IRON	NI RESIST	CERAMIC		CARBIDE		CARBON			
			85% (AL203)	99% (AL203)	TUNGSTEN (6% NI)	SILICON (SI-C)	RESIN	BABBITT	BRONZE	SILICONIZED CARBON
MODULUS OF ELASTICITY X 10E6 LB/IN2	13 - 16	10.5 - 16.9	32	50	90	48 - 57	2.5 - 4.0	1.04 - 4.1	2.9 - 4.4	2 2.3
TENSILE STRENGTH X 10E3 LB/IN2	65 - 120	20 - 45	20	39	123.25	20.65	8 - 8.6	8 - 8.6	7.5 - 9	2
CO-EFFICIENT OF THERMAL EXPANSION X 10E-6 IN/IN F	6.6	6.5 - 6.8	3.9	4.3	2.66	1.08	2.1 - 2.7	2.1 - 2.7	2.4 - 3.1	2.4 - 3.2
THERMAL CONDUCTIVITY BTU.FT/H X FT2 F	23 - 29	25 - 28	8.5	14.5	41 - 48	41 - 60	6 - 9	6 - 9	8 - 8.5	30
DENSITY LB/IN3	0.259 - 0.268	0.264 - 0.268	0.123	0.137	0.59	0.104	0.064 - 0.069	0.083 - 0.112	0.083 - 0.097	0.06 - 0.070
HARDNESS	-----BRINELL-----		--------ROCKWELL A----------			----ROCKWELL 45N---		-------SHORE-------		ROCKWELL 15T
	217 - 269	131 - 183	87	87	92	86 - 88	80 - 105	60 - 95	70 - 92	90

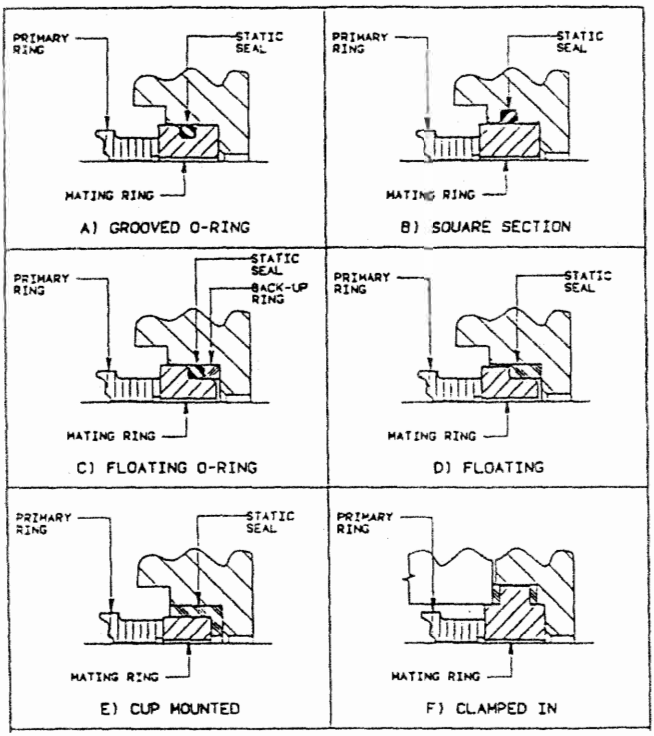


Figure 17-20. Common mating ring assemblies.

Adaptive Hardware

Structural parts of a seal installation such as shaft sleeve, gland plates, bushings, and collars used to fit a seal to a pump are referred to as adaptive hardware. These items may be furnished by the pump company or the seal manufacturer.

The success or failure of a seal installation can often be traced to the proper selection of the gland plate and associated piping arrangement. The purpose of the gland plate or end plate is to hold either the mating assembly or the seal assembly, depending on the type of seal construction. The gland plate is also a pressure containing component of the assembly. The alignment of one of the sealing surfaces, and possibly a bushing, is dependent on the fit of the gland plate to the stuffing box.

For a pump manufactured to API (American Petroleum Institute) specifications, a registered fit of the gland plate is required between the inside

or outside diameter of the stuffing box. This is an industry standard for refinery service. In addition, the gland plate at the stuffing box must completely confine the static seal.

There are three basic gland plate constructions, as shown in Figure 17-21.

A **plain** gland plate is used where seal cooling is provided internally through the pump stuffing box and where the liquid to be sealed is not considered to be hazardous to the plant environment and will not crystallize or carbonize at the atmospheric side of the seal.

A **flush** gland plate is used where internal cooling is not available. Here, coolant (liquid sealed or liquid from an external source) is directed to the seal faces where the heat is generated.

A **flush and quench** gland plate is required on those applications that need direct cooling, as well as a quench fluid at the atmosphere side of the seal. The purpose of a quench fluid, which may be a liquid, gas, or

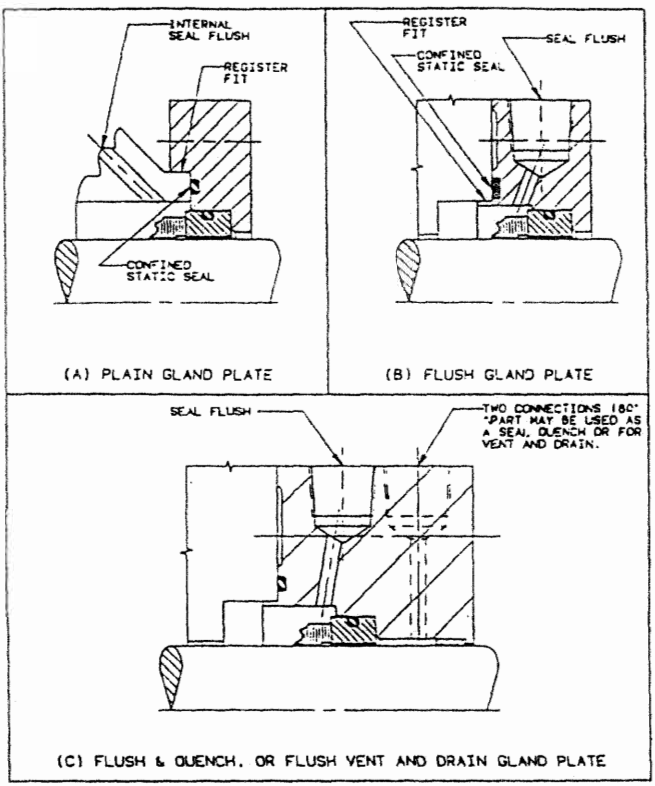


Figure 17-21. Basic gland plate designs.

steam, is to prevent the buildup of any carbonized or crystallized material along the shaft. When properly applied, a seal quench can increase the life of a seal installation by eliminating the loss of seal flexibility due to hangup.

A **flush, vent and drain** gland plate is used where seal leakage needs to be controlled. Flammable vapor from a seal can be vented to a flair and burned off, while non-flammable liquid leakage can be directed to a sump.

Figure 17-22 illustrates some common restrictive devices used in the gland when quench or vent and drain connections are used. These bushings may be pressed in place, as shown in Figure 17-22A, or allowed to float, as in figures 17-22B, C and D. Floating bushings allow for closer running fits with the shaft because such bushings are not restricted at their outside diameter. Floating bushings in Figures 17-22C and D, are

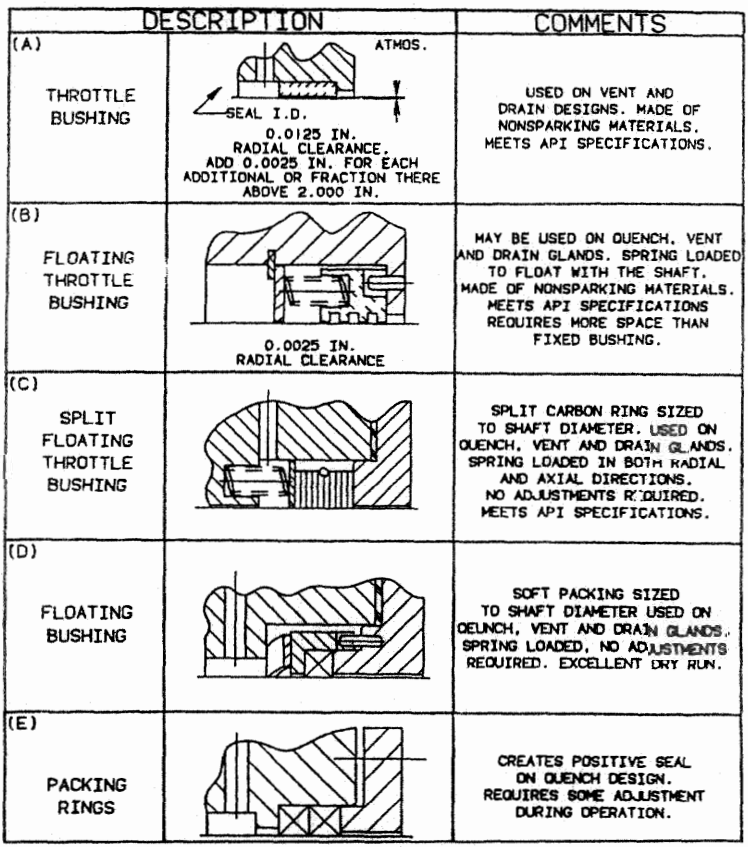


Figure 17-22. Common restrictive devices used with quench or vent-and-drain gland plate.

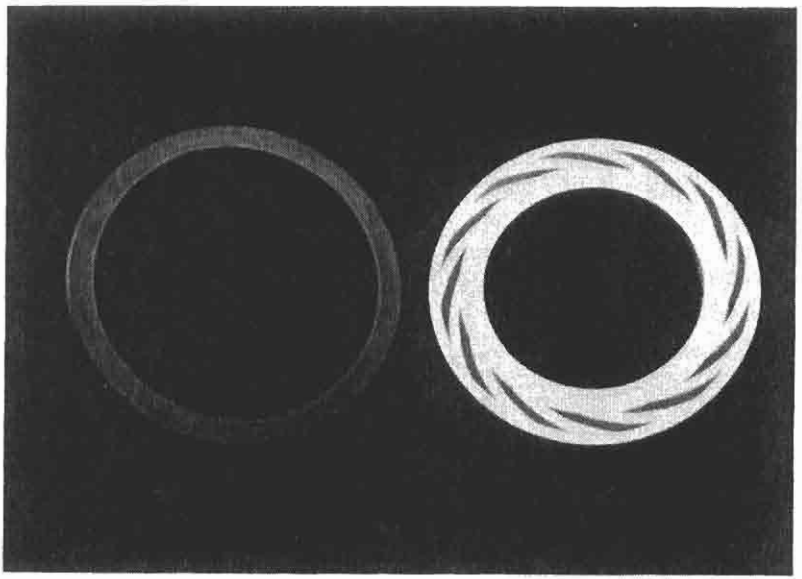


Figure 17-23. Seal face design for upstream pumping (courtesy of John Crane).

also sized to fit the diameter of the shaft. Small packing rings may also be used for a seal quench as shown in Figure 17-22E.

Shaft sleeves should be used on all applications where the shaft must be protected from the environment of the pumpage. Sleeves should be made of corrosion and wear resistant materials. Sleeves must be positively secured to the shaft and seal at one end. The sleeve should extend through the gland plate.

Upstream Pumping

The concept of upstream pumping is new to the sealing industry and is a further refinement of spiral groove technology developed for dry running gas compressor seals. Upstream pumping is defined as moving a small quantity of liquid from the low pressure side of a seal to the high pressure side. This is accomplished by a change in seal face geometry and by incorporating spiral grooves, Figure 17-23.

A conventional flat seal design is only capable of preventing leakage from a higher pressure stuffing box to atmosphere. A seal designed with the upstream pumping feature can seal high pressure and move clean liquid across the seal faces to a high pressure stuffing box. This type of design creates a full liquid film at the faces and reduces horsepower loss. This is also a new way to flush a seal face with just a few cc/minute

rather than a few gpm with a conventional flush. Figure 17-24 illustrates a seal with the upstream pumping concept at the inboard seal. Early experiments demonstrated that this 1.750-inch diameter seal operating in water at 1800 rpm can move 0.6 cc/min to a stuffing box pressure of 800 psig. This concept when applied to multiple seals would allow a plant operator to bring a supply of water or a neutral liquid directly to the space between the seals. The low pressure liquid can be made to flow in small quantities to higher pressures through the pumping action of the seal faces. This improvement in seal design has resulted in a new way to flush a seal, which can be applied to abrasive slurry service, hot water, hazardous liquids, heat transfer, and liquids with poor lubricating properties. This is a significant development resulting in a non-contacting, non-leaking seal for liquid service. The upstream pumping concept is a patented development by John Crane Inc.

Mechanical Seals for Chemical Service

Traditionally, mechanical seals for this type of service are fitted to pumps that meet American National Standards Institute (ANSI) Specification B73.1. This specification defined the stuffing box cross section by the size of the packing that could be used on a given application. Standard sizes of cross section were 5/16, 3/8, and 7/16 inches. The idea behind the specification was that if seals were used and had to be replaced they could be replaced by packing. However, in practice when a seal is worn out or

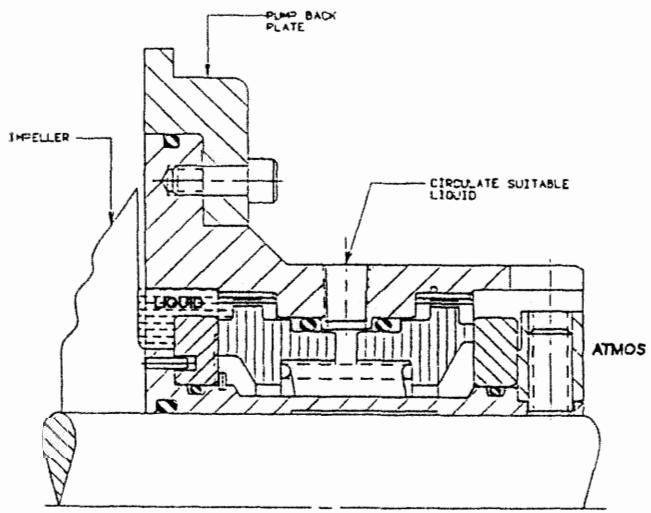


Figure 17-24. Package symmetrical seal with upstream pumping feature (courtesy of John Crane).

failed, it was replaced by another seal. Fitting a seal to packing cross sections was sometimes difficult, particularly when multiple seal arrangements were required. In addition, the smaller the space around the seal, the warmer it would run.

Figure 17-25 shows the original stuffing box and the new revised box that allows more than two times the clearance over the pump shaft. The new box dimensions allow a current ANSI seal to run 25°F cooler than the original design. By opening up the back of the stuffing box and providing a slight taper to the bore, the stuffing box will run approximately 40°F cooler than the original design. This data was developed testing conventional seals that are considered standard in the original stuffing box. These include elastomeric bellows, wedge, and O-ring seals. Stuffing box design plays an important part in providing a good environment for the mechanical seal.

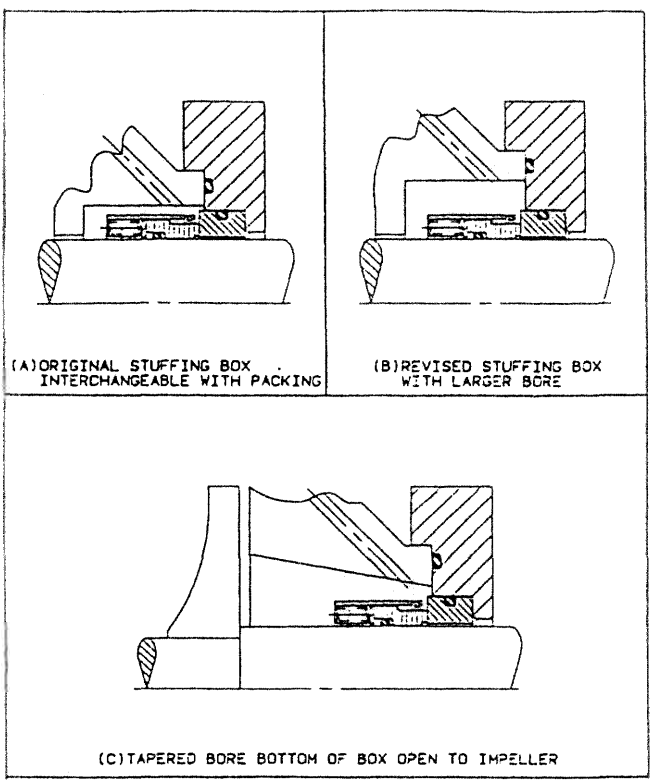


Figure 17-25. Basic stuffing box designs.

Mechanical Seals for Refinery Service

Refinery services can be broken down into two primary classes based on temperature:

- Applications below 350°F
- Applications above 350°F

For applications below 350°F elastomeric seals should be used. These applications may also involve the sealing of high pressure as well. A typical package seal for this class of service is shown in Figure 17-26.

For applications above 350°F, a metal bellows seal should be used. Pressure on most high temperature applications will be low and would not present a problem for most metal bellows available to industry. A typical package seal for this class of service is shown in Figure 17-27. Both seals illustrated in Figures 17-26 and 17-27 meet the requirements of American Petroleum Institute's (API) 610 Standard. This standard does not cover the design of individual seal components. That is the responsibility of the seal manufacturer. The standard does cover the minimum requirements of equipment for use in refinery services. These requirements would include:

- Seal chamber and shaft sleeve considerations.
- Hardness of seal materials to be used on a pump shaft.
- Preferred use of balanced seals unless otherwise specified.
- Containment of seal face leakage by one of the following arrangements:
  Single seal with an auxiliary sealing device such as a floating close clearance bushing or other suitable device.
  Tandem seal arrangement with a barrier fluid maintained at a pressure lower than seal chamber pressure.
  Double seal arrangement with a barrier fluid maintained at a pressure greater than seal chamber pressure.
- Gland plate design including register fit to the seal chamber and confinement of static seals.

Materials of construction are identified as well as piping plans for seal installations. Piping plans for seal installations are so important that they are shown in Figures 17-28A and B for easy reference. API Specification 610 represents the collective knowledge and experience of end users, pump and seal manufacturers. This specification is updated periodically, and anyone involved in refinery equipment should have an updated copy of the specification from the American Petroleum Institute.

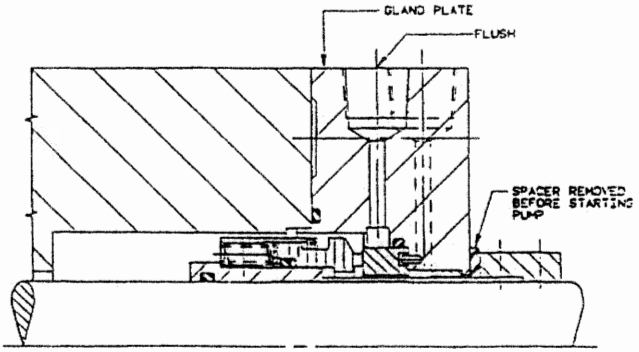


Figure 17-26. Package seal for refinery service for temperatures below 350°F (courtesy of John Crane, Inc.).

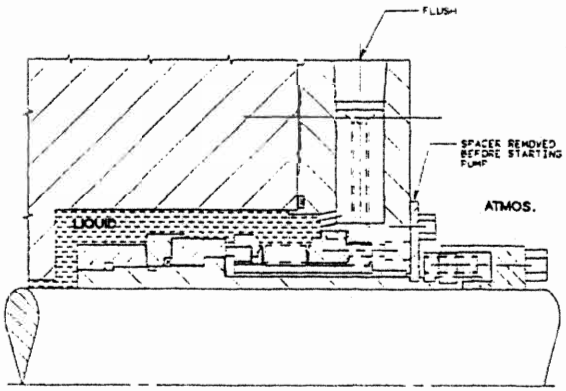


Figure 17-27. Package metal bellows seal (courtesy of John Crane).

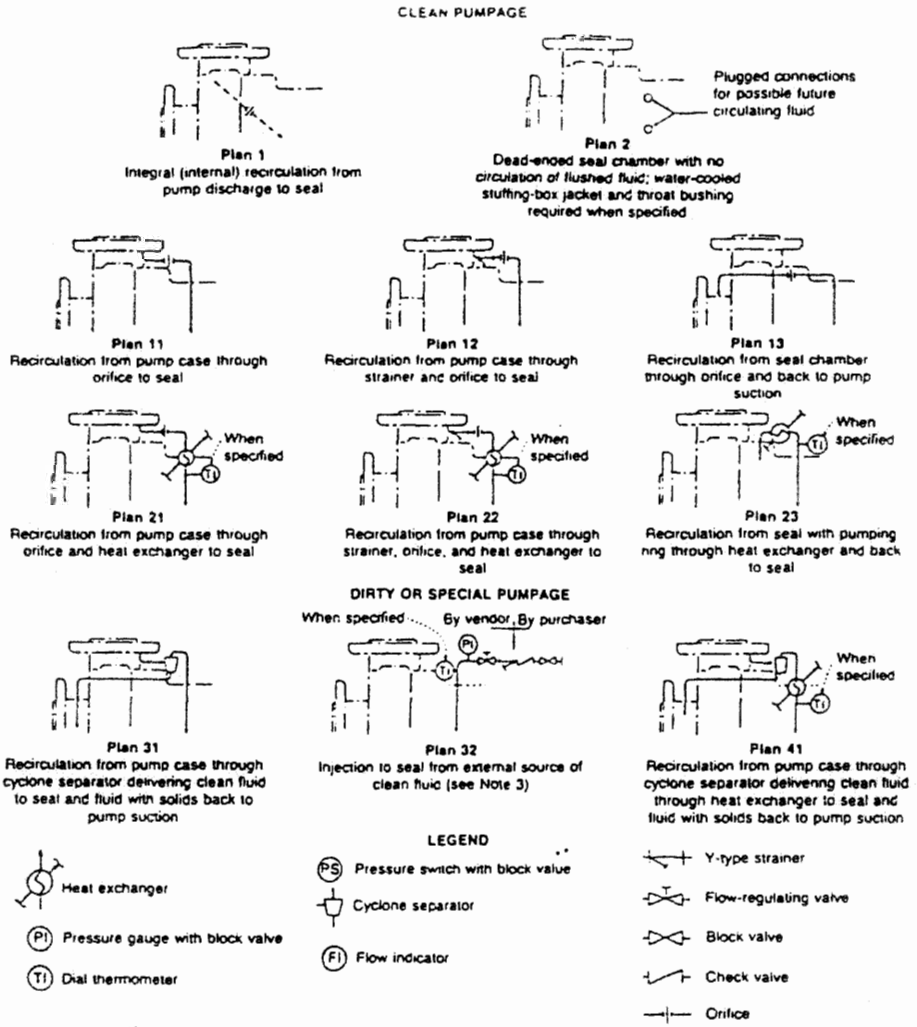
Typical Applications

Mechanical seals are used to seal a wide range of liquids and gases at various conditions of pressure, temperature, and speed. As seal technology advances, progress is being made to move from the traditional contacting seal to what is becoming known as a class of seals referred to as non-contacting, non-leaking. Certain types of liquids will require the seal to operate without any appreciable power loss or wear at the seal faces.

Light Hydrocarbon Service

When sealing a light hydrocarbon as a liquid, conventional flat sealing faces can be considered when the operating conditions do not require the

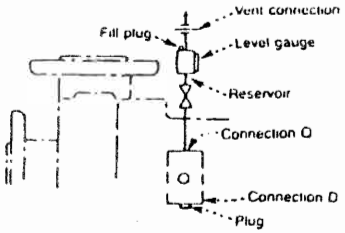
seal to work in an environment where the liquid in the seal chamber is near its vapor pressure. As a rule of thumb, when the stuffing box pressure is less than one-third of the vapor pressure, a seal face design with hydropads should be considered. Hydropads or lubrication recesses are



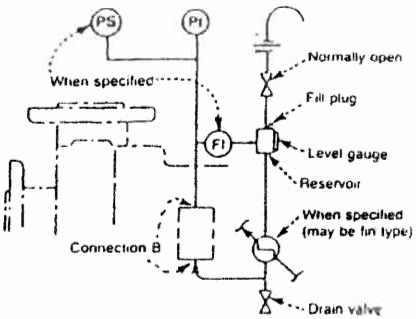
Notes:
1. These plans represent commonly used systems. Other variations and systems are available and should be specified in detail by the purchaser or mutually agreed upon by the purchaser and the vendor.
2. These plans are for use with Connection F on the single- and tandem-seal arrangements shown in Figure D-1.
3. For Plan 32, the purchaser will specify the fluid characteristics, and the vendor shall specify the volume, pressure, and temperature required.

Figure 17-28A. Piping plans for mechanical seals from A.P.I. Standard 610, 6th Edition. (Reproduced courtesy American Petroleum Institute.)

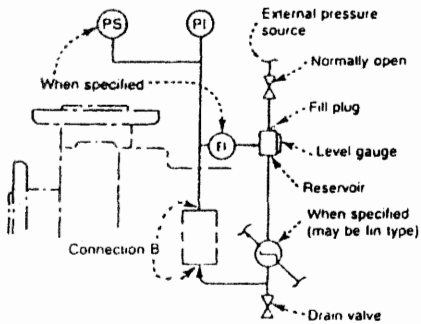
an effective way of reducing the power loss of a flat sealing face by as much as 30%. The hydropad design feature is shown in Figure 17-29. In most applications this will permit the seal to operate in a very stable condition with very low leakage to the atmosphere. Leakage rates from these



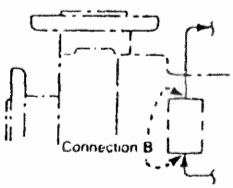
Plan 51
Dead-ended blanket (usually methanol, see Note 3); typically used with auxiliary sealing device (single- or double-seal arrangement in Figure D-1)



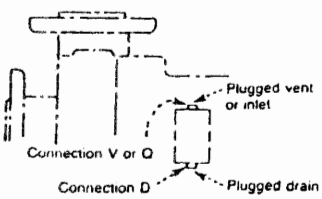
Plan 52
Nonpressurized external fluid reservoir (see Note 3) with forced circulation, typically used with tandem-seal arrangement in Figure D-1



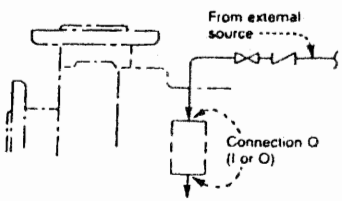
Plan 53
Pressurized external fluid reservoir (see Note 3) with forced circulation; typically used with double-seal arrangement in Figure D-1



Plan 54
Circulation of clean fluid from external system (see Note 3); typically used with double-seal arrangement in Figure D-1



Plan 61
Tapped connections for purchaser's use; Note 3 applies when purchaser is to supply fluid (steam, gas, water, etc.) to auxiliary sealing device (single- or double-seal arrangement in Figure D-1)



Plan 62
External fluid quench (steam, gas, water, etc.; see Note 3); typically used with throttle bushing or auxiliary sealing device (single- or double-seal arrangement in Figure D-1)

Notes:
1. These plans represent commonly used systems. Other variations and systems are available and should be specified in detail by the purchaser or mutually agreed upon by the purchaser and the vendor.
2. See Figure D-2 for explanation of symbols not specified here.
3. When supplemental seal fluid is provided, the purchaser will specify the fluid characteristics. The vendor shall specify the volume, pressure, and temperature required, where these are factors.

Figure 17-28B. Piping plans for mechanical seals from A.P.I. Standard 610, 6th Edition. (Reproduced courtesy American Petroleum Institute.)

Figure 17-29. Carbon primary ring with hydropad face design (courtesy of John Crane).

types of seals are measured in parts per million. The seal illustrated in Figure 17-30 has been successfully applied to sealing light hydrocarbon at the following conditions:

- Liquid sealed: ethylene, ethane, or propane
- Stuffing box pressure: 500 to 1200 psig
- Temperature: 30 to 60°F
- Specific gravity: 0.3 to 0.9
- Speed: 1500 to 3480 rpm
- Seal size: 5.375 inches

Many installations have been made and thousands of hours of seal life have been realized. The materials of construction for the seal faces are carbon versus tungsten carbide.

Hydropads have also been successfully applied to high temperature applications requiring a metal bellows seal. The seal shown in Figure 17-31 is a typical installation that has been in service for more than three years. This was a difficult application for the original seals that were specified.

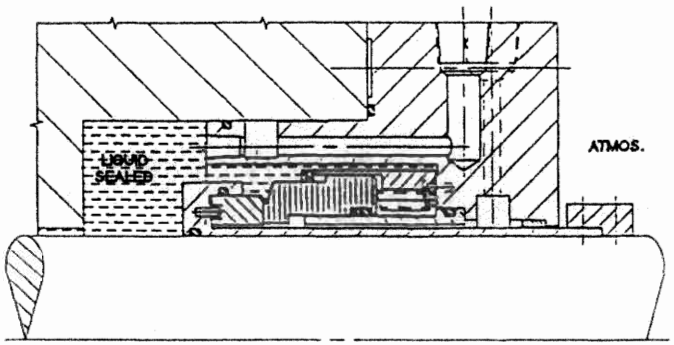


Figure 17-30. Package mechanical seal installation for a pipeline pump (courtesy of John Crane).

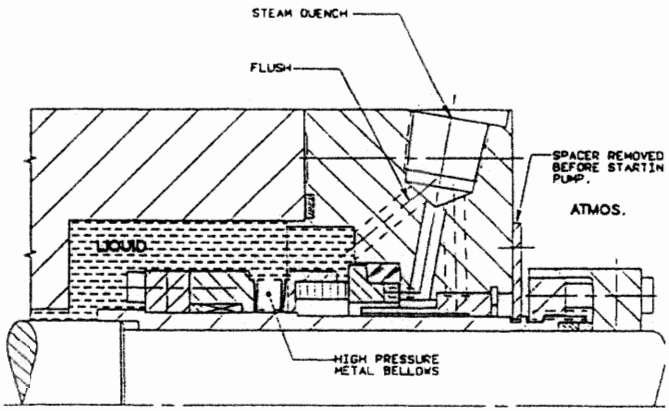


Figure 17-31. Mechanical seal for a high temperature refinery charge pump (courtesy of John Crane).

Failure occurred due to vaporization at the seal faces. The seal shown has been through many startups with no adverse effects. The operating conditions for this application are:

- Liquid sealed: gas oil
- Stuffing box pressure: 278 psig
- Temperature: 615 °F
- Specific gravity: 0.665
- Speed: 3550 rpm
- Seal size: 3.125 inches

The measured leakage from each end of this horizontal pump was 10 to 120 ppm. This installation is a rotating seal head with a tungsten carbide insert running on a silicon carbide mating ring. A steam quench is being used on the atmospheric side of the seal.

Hydropads are also used to aid the lubrication process on the refinery application shown in Figure 17-32. The pump is used in a hydrofluoric alkylation unit to move hydrofluoric acid. To protect the sealing system, a clean injection of isobutane is flushed in at the seal faces. The original seals experienced short seal life due to chemical attack from the hydrofluoric acid. The operating conditions for this application are:

- Liquid sealed: isobutane/hydrofluoric acid
- Stuffing box pressure: 240 psig
- Temperature: 89°F
- Specific gravity: 0.945
- Speed: 1180 rpm
- Seal size: 3.5 inches

This hydropadded seal has added years to the service life of the unit. The materials of construction for the seal faces are solid silicon carbide running on solid silicon carbide. This material combination was selected for its excellent corrosion resistance to hydrofluoric acid, because corrosion is a common cause for seal failure on these types of applications.

Hydropads or lubrication recesses when properly applied will result in improved seal life. This design feature is the first step in the process of advancement to non-contacting seals.

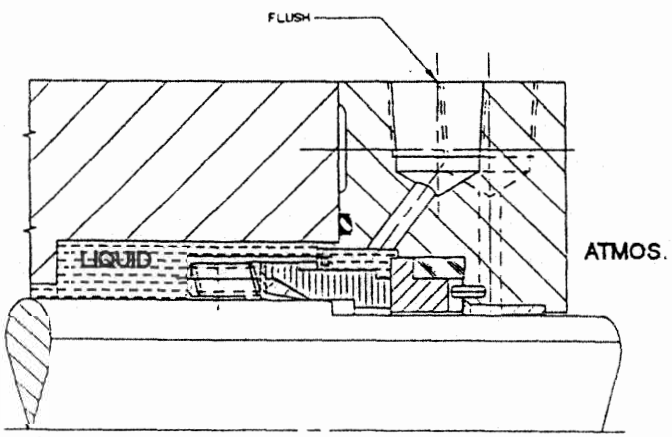


Figure 17-32. Mechanical seal installation for refinery HF Alkylation Unit (courtesy John Crane).

True non-contacting seals are being applied to pumping equipment and will change the way a seal is viewed for a given application. When pumping a light hydrocarbon near its vapor pressure, it is more efficient to allow the liquid to flash to a gas in the stuffing box and use a dry running seal to seal the gas. This is done with a non-contacting seal design. This most efficient method in achieving non-contact in operation is through the use of the spiral groove face seal. This patented concept was developed by John Crane Inc. for use in sealing high performance gas compressors and is illustrated in Figure 17-33. The spiral groove pattern is a series of logarithmic spirals recessed into the hard mating ring. The ungrooved portion of the seal face below the spiral is called the sealing dam. The spiral groove pattern is designed to rotate only in one direction. As the seal face begins to rotate from the outside diameter of the seal faces to the groove diameter, gas is compressed and then expanded across the sealing dam. This will generate sufficient opening force to separate the seal faces by a few micro-inches. Seal balance, face width and spiral groove diameters are all critical and determine the static and dynamic operation of the seal during periods of startup and shutdown when the seal faces contact. A comparison of hydropadded and spiral groove faces is made in Figure 17-34.

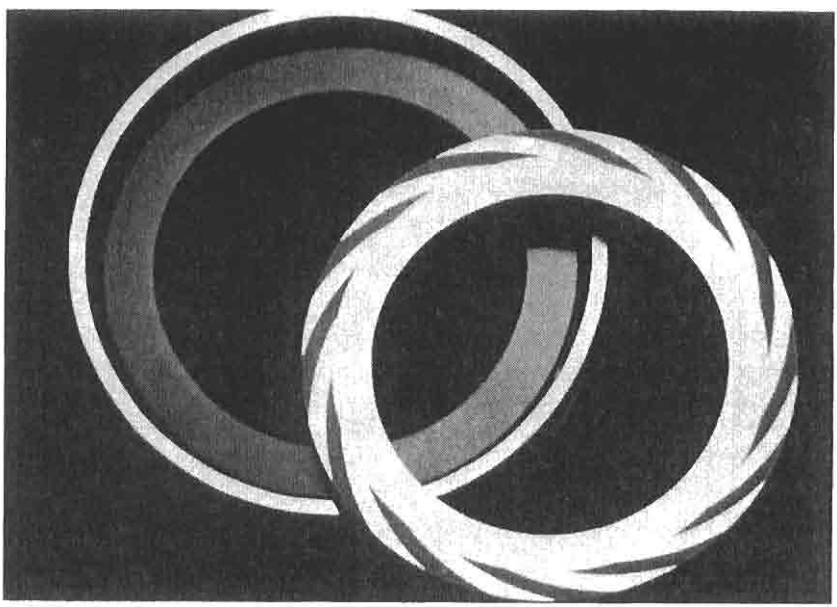


Figure 17-33. Dry running face design (courtesy of John Crane).

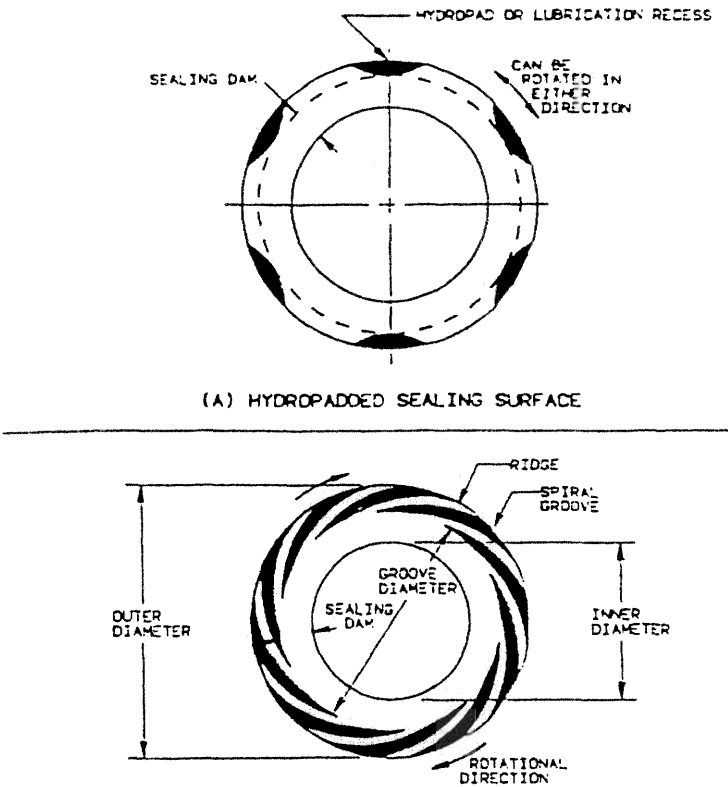


Figure 17-34. *Comparison of seal face geometry.*

A typical pump application is shown in Figure 17-35. The operating conditions for this seal are:

- Liquid sealed: light hydrocarbon flashed to gas in the stuffing box
- Stuffing box pressure: 387 psig
- Vapor pressure at pumping temperature: 375 psig
- Temperature: 31°F
- Specific gravity: 0.4
- Speed: 1500 rpm
- Seal size: 4.687 inches

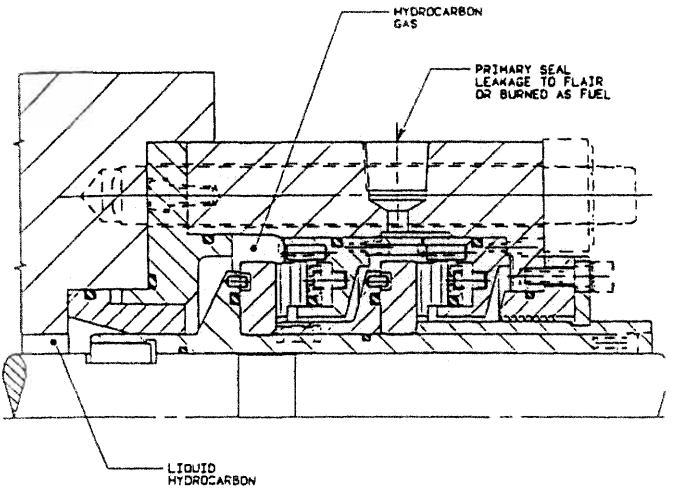


Figure 17-35. Package gas seal (courtesy of John Crane).

A bushing is used at the bottom of the box, and the seal chamber at the inboard seal is essentially dead ended. The only flow of hydrocarbon gas in this area is due to the leakage across the inboard seal face. Normal leak rates are in the range of 1.5 scfm. The space between the seal is vented to a flair or burned as fuel. The outboard seal prevents the gas from escaping to atmosphere. Since this is a non-contacting seal the life of the seal will not be affected by wear.

Some applications still require double seals to provide a good running environment for the seal. Figure 17-36 shows a solution to sealing high pressure liquid carbon dioxide service. These seals are used on pumping equipment used to inject the liquid carbon dioxide into the ground for crude oil extraction. On this application, oil is circulated through the seal chamber at a pressure 10% greater than the pressure at the inboard seal from the liquid carbon dioxide. This seal is operating at the following conditions.

- Liquid sealed: carbon dioxide
- Buffer liquid: oil
- Stuffing box pressure: 2400 psig
- Temperature: 65°F
- Specific gravity: 0.76 (oil)
- Speed: 3550 rpm
- Seal size: 3.750 inches

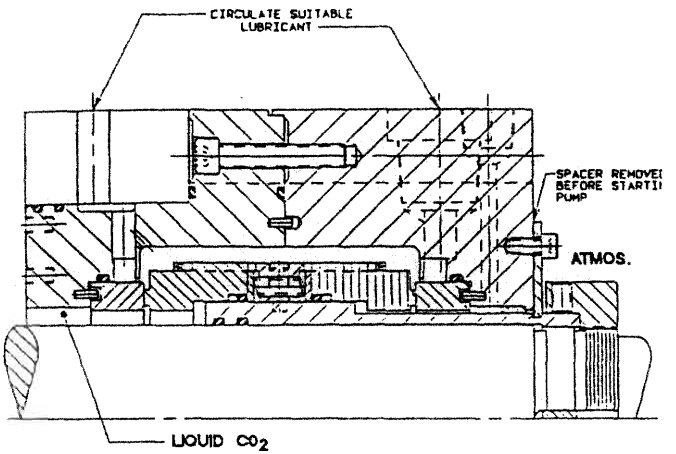


Figure 17-36. Package mechanical seal for CO_2 service (courtesy of John Crane).

The inboard seal faces are a metal alloy running against tungsten carbide. The choice of this material combination was to avoid blistering in a carbon material that would be exposed to the liquid carbon dioxide. The seal faces at the outboard position are running at a higher level of PV and here, carbon versus tungsten carbide is being used.

Double seals are also considered and used for harsh environments involving high temperature applications. Figure 17-37 shows a double-opposed stationary seal design. This seal is operating at the following conditions:

- Liquid sealed: residual bottoms
- Vapor pressure at pumping temperature: 1.25 psia
- Viscosity: 0.6 cp
- Temperature: 700°
- Buffer liquid: gas oil
- Temperature: 150°
- Stuffing box pressure: 200 psig
- Speed: 3550 rpm
- Seal size: 3.000 inches

Seal faces are carbon versus tungsten carbide. Double seals on applications such as this have a seal life that can be 2½ times longer than a single seal installation.

Figure 17-38 shows another multiple installation. This seal can be used as a double or a tandem seal, depending on pressure between the seal

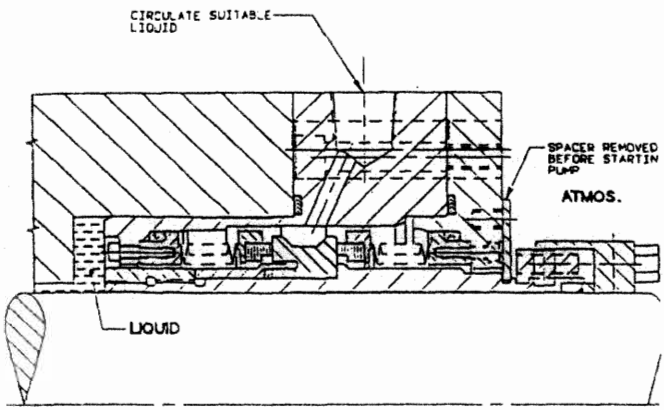


Figure 17-37. Package double opposed metal bellows seal for high temperature (courtesy of John Crane).

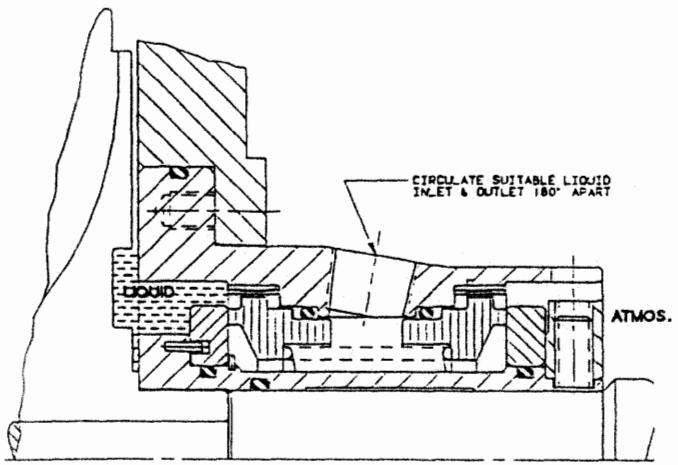


Figure 17-38. Package symmetrical seal (courtesy of John Crane).

heads. If the pressure is greater than the liquid being pumped, it is operating as a double seal. If the pressure is less than the liquid being pumped, it is operating as a tandem. This seal is designed to operate at the following operating conditions:

- Liquid sealed: various
- Buffered liquid: oil, water, or a non-hazardous liquid compatible with the process
- Temperature: 300°F (max)
- Speed: 3600 rpm

This design is a package seal that includes the stuffing box as an integral part of the seal assembly. This model is built for an end suction pump and can be removed from the pump without disassembling any bearings. This is referred to as a Symmetrical Seal and is a patented design by John Crane Inc. This seal design is ideally suited for incorporating the upstream pumping concept that was previously discussed.

High speed pump applications offer a unique opportunity for the use of a non-contacting, non-leaking seal. Through the use of the concept of upstream pumping, a stuffing box can be run dead ended. The only lubrication and cooling is achieved through the liquid being moved from the inside diameter of the seal face to its outside diameter. This design is illustrated in Figure 17-39. In this tandem seal arrangement, liquid between the tandem seals at 100 psig is being moved to a high pressure of 1200 psig in the stuffing box. Shaft speed is 15,600 rpm and seal size is 1.500 inches. Flow rate from low pressure to high pressure side of the seal is 17 cc/min. There is no measureable increase in surface temperatures at the inboard seal that is handling a differential pressure of 1100 psig. Statically and dynamically, there is no leakage of the liquid being pumped. Buffer liquid between the seals is water. Only a small amount of circulation is required to remove the seal heat generated at the outboard seal. Upstream pumping is a patented design by John Crane Inc. that will begin to change the way one views a seal application. A seal can also be used to do work in the system and reduce costly support equipment that is sometimes required. It also allows a totally different way to flush a seal with extremely small quantities of liquid.

Split seals are sometimes required on certain applications. Shown in Figure 17-40 is a patented split seal by John Crane. In this case all components are split, allowing the changing of a seal without dismantling the entire pump. Split seals have been applied to pumps with large shafts operating at speeds of 1800 rpm and pressures to 80 psig. The unique feature of the floating seal faces allows this type of seal to handle large amounts of shaft runout.

Many standard and unique seal designs are available from seal manufacturers. The most efficient design for a given application can be determined by making all of the operating conditions available to the seal manufacturer.

Mechanical Seal Installation and Troubleshooting

Long trouble-free operation of equipment includes proper seal design, selection of materials of construction, and correct seal installation. Any shortcuts taken during seal installation can result in the equipment being taken out of service in a few hours, days, or even weeks from startup. A

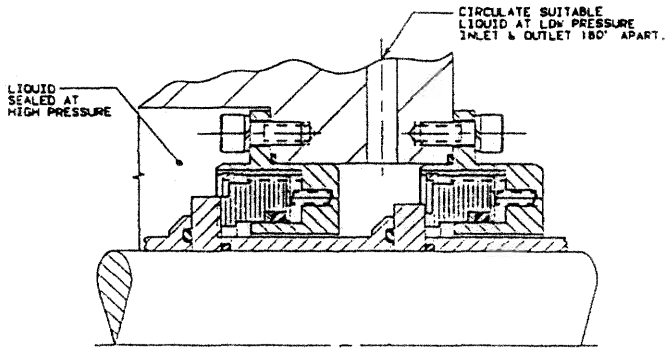


Figure 17-39. High speed application of patented upstream pump concept (courtesy of John Crane).

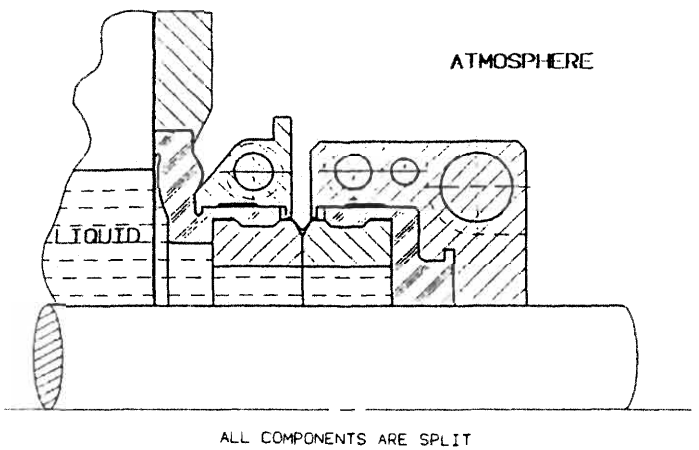


Figure 17-40. Split face seal design (courtesy of John Crane).

pump must operate within the pump manufacturer's specification and should be checked each time a seal is replaced. When a seal is removed due to excessive leakage, a study should be made to identify the cause for leakage. This study can yield results that can extend the life of the next installation.

When a pump shaft begins to rotate, the seal parts also begin to move relative to each other. Not only does sliding occur at the primary and mating rings, but other seal parts move as well. The key to successful seal life is to keep the types of motion transferred to a seal to a minimum. The common types of motion that influence seal performance are angular and axial movement. These types of motion are the result of angular mis-

alignment, parallel misalignment, end play, and radial shaft runout (Figure 17-41). If wear has occurred between seal parts or between seal parts and a shaft sleeve or shaft, the pump should be checked for the type of condition illustrated and corrective measures should be taken. A pump must operate within the pump manufacturer's tolerances.

Angular misalignment results when the mating ring is not square with the shaft. As shaft rotation begins, the primary ring will try to follow the surface of the mating ring. This often results in uneven wear of the softer carbon primary ring, possible internal wear of seal parts, fretting of seal hardware or shaft sleeve. Fretting on a shaft or shaft sleeve occurs when the protective oxide coating is removed from the metal by rubbing motion from a secondary seal in a pusher-type seal design. A hard coating on the shaft or shaft sleeve can be used to eliminate this type of wear. A full convolution elastomeric bellows is designed to protect the shaft or shaft sleeve by taking all the movement in the bellows convolution instead of along the shaft. Angular misalignment may also occur from a stuffing box that has been distorted by piping strain developed at operating temperatures. In this case, damage to the wear rings in the pump may also be found if the stuffing box has been distorted.

Parallel misalignment results when the stuffing box is not properly aligned with the rest of the pump. Generally, with small amounts of parallel misalignment, the seal will operate properly. However, with a large amount of parallel misalignment, the shaft may strike the inside diameter of the mating ring. If damage has occurred, there will also be damage to any bushing located at the bottom of the stuffing box. The point of damage will be in the same location as that on the mating ring.

Excessive axial end play can damage the sealing surfaces and can also cause fretting. If the seal is continually being loaded and unloaded, abrasives can penetrate the seal faces and cause premature wear of the primary and mating rings. Thermal damage in the form of heat checking on the seal faces can result if excessive end play occurs and the seal is operated below its working height.

Radial runout in excess of limits established by the pump manufacturer will cause excessive vibration at the seal. This vibration, coupled with small amounts of other types of motion which have been defined, can shorten seal life. Radial runout from a bent shaft can damage the inside diameter of a mating ring through 360°.

Seal installation begins with a review of the assembly instructions and any seal layout drawings. The proper seal installation dimension or spacing is required to ensure that the seal is at its proper working height (Figure 17-42). The installation reference can be determined by locating the face of the box on the surface of the sleeve and then measuring along the sleeve after it has been removed from the unit. It is not necessary to use

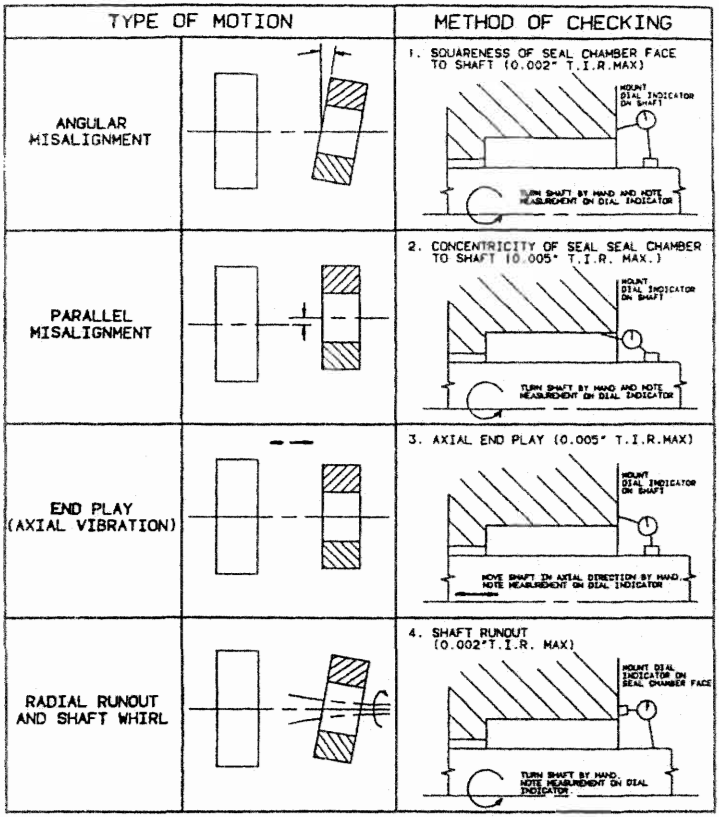


Figure 17-41. Common types of motion that influence seal performance and method of checking the equipment.

this procedure if a step in the sleeve or collar has been designed into the sleeve assembly to provide for proper seal setting. Assembly of other parts of the seal will bring the unit to its correct working height. A single seal mounted as shown in Figure 17-42 is the most economical seal installation available to industry.

Package seals as shown in Figure 17-26 can be assembled with relative ease because only the bolts on the gland plate and set screws in the collar need to be fastened to the stuffing box and shaft. After the spacer is removed, the pump is ready to operate, provided the piping has been replaced.

Because a seal has precision lapped faces and secondary seal surfaces are critical in the assembly, installation should be kept as clean as possible. All lead edges on sleeves and gland plates should have sufficient chamfers to facilitate installation.

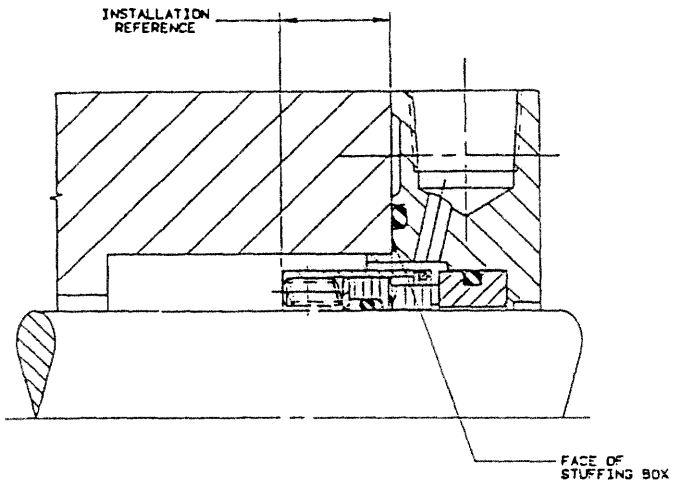


Figure 17-42. Typical installation reference dimensions.

When mechanical seals are properly applied, there should be no static leakage under normal conditions. Dynamic leakage can range from none to several drops per minute, depending on the operating conditions for the unit. For estimates of static and dynamic leakage, the user should consult the seal manufacturer. When conditions call for a vacuum, a mechanical seal is used to prevent air from leaking into the pump.

Seal leakage can occur in many ways. The cause of leakage may be the result of an improper seal selection, installation, environment, or operation of the pump. Leakage may occur past static and secondary seals or past the seal faces. Leakage may occur while the pump shaft is rotating and stationary or just when the pump shaft is rotating. Each observation made about the pump and its operation, as well as the condition of the failed seal parts, plays an important role in identifying the cause for seal leakage. Analyzing the causes for seal leakage can best be done with pictures of actual seal parts.

A leakage path past secondary and static seals are usually confined to elastomeric seals. These types of seals are more vulnerable to nicks, scratches, and cuts. Poor handling and installation practices are the most common causes for surface damage to these types of seal parts. Care should always be taken to remove all sharp burrs and edges from steps and keyways as well as previous set screw identations from prior seal installations. Extrusion failures of secondary elastomeric seals such as O-rings are caused by excessive pressure and can be prevented by reducing the pressure and/or temperature on the seal. This can be accom-

plished by using a tandem seal arrangement as previously discussed. Upgrading the seal materials may be another solution to consider. An example of an extrusion failure is shown in Figure 17-43. This is a corrosion resistant elastomer that had been in chemical service at 700 psig and a temperature of 420°F. The seal had also been subject to excessive axial motion. The seal leakage problem was corrected by reducing the temperature in the seal cavity and eliminating the excessive axial motion from the equipment.

Secondary elastomeric seals may also be subject to excessive volumetric change, either swell or shrinkage. Volumetric swell will result in the loss of seal face flexibility. Volumetric swell of a secondary seal may also result in seal face distortion that can also contribute to leakage. Shrinkage of the secondary seal will result in the loss of interference between the shaft and primary ring, resulting in seal leakage. An O-ring subjected to shrinkage and corrosion is shown in Figure 17-44. This type of condition was corrected by upgrading the material of construction.

Elastomeric O-rings are also subject to a condition referred to as explosive decompression. This can occur when sealing some liquefied gases such as carbon dioxide. Small amounts of gas can penetrate the O-ring at high pressure. As long as pressure is kept on the seal, nothing will happen to the O-ring. Once the pressure in the seal cavity is reduced, the O-ring will expand and burst at the point where the gas has been absorbed into the O-ring. This blistered condition can be avoided by upgrading the material used for the O-ring.

The advantages of using an O-ring as a secondary seal outweigh the disadvantages. This type of secondary seal offers an excellent solution to high pressure applications.

Metal bellows seals offer an excellent solution for temperatures greater than 400°F and pressures to 500 psig. A seal fitted with a metal bellows can be applied to those applications that are essentially non-corrosive and where angular misalignment is not a problem. A bellows used in an environment with a large amount of angular misalignment will fail from fatigue. Cracking of a metal bellows will occur above the weld near the inside diameter of the bellows, as shown in Figure 17-45. This type of seal problem can be eliminated by correcting the motion from angular misalignment.

Motion from angular misalignment, parallel misalignment, shaft runout or whirl, and shaft end play have an effect on seal operation. Motion from parallel misalignment is believed to aid the lubrication process. Motion from other types of conditions will result in larger than normal forces on a primary seal ring. This will lead to unstable operation for the mechanical seal. The increase in load from angular misalignment can be many times larger than the design load of the seal. This increase in load

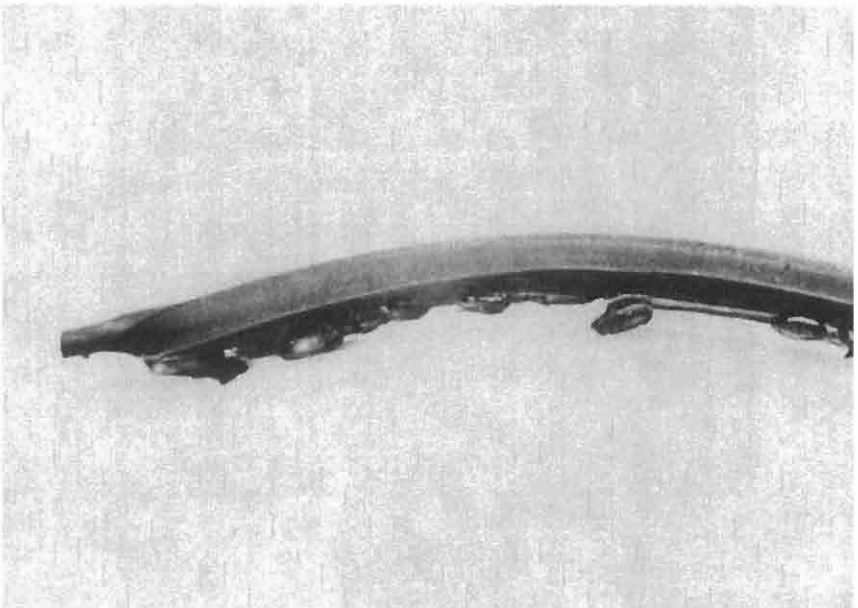


Figure 17-43. Extruded O-ring from high temperature and pressure.

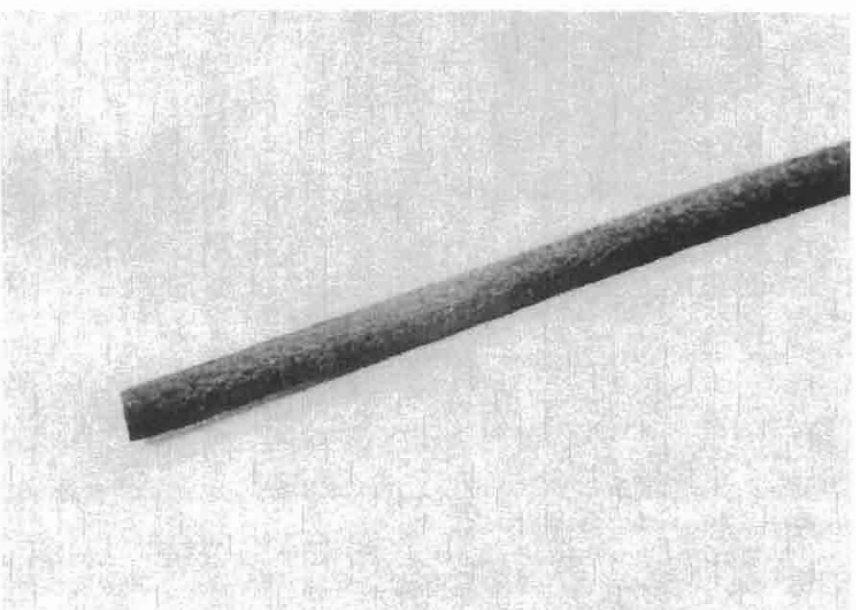


Figure 17-44. Condition of O-ring subject to corrosion, abrasion, and shrinkage.

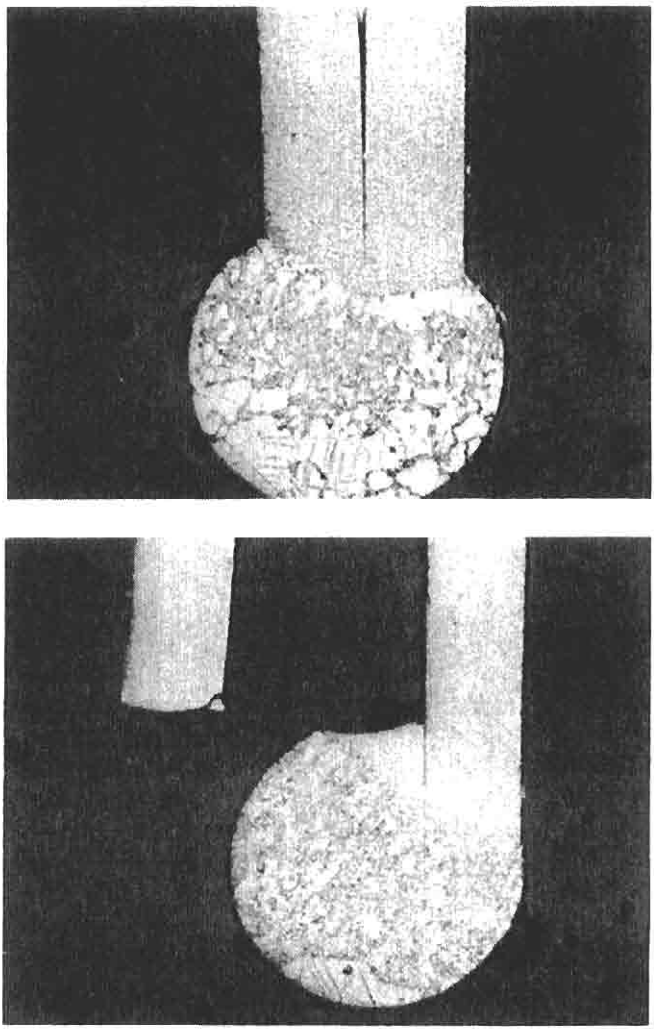


Figure 17-45. Metal bellows welds. A. Typical weld. B. Fatigue crack.

will result in additional frictional heat that will be detrimental to the lubricating film at the seal faces.

The primary ring shown in Figure 17-46 is from a 5.375-inch seal operating at a surface speed of 5000 fpm in water at 180°F. The stuffing box pressure was 150 psig. Misalignment of the seal faces resulted in wear at only one anti-rotation notch. At this point of greatest friction in

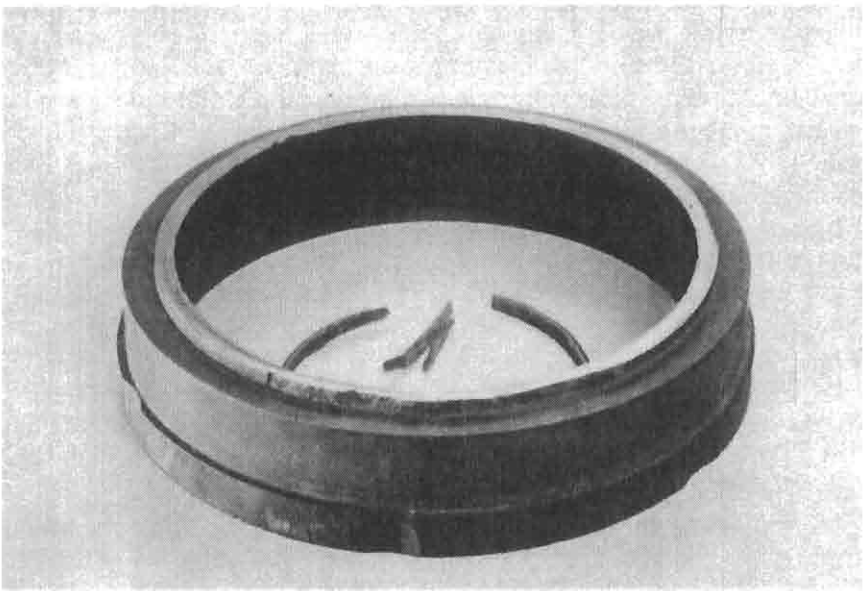


Figure 17-46. Damage to a carbon primary ring from angular misalignment.

the drive system, edge chipping has occurred at the seal face. This impact type of failure occurred after two months of operation. The surface waviness of this part measured 2550 micro-inches through 360°. This is an extremely large value of surface waviness for a sealing plane and would account for a large volume of leakage. The surface profile of this part is irregular, alternating from convex to concave at approximately every 90°. At the chipped edge, the profile surface is 350 micro-inches convex. The maximum concave surface is 200 micro-inches. The secondary seal has been severely damaged due to the motion transmitted to this seal from the misalignment problem. High surface waviness and irregular surface profile readings appear to be typical for seals subjected to impact type of failure from misalignment. In cases where the misalignment is very small, cracking and edge chipping of the carbon ring would not occur; however, uneven wear of the softer seal face materials would be experienced. An alignment problem on this application was corrected to achieve long seal life.

Leakage past the seal faces may be the result of distortion, heat checking, environmental causes, and also equipment motion. Leakage past the seal faces due to distortion can be the result of the pressure in the seal cavity, thermal effects from the environment and mechanical causes from gland plate loading. Leakage from pressure is the result of deflection at

the seal faces. At higher pressure, the seal faces may deflect in a positive direction to the shaft centerline. As the seal is operated at higher pressure, no leakage occurs. At lower pressure, however, the seal will begin to leak because the wear pattern has changed and the seal faces are now opening to the pressure at the outside diameter. This type of leakage from coning or positive seal face rotation can be eliminated by upgrading seal materials or the design of the seal rings.

Leakage due to thermal distortion is just the opposite to what is observed when the seal faces deflect from pressure. Here, wear occurs at the inside diameter of the seal face. The seal face continues to expand, opening up to the liquid in the stuffing box. As the seal opens up at higher temperatures, leakage increases and little or no contact is observed at the outside diameter of the seal faces. This condition is corrected by increasing the cooling to the seal faces. An illustration of damage from thermal distortion is given in Figure 17-47. Damage to this cast iron mating ring was eliminated by providing adequate flow to the seal faces.

Mechanical distortion may result from bolt loading from the gland plate on the pump. Illustrations of mechanical distortion are shown in Figures 17-48, 17-49, and 17-50. In Figure 17-48 a tungsten carbide mating ring had been installed into a unit where the bolts had been overtightened. This changed the complete waviness pattern at the seal faces and resulted in two high spots 180° apart. When this condition occurs, the separation between the mating and primary rings can result in erosion of the softer carbon primary ring. This erosion can take two forms. If the unit is idle with pressure in the stuffing box and the shaft is not rotated, wire drawing can occur across the seal face. This condition is shown in Figure 17-49. In addition to wire drawing shown on the seal faces, erosion of the outside carbon shoulder can be seen. Erosion of the carbon can be eliminated by redirecting the seal flush to the side of the faces, and where possible, an abrasive separator should be added to the seal flush line.

Wire brushing of a seal face can occur when abrasives are present in the stuffing box and the seal is operated with a mechanically distorted mating ring, Figure 17-50. In both cases, erosion to the seal faces can be eliminated by not overtightening the gland plate bolts or by supporting the gland plate outside the bolt circle. Distortion can also occur when the top and bottom halves of a split case pump are not properly lined up. In this case, a review of the application should be made to determine if an abrasive separator should be included in the seal flush line.

Heat checking of a seal face can occur in a pattern through 360° as shown in Figure 17-51. When this condition occurs a few minutes after startup, it would indicate that the seal was operated below its working height or that the seal was operated without any flush to the seal cavity.

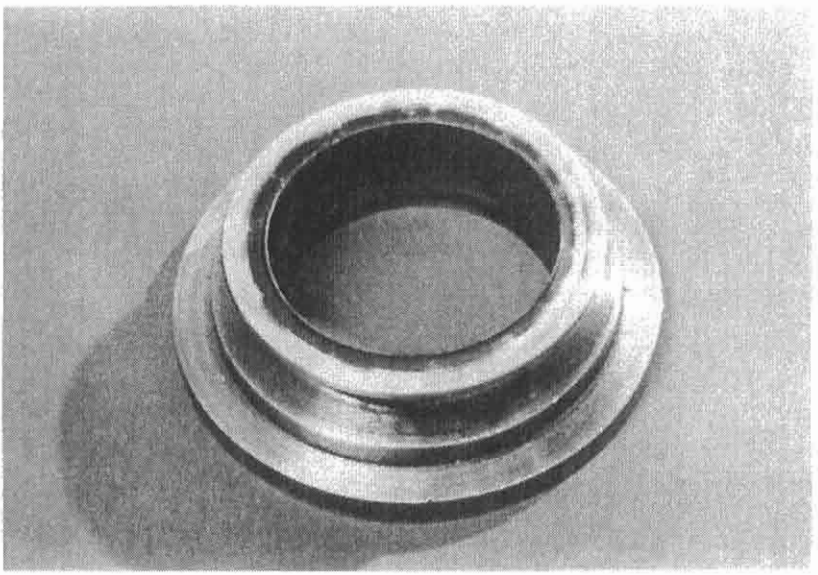


Figure 17-47. Damage to a mating ring surface from thermal distortion.

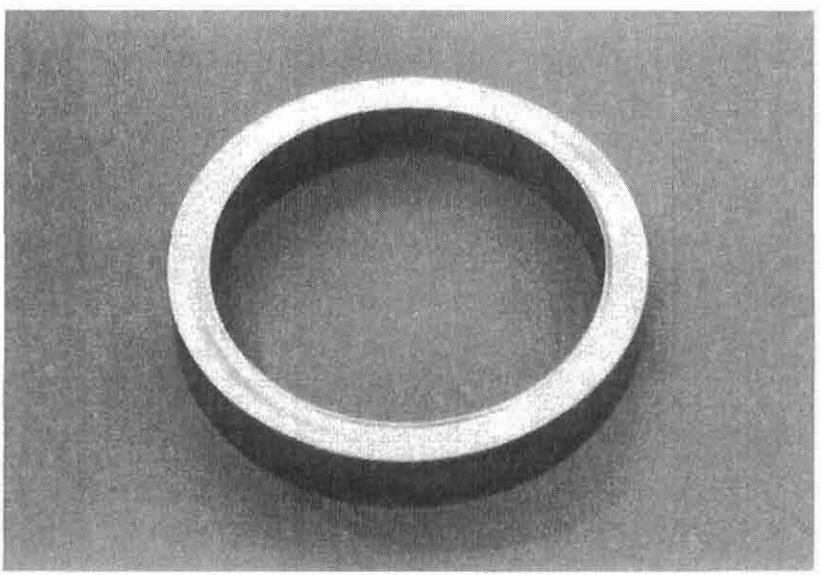


Figure 17-48. A form of damage from mechanical distortion to a mating ring surface.

Figure 17-49. Damaged carbon primary ring showing wire drawing across the seal face and erosion on exterior surfaces.

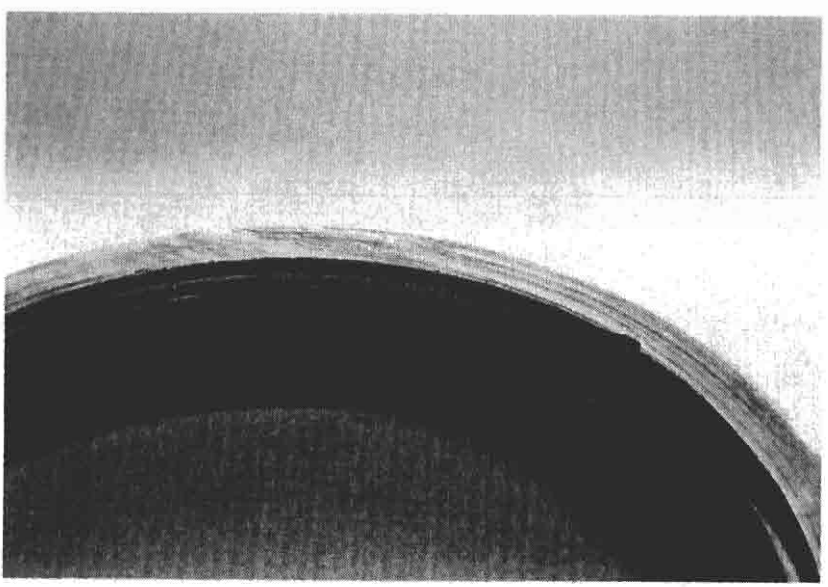


Figure 17-50. A carbon primary ring with surface damage referred to as wire brushing.

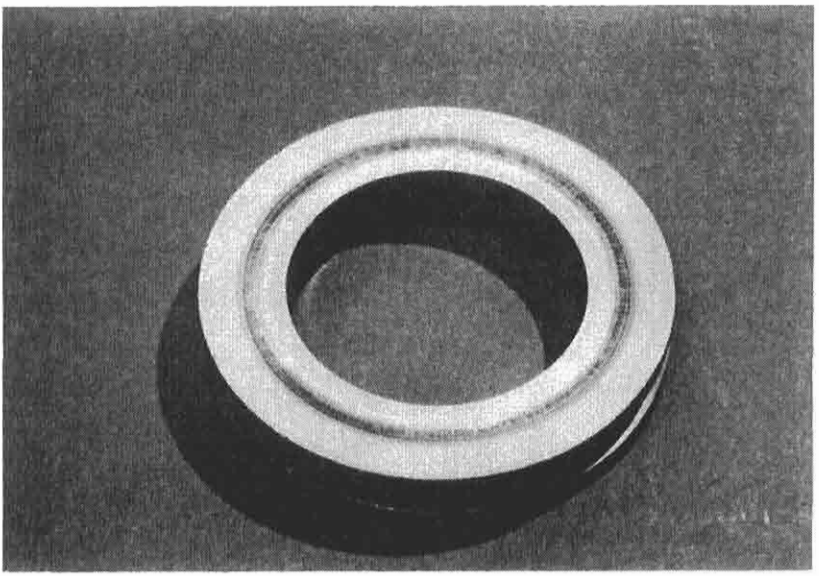


Figure 17-51. Heat checked mating ring surface damaged through 360°.

This type of failure may also occur after a longer period of time in service if the pump had developed excessive axial end play, causing the seal to run solid. In this case, checks should be made to determine if the seal had been properly set at its working height and had sufficient flow to the seal cavity.

Partial heat checking of the sealing surface through an arc less than 180° opposite the flush inlet is the result of not properly distributing the flow of coolant in the stuffing box. In this case, a circumferential groove may be added to the gland plate at the outside diameter of the mating ring or an additional inlet 180° from the existing inlet should be considered. In this particular case, it would also be a good idea to check the squareness of the face of the stuffing box with the shaft to be sure the alignment is correct and that the mating ring has not been pulled to one side when installed.

Hot localized patches of material have been previously discussed in the theory of operation for a mechanical seal. The development of a hot spot is illustrated in Figure 17-52. When a seal operates in an unstable condition for a short period of time, the localized hot spot will appear to be very small. The surface distress at the hot spot occurs because of rapid heating in operation followed by rapid cooling due to the liquid at the seal faces changing to a gas. When the liquid at the seal faces flashes or va-

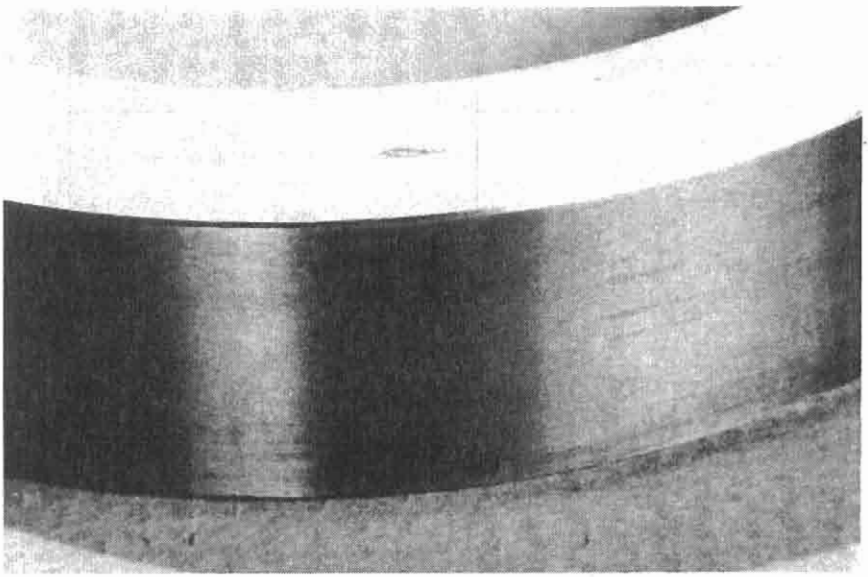


Figure 17-52. Small hot spot on a seal face.

porizes, the seal will open, cooling the spot which results in heat checking of the surface. When removed from the equipment after running for a short period of time, the deformed surface may measure between 5 to 10 micro-inches larger than the surrounding surface. In operation, however, due to the developed heat at the hot spot, the deformed surface will appear to be larger than the lubricating film. If this seal is allowed to run with this type of surface condition, the liquid being sealed will continue to flash and vaporize. Carbonized debris from the liquid being sealed and/or carbon wear particles from the seal faces will begin to build up on exterior surfaces of the seal parts as illustrated in Figure 17-53. The flashing or vaporization of the liquid being sealed may be heard above the normal equipment sounds as a spitting or sputtering sound that may occur every few seconds to several minutes. This type of condition indicates abnormal wear occurring from the damaged mating ring. If a seal is believed to be operating in an unstable condition, it may be checked for sound as well as for the amount of carbonized debris on the exterior surfaces of the gland plate. Eliminating hot patches of thermally distressed material from instability can be achieved through increased cooling direct to the seal faces. If this is not possible, then the seal manufacturer should be contacted to determine whether or not hydropads could be incorporated into the seal design. Thermal distressed or heat checked seal faces commonly occur when a seal is run in liquids that are exceptionally

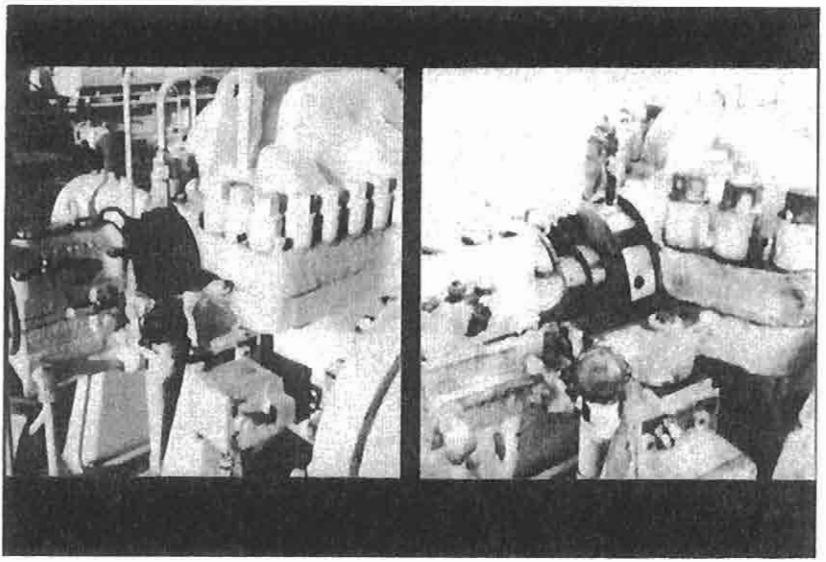


Figure 17-53. Condition of external seal parts at stuffing box. (A) Unstable operation with surface disturbance. (B) Stable operation without any surface disturbance.

poor lubricants. In this case, it is also a good idea to check for mechanical distortion as well.

Mechanical seals are applied to an infinite number of environmental conditions. Some of the liquids to be sealed are extremely poor lubricants for the seal faces and can result in severe wear problems. Figure 17-54 illustrates the faces of an inboard double seal from an agitator. The material of construction for both the primary and mating rings is tungsten carbide. This is the result of running in a liquid with a very low viscosity. The seal lubricant in this case was a silicone oil. The seal life was 11 months. During that time the primary ring wore into the mating ring by as much as 0.125 inch. This type of wear condition was overcome by replacing the lubricant in the seal chamber. In this case, water was selected as the replacement liquid. The result was a substantial increase in seal life.

Normally, in poor lubricating medias, the liquid to be sealed cannot develop a sufficient film to separate both the primary and mating rings. Figure 17-55 illustrates the results of running a silicon carbide primary ring against a tungsten carbide mating ring in a liquid with poor lubricating qualities. The tungsten carbide heat checked and the silicon carbide wore approximately 0.030 inch into the tungsten carbide. The liquid

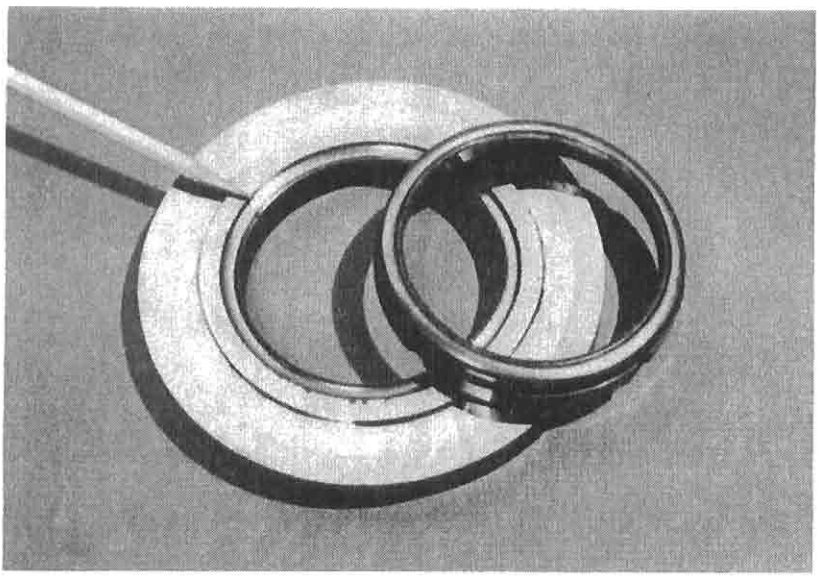


Figure 17-54. Wear of hard seal faces from an environment with poor lubricant.

sealed was ethane at 1200 psig and 3600 rpm. These hard seal faces offered no additional lubricant from the materials of construction. This problem was solved by using a mechanical grade of carbon with hydropads running against a tungsten carbide mating ring.

Poor lubrication can also be a problem on higher temperature applications. Figure 17-56 illustrates the results of operating a metal bellows seal on or near its atmospheric boiling point. Note the large amount of wear on the tungsten carbide surfaces. This also resulted in seal face vibration being transferred to the bellows, causing a leak at that point. This problem was corrected by lowering the temperature in the stuffing box to 50° below the liquid's atmospheric boiling point. Another solution is to consider the possible use of double opposed metal bellows seals with a good lubricant between the seals. Double seals, when properly applied, can last 2½ times longer than single seals on high temperature applications.

Corrosion of seal parts can be a serious problem that will appear over a longer period of time. Corrosion may not be isolated to just one structural part of the seal, but may be common to all parts of the assembly. Carbon materials can become gray, appear to be porous, and lose their hardness. Metals such as tungsten carbide, as shown in Figure 17-57, can have their binder etched out. In this case, once the corrosion covers the entire seal face, leakage occurs. Material selection should be made care-

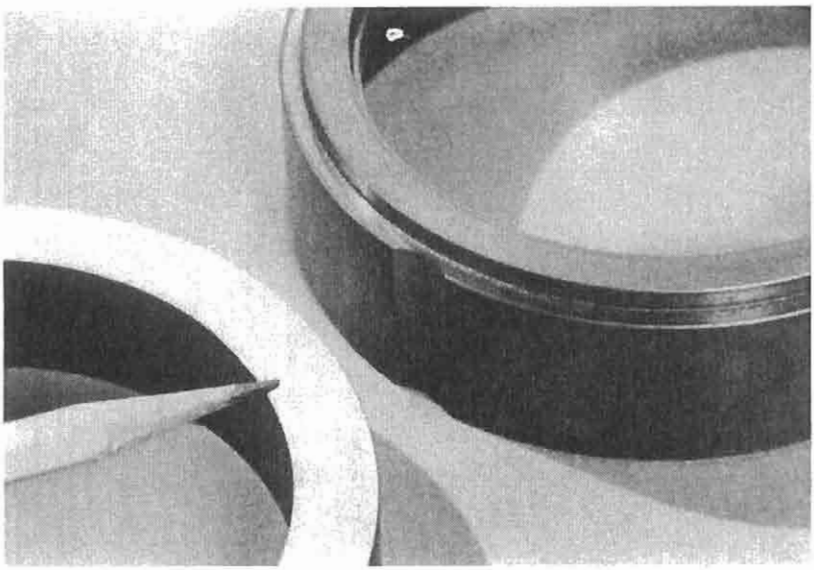


Figure 17-55. Wear of hard faces in light hydrocarbon service at high pressure and high speed.

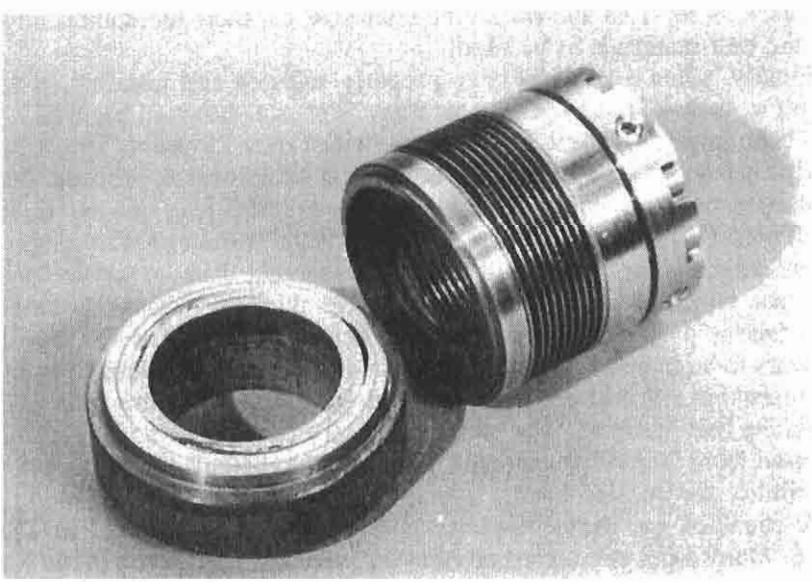


Figure 17-56. Wear of hard seal faces at high temperature.

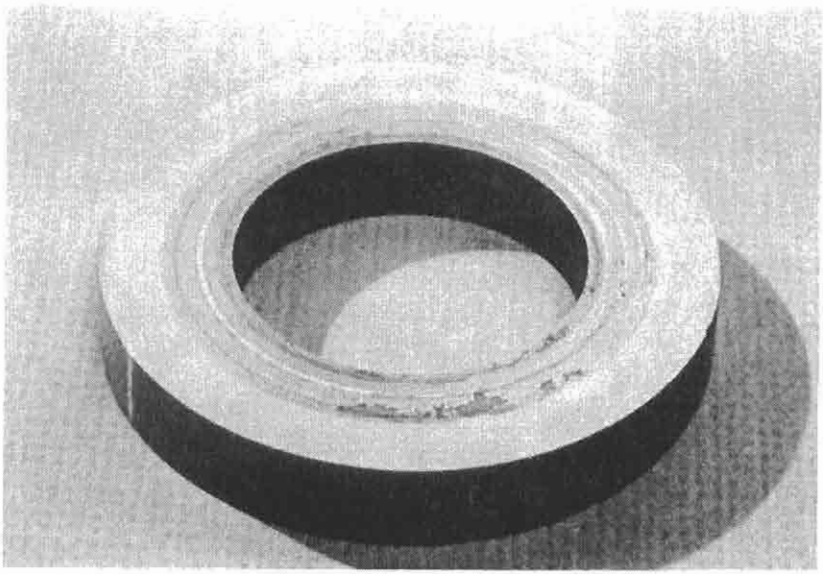


Figure 17-57. Corrosion of a mating ring surface exposed to the product being sealed.

fully. Most seal manufacturers have a wealth of information on materials for various services and should be contacted for their recommendations on the best materials to be used.

Finally, when a seal has been properly selected and installed, it will result in years of trouble-free operation. The seal shown in Figure 17-58 has been in service for 220,000 hours or just over 25 years. This 3-inch seal was used to seal gasoline, fuel oil, and occasionally, propane. Normal operating pressure and temperature were 800 psig and 60°F. The shaft speed was 3560 rpm. The condition of the carbon primary ring exhibits virtually no adverse wear of the sealing face or at the drive notches, which represent the drive system on this seal. Wear at the carbon seal face is only 0.032 inches for this period of time. The mating ring appears to be in good condition after running for this length of time. It is reported that this seal had not been leaking even though some minor heat checking had occurred. The seal aged in service; the O-ring hardness increased from 70 to 90 durometer. This would eventually lead to a loss of flexibility for the seal head.

A checklist for identifying causes of seal leakage is found in Table 17-6. Many other references on causes of seal leakage are available from seal manufacturers. One such source is found in *Identifying Causes of Seal Leakage, Chart and Bulletin S-2031*, available from John Crane Inc.

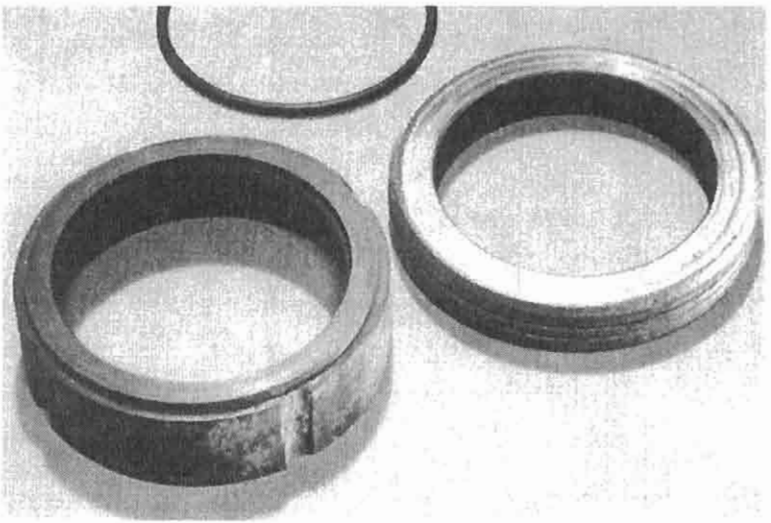


Figure 17-58. Three-inch diameter seal after approximately 220,000 hours of service in petroleum product.

Table 17-6
Checklist for Identifying Causes for Seal Leakage

SYMPTOM	POSSIBLE CAUSES	CORRECTIVE PROCEDURES
SEAL SPITS AND SPUTTERS ("FACE POPPING") IN OPERATION	SEAL FLUID VAPORIZING	o INCREASE COOLING OF SEAL FACES o ADD BY-PASS FLUSH LINE IF NOT IN USE o ENLARGE BY-PASS FLUSH LINE AND/OR ORIFICES IN GLAND PLATE o CHECK WITH SEAL MANUFACTURER FOR PROPER SEAL BALANCE o CHECK SEAL MANUFACTURER FOR ADDED COOLING BY USING HYDROPADS
SEAL DRIPS STEADILY	o FACES NOT FLAT o CARBON GRAPHITE FACES BLISTERED o SEAL FACES THERMALLY DISTORTED	o CHECK FOR INCORRECT INSTALLATION DIMENSION o CHECK FOR IMPROPER MATERIALS OR SEAL DESIGN FOR APPLICATION o IMPROVE COOLING FLUSH LINES o CHECK FOR GLAND PLATE DISTORTION DUE TO OVER TORQUING GLAND BOLTS o CHECK GLAND GASKET FOR PROPER COMPRESSION o CLEAN OUT FOREIGN PARTICLES BETWEEN SEAL FACES; RELAP FACES o CHECK FOR CRACKS AND CHIPS AT SEAL FACES; REPLACE IF NECESSARY

(table continued on next page)

Table 17-6 Continued
Checklist for Identifying Causes for Seal Leakage

SYMPTOM	POSSIBLE CAUSES	CORRECTIVE PROCEDURES
SEAL DRIPS STEADILY	O SECONDARY SEALS NICKED OR SCRATCHED DURING INSTALLATION O O-RINGS OVERAGED O SECONDARY SEALS HARD AND BRITTLE FROM COMPRESSION SET O SECONDARY SEALS SOFT AND STICKY FROM CHEMCIAL ATTACK	O REPLACE SECONDARY SEALS O CHECK FOR PROPER LEAD IN CHAMFER, REMOVE ANY BURRS O CHECK FOR PROPER SEALS WITH SEAL MANUFACTURER O CHECK WITH SEAL MANUFACTURER FOR OTHER MATERIALS
	O SPRING FAILURE O HARDWARE DAMAGED BY EROSION O DRIVE MECHANISMS CORRODED	O REPLACE PARTS O CHECK WITH SEAL MANUFACTURER FOR OTHER MATERIALS
SEAL SQUEALS DURING OPERATION	O AMOUNT OF LIQUID INADEQUATE TO LUBRICATE SEAL FACES	O ADD BY-PASS FLUSH LINE O ENLARGE BY-PASS LINE AND/OR ORIFICES IN GLAND PLATE
CARBON DUST ACCUMULATES ON OUTSIDE OF GLAND PLATE	O AMOUNT OF LIQUID INADEQUATE TO LUBRICATE SEAL FACES	O ADD BY-PASS FLUSH LINE O ENLARGE BY-PASS FLUSH LINE AND/OR ORIFICES IN GLAND PLATE
	O LIQUID FILM EVAPORATING BETWEEN SEAL FACES	O IF CHAMBER PRESSURE IS HIGH, CHECK SEAL DESIGN WITH SEAL MANUFACTURER
SEAL LEAKS	O NOTHING APPEARS TO BE WRONG	O REFER TO LIST UNDER "SEAL DRIPS STEADILY" O CHECK FOR SQUARENESS OF CHAMBER FACE TO SHAFT O PROPERLY ALIGN SHAFT, IMPELLER, BEARING ETC.TO PREVENT SHAFT VIBRATION AND/OR DISTORTION AT GLAND PLATE AND MATING RING
SHORT SEAL LIFE	O ABRASIVE FLUID	O PREVENT ABRASIVES FROM ACCUMULATING AT SEAL FACES O ADD BY-PASS FLUSH LINE O USE ABRASIVE SEPARATOR
	O ABRASIVE MATERIALS BUILDING UP AT ATMOSPHERIC SIDE OF SEAL	O ADD A LIQUID OR STEAM QUENCH
	O SEAL RUNNING TOO HOT	O INCREASE COOLING TO SEAL FACES O INCREASE BY-PASS FLUSH LINE FLOW O CHECK FOR OBSTRUCTED FLOW IN COOLING LINES
	O EQUIPMENT MECHANICALLY OUT-OF-LINE	O ALIGN--CHECK FOR RUBBING OF SEAL ON SHAFT

References

American Petroleum Institute: Centrifugal Pumps for General Refinery Services, API Standard 610, 7th Edition, Washington, DC, 1980.

ANSI/ASME B73.1 Specifications for Horizontal, End Suction Centrifugal Pumps for Chemical Pumps. Mechanical Engineering, June 1985, pp. 93.

Burton, R. A., "Thermal Deformation in Frictionally Heated Contact," *Wear,* 59, 1980, pp. 1–20.

Hamaker, J. B., "Mechanical Seals Lubrication Systems," 1977 ASLE Education Program-Fluid Sealing Course, American Society of Lubrication Engineers and John Crane Inc., Morton Grove, IL, May 1977.

John Crane Inc. *Engineered Fluid Sealing: Materials, Design, and Applications,* Morton Grove, IL, 1979.

John Crane Inc. *Identifying Causes of Seal Leakage,* S-2031 and Bulletin, Morton Grove, IL, 1979.

Kennedy, Jr., F. E. and Grim, J. N., "Observation of Contact Conditions in Mechanical Face Seals," ASLE Trans. 27, 1984, pp. 122–128.

Kilaparti, R. and Burton, R. A., "A Moving Hot Spot Configuration for a Seal-like Geometry, With Frictional Heating, Expansion and Wear," ASLE Trans. 20, 1977, pp. 64–70.

Netzel, J. P., "Identifying Causes of Seal Leakage," Pacific Energy Association, Pump Workshop, Anaheim, CA, October 13, 1982.

_____, "Mechanical Seals for Light Hydrocarbon Service," Fifth Annual Energy-Sources Technology Conference, New Orleans, Louisiana, March 8–10, 1982, The Petroleum Division, ASME, pp. 47–53.

_____, "Observation of Thermoelastic Instability in Mechanical Face Seals," *Wear,* 59, 1980, pp. 135–148.

_____, "Sealing Light Hydrocarbon as a Liquid or Gas," ASME, South Texas Section, Houston, Texas, October 29, 1985.

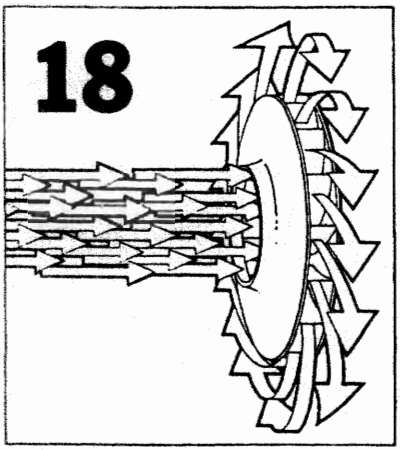
_____, "Symmetrical Seal Design: A Sealing Concept For Today," First International Pump Symposium, Houston, Texas, May 1984, Proceedings, pp. 109–112.

_____, "Wear of Mechanical Seals in Light Hydrocarbon Service," *Wear* 102, 1985, pp. 141–151.

Schoenherr, K., "Design Terminology for Mechanical Face Seals," SAE Trans 74, (65030), 1966.

Schoenherr, K. and Johnson, R. L., "Seal Wear," *Wear Control Handbook,* M. Peterson and W. Winer, Eds., American Society of Mechanical Engineers, New York, 1980.

Snapp, R. B., "Theoretical Analysis of Face Type Seals With Varying Radial Face Profiles," 64-WA/Lub 6, American Society of Mechanical Engineers, New York, 1964.



Vibration and Noise in Pumps

by **Fred R. Szenasi***
Engineering Dynamics Incorporated

Introduction

Although a certain amount of noise is to be expected from centrifugal pumps and their drivers, unusually high noise levels (in excess of 100 dB) or particularly high frequencies (whine or squeal) can be an early indicator of potential mechanical failures or vibration problems in centrifugal pumps. The purpose of this chapter is to concentrate on the mechanisms that may produce noise as a by-product; however, reduction of the noise, *per se*, is not the main concern. The main point of interest of this chapter is to study the mechanisms and their effect on the reliability of the pump system. Methods will be presented to reduce the vibration (and noise) or eliminate the basic causes by modifying the pump or piping system.

The occurrence of significant noise levels indicates that sufficient energy exists to be a potential cause of vibrations and possible damage to the pump or piping. Defining the source and cause of noise is the first step in determining whether noise is normal or whether problems may exist. Noise in pumping systems can be generated by the mechanical motion of pump components and by the liquid motion in the pump and piping systems. Noise from internal mechanical and liquid sources can be transmitted to the environment.

Effective diagnosis and treatment of noise sources to control pump noise require a knowledge of the liquid and mechanical noise-generation

* The author wishes to acknowledge the contributions by the engineering staff of Engineering Dynamics Inc., who performed many of the analyses and field tests.

mechanisms and common noise conduction paths by which noise can be transmitted. If noise itself is the major concern, it can be controlled by acoustic enclosures or other treatment [1, 2].

Sources of Pump Noise

Mechanical Noise Sources

Common mechanical sources that may produce noise include vibrating pump components or surfaces because of the pressure variations that are generated in the liquid or air. Impeller or seal rubs, defective or damaged bearings, vibrating pipe walls, and unbalanced rotors are examples of mechanical sources.

In centrifugal machines, improper installation of couplings often causes mechanical noise at twice pump speed (misalignment). If pump speed is near or passes through the lateral critical speed, noise can be generated by high vibrations resulting from imbalance or by the rubbing of bearings, seals, or impellers. If rubbing occurs, it may be characterized by a high-pitched squeal. Windage noises may be generated by motor fans, shaft keys, and coupling bolts. Damaged rolling element bearings produce high-frequency noise [3] related to the bearing geometry and speed.

Liquid Noise Sources

These are pressure fluctuations produced directly by liquid motion. Liquid noise can be produced by vortex formation in high-velocity flow (turbulence), pulsations, cavitation, flashing, water hammer, flow separation, and impeller interaction with the pump cutwater. The resulting pressure pulsations and flow modulations may produce either a discrete or broad-band frequency component. If the generated frequencies excite any part of the structure including the piping or the pump into mechanical vibration, then noise may be radiated into the environment. Four types of pulsation sources occur commonly in centrifugal pumps [2]:

- Discrete-frequency components generated by the pump impeller such as vane passing frequency and multiples.
- Flow-induced pulsation caused by turbulence such as flow past restrictions and side branches in the piping system.
- Broad-band turbulent energy resulting from high flow velocities.
- Intermittent bursts of broad-band energy caused by cavitation, flashing, and water hammer.

A variety of secondary flow patterns [4] that produce pressure fluctuations are possible in centrifugal pumps, as shown in Figure 18-1, particularly for operation at off-design flow. The numbers shown in the flow stream are the locations of the following flow mechanisms:

1. Stall
2. Recirculation (secondary flow)
3. Circulation
4. Leakage
5. Unsteady flow fluctuations
6. Wake (vortices)
7. Turbulence
8. Cavitation

Causes of Vibrations

Causes of vibrations are of major concern because of the damage to the pump and piping that generally results from excessive vibrations. Vibrations in pumps may be a result of improper installation or maintenance,

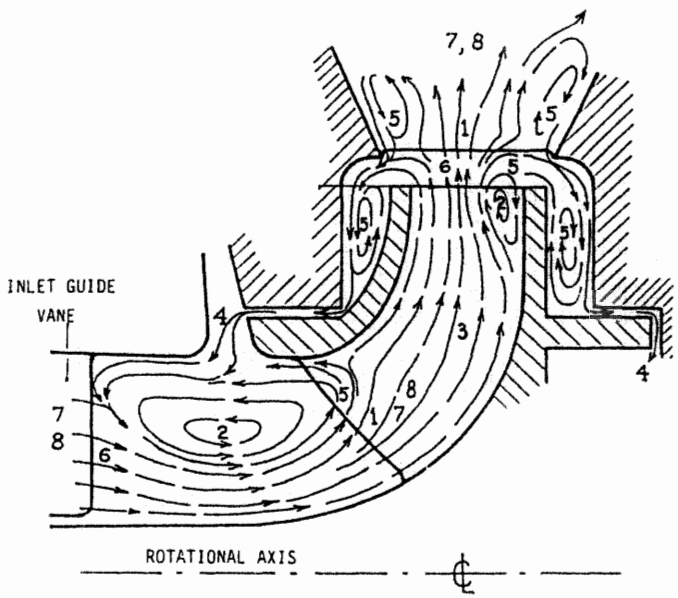


Figure 18-1. Secondary flow around pump impeller off-design flow EPRI Research Project 1266-18, Report CS-1445 [4].

incorrect application, hydraulic interaction with the piping system, or design and manufacturing flaws. Some of the common causes of excessive vibrations and failures are [5]:

Installation/Maintenance

Unbalance
Shaft-to-shaft misalignment
Seal rubs
Case distortion caused by piping loads
Piping dynamic response (supports and restraints)
Support structural response (foundation)
Anchor bolts/grout
Improper assembly

Application

Operating off of design point
Improper speed/flow
Inadequate NPSH
Entrained air

Hydraulic

Interaction of pump (head-flow curve) with piping resonances
Hydraulic instabilities
Acoustic resonances (pressure pulsations)
Water hammer
Flow distribution problems
Recirculation
Cavitation
Flow induced excitation (turbulence)
High flow velocity

Design/Manufacturing

Lateral critical speeds
Torsional critical speeds
Improper bearings or seals
Rotor instability
Shaft misalignment in journals
Impeller resonances
Bearing housing/pedestal resonances

Many of these causes are a result of an interaction of the pump (or its driver) with the fluid or the structure (including piping). This interactive relationship requires that the complete system be evaluated rather than investigating individual components when problems occur. Although prototype pumps or a new design may run the gambit of these problems, standard design or "off-the-shelf" pumps are not immune, particularly to system problems.

Installation/Maintenance Effects

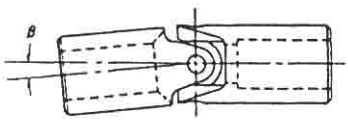
Unbalance. Unbalance of a rotating shaft can cause large transverse vibrations at certain speeds, known as critical speeds, that coincide with the lateral natural frequencies of the shaft. Lateral vibration due to unbalance is probably the most common cause of downtime and failures in centrifugal pumps. Damage due to unbalance response may range from seal or bearing wipes to catastrophic failures of the rotor. Excessive unbalance can result from rotor bow, unbalanced couplings, thermal distortion, or loose parts. All too often, field balancing is required even after careful shop balancing has been performed.

Although a pump rotor may be adequately balanced at startup, after a period of operation the pump rotor may become unbalanced by erosion, corrosion, or wear. Unbalance could also be caused by non-uniform plating of the pumped product onto the impeller. In this instance, cleaning the impeller could restore the balance. Erosion of the impeller by cavitation or chemical reaction with the product may cause permanent unbalance requiring replacement of the impeller. Wear of the impeller or shaft caused by rubs will require the repair or replacement of the damaged component. Another cause of unbalance can occur if lubricated couplings have an uneven build-up of grease or sludge.

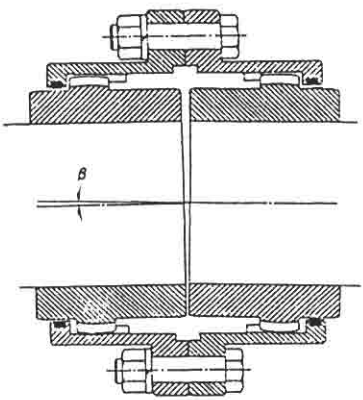
Assembly or manufacturing procedures may cause a new pump rotor to be unbalanced because of slight manufacturing imperfections or tolerance build-up resulting in the center of mass of the rotor not being exactly at the center of rotation. Forging or casting procedures can produce local variations in the density of the metal due to inclusions or voids. On large cast impellers, the bore for the shaft may not be exactly centered with the casting geometry. Stacking a rotor can result in thermal distortions of the shaft or impellers that can result in a cocked impeller. Nonsymmetries of just a few mils caused by these manufacturing or assembly methods can result in significant forces generated by a high speed rotor. Most of these nonsymmetries can be compensated for by balancing the rotor.

Misalignment. Angular misalignment between two shafts connected with a flexible coupling introduces an additional driving force that can

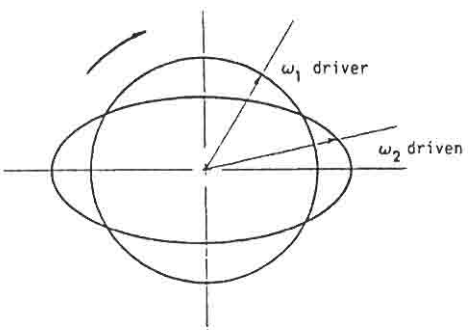
produce torsional or lateral vibrations. The forces in a typical industrial coupling are similar to those in a universal joint (Figure 18-2). When a small angular misalignment occurs, the velocity ratio across the joint is not constant. If one shaft speed is assumed constant, then the other shaft has a faster rotational rate [6] for part of the revolution and a slower rotational rate for part of the revolution. This variation of rotating speed results in a second harmonic (twice shaft speed) vibrational component.



(a) Universal Joint



(b) Geared Coupling with Angular Misalignment



(c) Polar Angular Velocity Diagram

Figure 18-2. Effects of angular misalignment in shaft couplings.

Piping and Structure. The pump should be relatively isolated from the piping. The weight and thermal loading on the suction and discharge connections should be minimized. The American Petroleum Institute (API) Standard 610 [7] specifies allowable external nozzle forces and moments. Most pump manufacturers specify allowable weight and thermal loads transferred from the pipe to the pump case. Static forces from the piping may misalign the pump from its driver, or for excessive loading, the pump case may become distorted and cause rubs or seal and bearing damage. Thermal flexibility analyses of the piping should be performed to evaluate piping loads and to design the necessary supports and restraints to minimize the transfer of piping loads to the operating equipment.

Vibrations of the piping or the support structure can be mechanically transferred to the pump. The piping and the structure should not have their resonant frequencies coincident with any of the pump excitations such as vane passing frequency or multiples. The vibrations transferred from the pipe to the structure can be minimized by using a visco-elastic material (i.e., belting material) between the pipe and the pipe clamp.

Application

The initial stage of pump system design should include the task of defining the range of operating conditions for pressure, flow, temperatures, and the fluid properties. The vendors can provide the correct pump geometry for these design conditions. Expected variations in operating conditions and fluid composition, if a significant percentage, may influence the design.

Improper application or changing conditions can result in a variety of problems. Operation at high-flow, low-head conditions can cause vibrations of the rotor and case. Inadequate NPSH can result in cavitation that will cause noise and vibration of varying degrees.

Bearings. General purpose, small horsepower pumps in process plants generally have rolling element bearings. Noise and vibrations are commonly a result of bearing wear. As the rolling elements or races wear, the worn surfaces or defects initially produce a noise and as wear increases vibrations may become noticeable. Several vibrational frequencies may occur that depend on the geometry of the bearing components and their relative rotational speeds [3]. The frequencies are generally above operating speed.

Many ball bearing failures [8] are due to contaminants in the lubricant that have found their way into the bearing after the machine has been placed in operation. Common contaminants include moisture, dirt, and

other miscellaneous particles which, when trapped inside the bearing, may cause wear or permanently indent the balls and raceways under the tremendous stresses generated by the operating load.

Special purpose pumps and large boiler feed pumps commonly have oil film (hydrodynamic) bearings. The hydrodynamic bearing is superior to rolling element bearings for high speed or high load application. The hydrodynamic bearing supports the rotor on a film of oil as it rotates. The geometry of the hydrodynamic bearing and the oil properties play an important role in controlling the lateral critical speeds and consequently the vibrational characteristics of the pump.

Seals. The fluid dynamics of flow through seals have a dramatic effect on rotordynamics [9]. Hydrodynamic forces involved may contribute to the stabilization of rotating machinery or make it unstable. Seals with large axial flow in the turbulent range, such as in feed water pumps, tend to produce large stiffness and damping coefficients that are beneficial to rotor vibrations and stability. Wear of the seals will increase the clearance and cause greater leakage and possibly change the rotordynamic characteristics of the seal resulting in increased vibrations.

Hydraulic Effects

Hydraulic effects and pulsations can result in almost any frequency of vibration of the pump or piping from once per revolution up to the vane passing frequency and its harmonics. Frequencies below running speed can be caused by acoustical resonances. Generally, these effects are due to the impeller passing the discharge diffuser or some other discontinuity in the case. Any nonsymmetry of the internals of the pump may produce an uneven pressure distribution that can result in forces applied to the rotor.

Transients. Starting and stopping pumps with the attendant opening and closing of valves is a major cause of severe transients in piping systems. The resulting pressure surge, referred to as water hammer, can apply a sudden impact force to the pump, its internals, and the piping. Severe water hammer has caused cracks in concrete structures to which the pipe was anchored.

Rapid closure of conventional valves used in feedwater lines can cause severe water hammer. Increasing the closure time of the valve can reduce the severity of the surge pressure. Analytical methods are available to evaluate the severity of water hammer in a particular piping configuration for various closure rates [10].

Cavitation and Flashing. For many liquid pump piping systems, it is common to have some degree of flashing and cavitation associated with the pump or with the pressure control valves in the piping system. High flow rates produce more severe cavitation because of greater flow losses through restrictions.

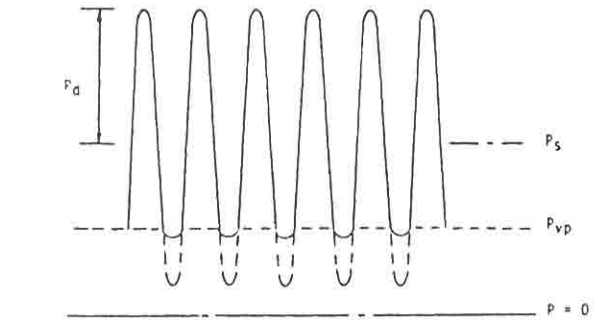
Cavitation produces high local pressures that may be transmitted directly to the pump or piping and may also be transmitted through the fluid to other areas of the piping. Cavitation is one of the most commonly occurring and damaging problems in liquid pump systems. The term cavitation refers to the formation and subsequent collapse of vapor bubbles (or cavities) in a liquid caused by dynamic pressure variations near the vapor pressure. Cavitation can produce noise, vibration, loss of head and capacity as well as severe erosion of the impeller and casing surfaces.

Before the pressure of the liquid flowing through a centrifugal pump is increased, the liquid may experience a pressure drop inside the pump case. This is due in part to acceleration of the liquid into the eye of the impeller and flow separation from the impeller inlet vanes. If flow is in excess of design or the incident vane angle is incorrect, high-velocity, low-pressure eddies may form. If the liquid pressure is reduced to the vaporization pressure, the liquid will flash. Later in the flow path the pressure will increase. The implosion which follows causes what is usually referred to as cavitation noise. The collapse of the vapor pockets, usually on the nonpressure side of the impeller vanes, causes severe damage (vane erosion) in addition to noise.

When a centrifugal pump is operated at flows away from the point of best efficiency, noise is often heard around the pump casing. The magnitude and frequency of this noise may vary from pump to pump and are dependent on the magnitude of the pump head being generated, the ratio of NPSH required to NPSH available, and the amount by which actual flow deviates from ideal flow. Noise is often generated when the vane angles of the inlet guides, impeller, and diffuser are incorrect for the actual flow rate.

Cavitation can best be recognized by observing the complex wave or dynamic pressure variation using an oscilloscope and a pressure transducer. The pressure waveform will be non-sinusoidal with sharp maximum peaks (spikes) and rounded minimum peaks occurring at vapor pressure as shown in Figure 18-3. As the pressure drops, it cannot produce a vacuum less than the vapor pressure.

Cavitation-like noise can also be heard at flows less than design, even when available inlet NPSH is in excess of pump required NPSH, and this has been a puzzling problem. An explanation offered by Fraser [11, 12] suggests that noise of a very low, random frequency but very high inten-



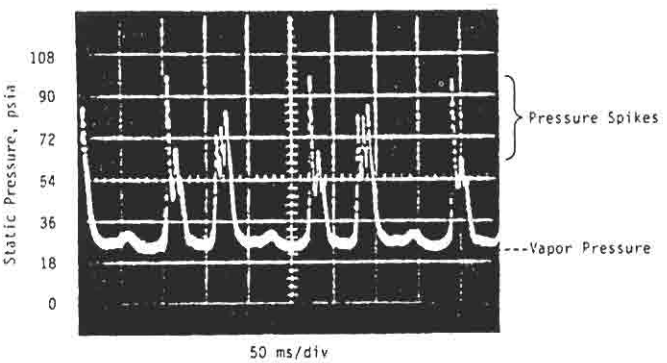
(a) Cavitation Theory

If $P_d > P_s - P_{vp}$ then cavitation will occur.

P_s = Static Pressure

P_d = Dynamic Pulsations, zero-peak

P_{vp}= Vapor Pressure



(b) Complex Waveform of Pressure Showing Effects of Cavitation

Figure 18-3. Cavitation effects on the dynamic pressure.

sity results from backflow at the impeller eye or at the impeller discharge, or both. Every centrifugal pump has this recirculation under certain conditions of flow reduction. Operation in a recirculating condition can be damaging to the pressure side of the inlet and/or discharge impeller vanes (and also to casing vanes). Recirculation is evidenced by an increase in loudness of a banging type, random noise, and an increase in suction and/or discharge pressure pulsations as flow is decreased.

Sound levels measured at the casing of an 8000 hp pump and near the suction piping during cavitation [2] are shown in Figure 18-4. The cavitation produced a wide-band shock that excited many frequencies; however, in this case, the vane passing frequency (number of impeller vanes

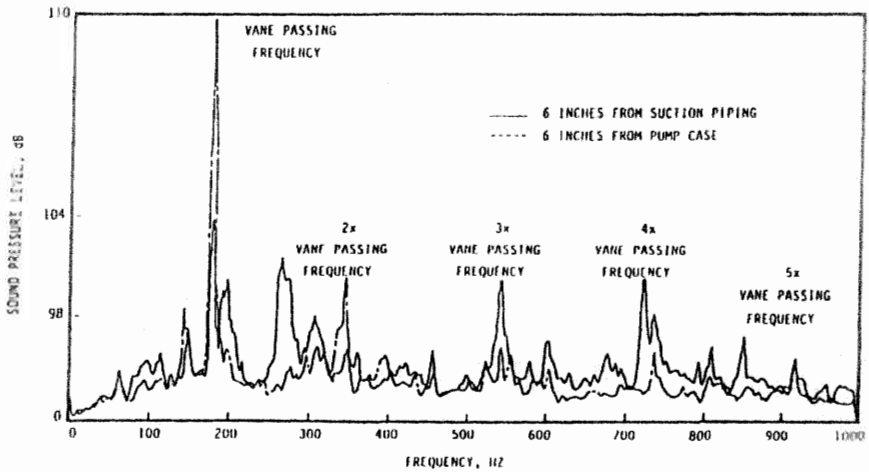


Figure 18-4. Noise spectra of cavitation in centrifugal pump.

times revolutions per second) and multiples of it predominated. Cavitation noise of this type usually produces very high frequency noise, best described as "crackling."

Flashing is particularly common in hot water systems (feedwater pump systems) when the hot, pressurized water experiences a decrease in pressure through a restriction (i.e., flow control valve). This reduction of pressure allows the liquid to suddenly vaporize, or flash, which results in a noise similar to cavitation. To avoid flashing after a restriction, sufficient back pressure should be provided. Alternately, the restriction could be located at the end of the line so that the flashing energy can dissipate into a larger volume.

Flow Turbulence. Pump generated dynamic pressure sources include turbulence (vortices or wakes) produced in the clearance space between impeller vane tips and the stationary diffuser or volute lips. Dynamic pressure fluctuations or pulsations produced in this manner can cause impeller vibrations or can result in shaft vibrations as the pressure pulses impinge on the impeller.

Flow past an obstruction or restriction in the piping may produce turbulence or flow-induced pulsations [2]. These pulsations may produce both noise and vibration over a wide-frequency band. The frequencies are related to the flow velocity and geometry of the obstruction. These pulsations may cause a resonant interaction with other parts of the acoustic piping system.

Most of these unstable flow patterns are produced by shearing at the boundary between a high-velocity and low-velocity region in a fluid

field. Typical examples of this type of turbulence include flow around obstructions or past deadwater regions (i.e., a closed bypass line) or by bi-directional flow. The shearing action produces vortices, or eddies that are converted to pressure perturbations at the pipe wall that may result in localized vibration excitation of the piping or pump components. The acoustic natural response modes of the piping system and the location of the turbulence has a strong influence on the frequency and amplitude of this vortex shedding. Experimental measurements have shown that vortex flow is more severe when a system acoustic resonance coincides with the generation frequency of the source. The vortices produce broad-band turbulent energy centered around a frequency that can be determined with a dimensionless Strouhal number (S_n) from 0.2 to 0.5, where

$$f = \frac{S_n V}{D}$$

where f = vortex frequency, Hz
S_n = Strouhal number, dimensionless (0.2 to 0.5)
V = flow velocity in the pipe, ft/sec
D = a characteristic dimension of the obstruction, ft

For flow past tubes, D is the tube diameter, and for excitation by flow past a branch pipe, D is the inside diameter of the branch pipe. The basic Strouhal equation is further defined in Table 18-1, items 2A and 2B. As an example, flow at 100 ft/sec past a 6-inch diameter stub line would produce broad-band turbulence at frequencies from 40 to 100 Hz. If the stub were acoustically resonant to a frequency in that range, large pulsation amplitudes could result.

Pressure regulators or flow control valves may produce noise associated with both turbulence and flow separation. These valves, when operating with a severe pressure drop, have high-flow velocities which generate significant turbulence. Although the generated noise spectrum is very broad-band, it is characteristically centered around a frequency corresponding to a Strouhal number of approximately 0.2.

Pulsations. Pumping systems produce dynamic pressure variations or pulsations through normal pumping action. Common sources of pulsations occur from mechanisms within the pump. The pulsation amplitudes in a centrifugal pump are generated by the turbulent energy that depends upon the clearance space between impeller vane tips and the stationary diffuser or volute lips, the installed clearances of seal, wear rings and the symmetry of the pump rotor and case. Because these dimensions are not accurately known, predicting the pulsation amplitudes is difficult. Even identical pumps often have different pressure pulsation amplitudes.

Table 18-1
Pulsation Sources

Generation Mechanism	Excitation Frequencies	
1. Centrifugal Compressors & Pumps	$f = \frac{nN}{60}$	
	$f = \frac{nBN}{60}$	B = Number of Blades
	$f = \frac{nvN}{60}$	v = Number of Volutes or Diffuser Vanes
2. Flow Excited		
A. Flow through Restrictions or Across Obstructions	$f = S\frac{V}{D}$	S = Strouhal Number = .2 to .5 V = Flow Velocity, ft/sec D = Restriction diameter, ft
B. Flow Past Stubs	$f = S\frac{V}{D}$	S = .2 to .5
C. Flow Turbulence Due to Quasi Steady Flow	$f = 0 - 30$ Hz (Typically)	
D. Cavitation and Flashing	Broad Band	

Even with the pump operating at its best efficiency point and proper conditions (NPSH, etc.) pulsations may be generated by high-flow velocities and turbulence at the vane tips or at the cutwater. As operating conditions deviate from the design conditions, more sources may come into play such as cavitation, recirculation, flow instabilities, etc.

These pulsations can interact with the hydraulic or acoustic natural frequencies of the piping system to amplify the pulsation. Acoustic natural frequencies in piping systems are a function of the fluid properties, the piping, and pump geometry. The acoustic interaction can be compared to the action of an organ pipe resonance where turbulence produced at the lip is amplified into an audible tone. Similarly, pulsations from the pump are amplified into pressure pulsations that react at elbows, restrictions, closed valves, and piping size changes to cause dynamic shaking forces. This conversion of hydraulic energy into mechanical forces can result in vibrations of the pump, piping, and their support structure.

In the design stage, the acoustical natural frequencies of piping systems can be calculated using either digital [13] or analog [14] modeling

procedures. As an example, a model of a piping system analyzed by a digital acoustic analysis technique is given in Figure 18-5. The system was for chemical service with three pumps (3000 gpm, 250 psi, 3600 rpm) each 50% capacity (one spare). The predicted frequency response in the pump system at selected locations is given in Figure 18-6. The natural frequencies of the energy in the piping system can be compared to discrete frequencies generated by the pump (i.e., vane passing frequency, etc). It can immediately be seen that these 3600 rpm pumps (A and B operating) with a six-vane impeller could cause severe pulsations in the piping system because its vane passing frequency (6 × 60 rps) matches an acoustic response at 360 Hz. Based on the acoustic analysis, a seven-vane impeller should be used that would have its vane passing frequency at 420 Hz (7 × 60 rps) which has minimal response.

While in the design stage, changing the pump impeller for this system was a simple solution; however, the primary use of this acoustic analysis technique is to evaluate alternate piping configurations when the pump cannot be readily changed as in existing installations. Modifications to the piping system (i.e., lengths, routing) can be easily simulated to evaluate the effectiveness in attenuating a particular response mode.

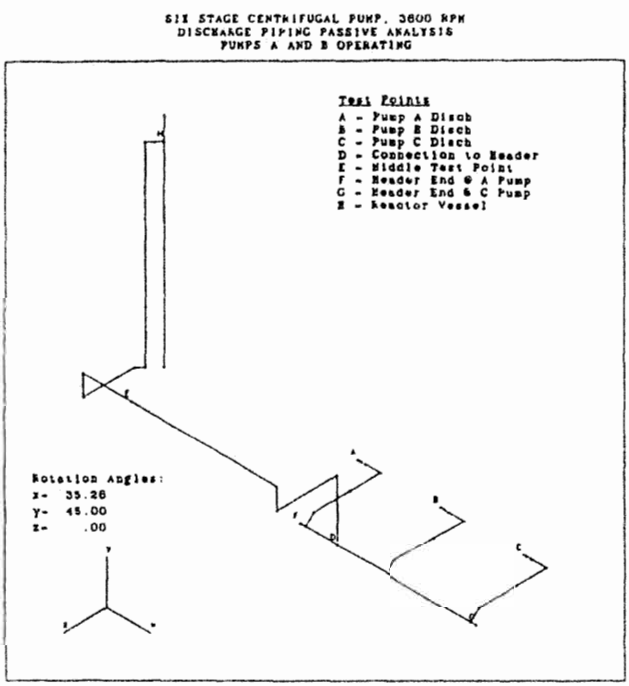


Figure 18-5. Simulation of centrifugal pump piping system.

The mechanical natural frequencies of the piping spans should not occur in the same range as the acoustic response frequencies. The analysis aids in determining the allowable frequency range that can be used to establish the proper design of the piping supports and span lengths to minimize the potential for exciting a piping resonance. The acoustic analysis can be used to redesign the piping to modify the frequency response and particular acoustic modes that may be predominant.

Acoustic Resonance. When a dynamic pressure pulse propagates down a pipe and reaches a restriction or pipe size change (flow area), the pulse is reflected [13, 15]. As the series of pressure pulses continue to be reflected, a standing wave is generated; that is, at a point in the pipe, the pressure periodically rises above and drops below the average line pressure (simple harmonic variation). The super-position of an incident pulse and a reflected pulse, being the sum of two pulses traveling in opposite directions, produce the standing wave.

If the timing (phasing) of a reflected pulse matches a new pulse, the two pulses will add, or amplify. The timing of the pulses are dependent

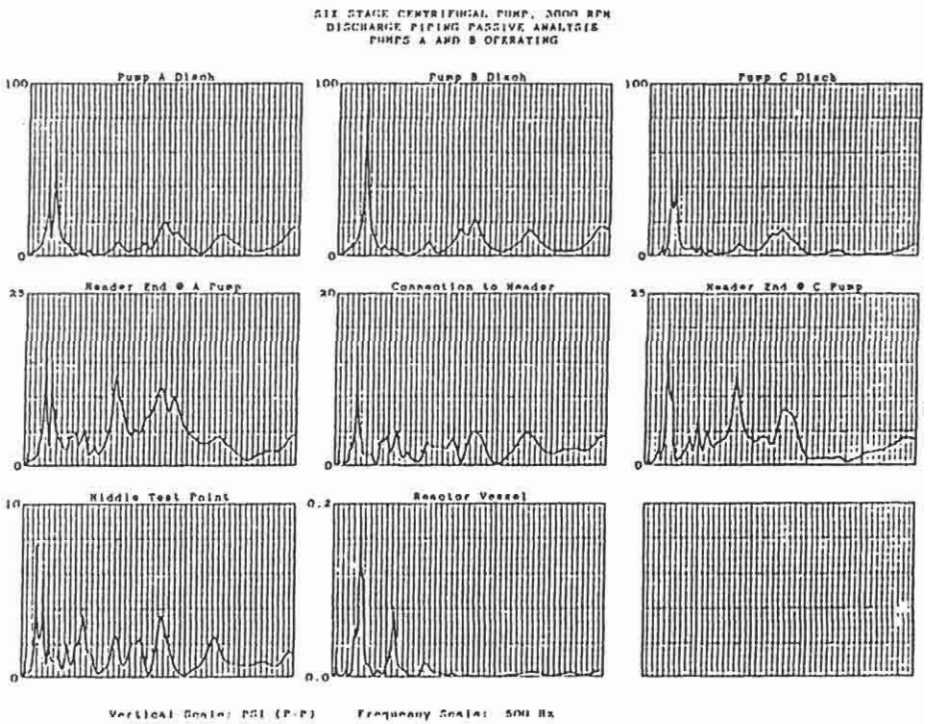


Figure 18-6. Passive acoustic response of piping system.

upon the pump speed (frequency) and pipe length (distance traveled) and the physical properties of the fluid.

The acoustic velocity, a function of the fluid density and bulk modulus, is an important factor in determining the resonant frequency of a pipe length. The API has published a comprehensive handbook for the physical properties of hydrocarbon gasses and liquids [16]. Calculation procedures for the acoustic velocity of water and other common liquids are presented in the Appendix at the end of this chapter.

A resonant condition [17] exists when the standing wave amplitude is reinforced so that the actual maximum dynamic pressure (pulsation) amplitude is substantially greater than the induced pulsation. Thus at frequencies (pump speeds) corresponding to resonance, there would be considerably higher amplitude levels generated from the same amount of energy than for frequencies off resonance.

If the wave frequencies are such that the incident and reflected waves are additive, the pulsations are amplified. If no damping is present, the pressure amplitudes at the anti-nodes would, theoretically, go to infinity. Actual piping systems have acoustic damping as a result of:

- Viscous fluid action (intermolecular shearing).
- Transmission, i.e., lack of total reflection, at a line termination.
- Piping resistance, i.e., pipe roughness, restrictions, orifices.

Therefore, damping of acoustic modes may be accomplished by placement of resistance elements, such as an orifice, that will work most effectively at velocity maxima.

Length Resonances in Distributed Acoustic Systems. The concepts of acoustic waves, reflections, and resonance can be applied to describe some of the classical length resonances [13].

The length resonances of certain piping elements are described in terms of a full-wave length. The acoustic wave length is the distance required for a complete cycle of dynamic pressure reversal. The wave length is related to the driving frequency and the acoustic velocity or speed of sound:

$$\lambda = \frac{c}{f}$$

where λ = wave length, ft/cycle
c = acoustic velocity, ft/sec
f = driving frequency, Hz

Half-Wave Resonance (Open-Open and Closed-Closed)

The first three modes for an open-open pipe are shown in Figure 18-7. Resonances may also occur at integer multiples of the half-wave frequency. For a closed-closed pipe, the formula also applies since both elements have a standing wave that is one-half of a sine wave even though the peaks occur at different locations. The pressure mode shapes of the first three modes are also shown for the open-closed configuration. The length should be corrected for entrance and exit effects (add approximately 80% of the pipe inside diameter) to calculate the half-wave resonance of open-open configurations. The end correction factor becomes very important in short pipes.

Quarter-Wave Resonance (Open-Closed)

The first three modes for an open-closed pipe, commonly referred to as a "quarter-wave stub" are depicted in Figure 18-7. The stub has its resonant frequencies at odd integer multiples of the fundamental quarter-wave frequency. Examples of a quarter-wave stub include a bypass line with a closed valve or a test connection with a pressure gauge.

A quarter-wave resonance can cause erroneous measurements [13] when obtaining dynamic pressure data. A typical test connection, depicted in Figure 18-8, with a short nipple and valve connected to a main line is an acoustical quarter-wave stub. This length can tune up to pulsations in the main line and cause the needle on a pressure gauge to wobble or indicate severe pressure variations that do not actually exist in the main line. Similarly, the data from a dynamic pressure transducer can be misinterpreted.

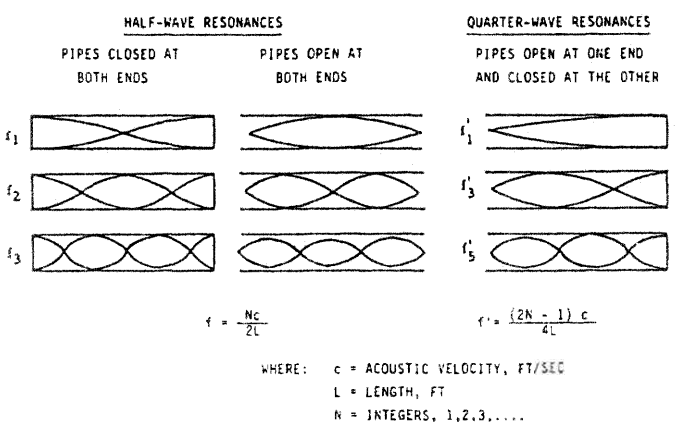


Figure 18-7. Acoustic resonances of piping elements.

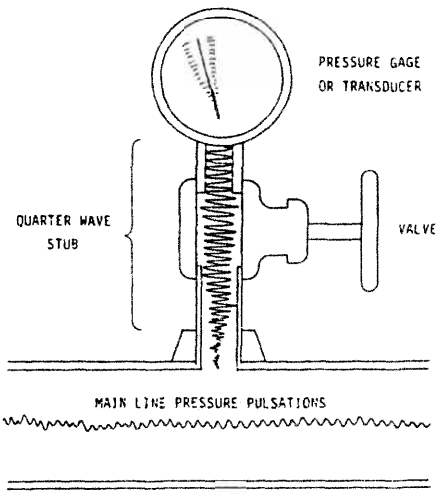


Figure 18-8. Erroneous pressure measurement caused by stub resonance.

For example, an installation where the speed of sound was 4500 ft/sec and the transducer was mounted one foot from the inside surface, the stub (quarter-wave) frequency would be 1125 Hz. A fictitious pressure pulsation component will be measured at this frequency and also at 3X, 5X, 7X, etc. The pulsation exists in the stub; however, it may not be present in the main pipe. If this stub frequency is close to the vane or blade passing frequency, the measured amplitude at the stub frequency will not be valid because it would be amplified by the acoustical resonance. The acoustical amplification factor can be as high as 200. To measure high-frequency pulsations, the transducer should be mounted flush to the inside surface of the pipe.

Measurements of peak-to-peak pulsations on an oscilloscope are often suspect. If the transducer signal frequency spectrum is analyzed and the quarter-wave resonance identified, then credible results can be obtained. An electronic filter may be used to eliminate the undesired frequency.

Coincidence of Driving Acoustic Frequencies and Length Resonances. The existence of quarter or half-wave natural frequencies alone does not constitute resonances. For resonance to occur, a dynamic pulse must be generated at a frequency equal to an acoustical natural frequency. The build-up in amplitude occurs because a reflected wave arrives at the proper time to reinforce the wave generated at the pump. The arrival of the reflected wave is dependent upon the path length of the piping elements. Therefore, the standing wave pattern amplitude is reinforced so that the actual maximum pulsating wave amplitude is substan-

tially greater than the induced level. Multiples of the resonant modes can be excited; however, the multiple wave length resonances generally decrease in severity at the higher multiples because more acoustic energy is required to drive the higher frequency modes.

The acoustic resonances of piping systems for constant-speed pumps can usually be adjusted to detune them from the pump operating speed and vane-passing frequencies and avoid pulsation amplification. However, if the pump is operated over a speed range, the frequency band of the excitations is widened, requiring more careful placement of acoustic resonances.

Actual piping systems are more complex than the simple quarter-wave and half-wave elements. A typical piping system with tees, flow control valves (restrictions) pipe size changes, vessels, etc., will have a complicated pattern of pressure pulse reflection patterns (standing waves). Some of the standing waves may be amplified and others, attenuated. Each of the standing waves will have a particular acoustic length pertaining to a pipe segment between two end conditions. Calculations of the acoustic resonances of a complex piping system require the use of computer codes to consider the acoustic interaction between the pump and its piping system.

Instabilities. Hydraulic instabilities [14] can be a result of the dynamic interaction of a centrifugal pump (particularly the head-flow characteristics) and the acoustic response of the piping system. A centrifugal pump operating at constant speed in a piping system may amplify or attenuate pressure disturbances that pass through the pump. The action of the pump in causing amplification or attenuation of this energy is quite complex, but basically is dependent upon:

- The head curve slope and operating point
- System flow damping (in the piping)
- The existence of strong reactive resonances in the piping, particularly if they coincide with vortex frequencies
- The location of the pump in the standing wave field (i.e., at a velocity maximum rather than a pressure maximum)
- The compressibility (bulk modulus) of the liquid

Pulsations can be amplified by the piping system and cause a variety of problems such as damage to pump internals, torsional reactions, cavitation, vibrations at elbows, valves, or other restrictions. The amplitude of the pulsation is dependent upon operating conditions such as speed, flow rate, and losses (pressure drop) as well as fluid properties and acoustic

natural frequencies. Consequently, pulsation amplitudes are usually affected by changes in the operating conditions or fluid composition.

Pulsations are commonly initiated by flow turbulence at changes in flow cross section, at restrictions (orifices, valves, etc.) or at the pump impeller. When the frequency of this turbulent energy excites one of the acoustic resonances of the piping system, and if the pump is situated near a velocity maximum in the resonant piping system, then high amplitude, self-sustaining pulsations can result. Pulsations can be minimized by moving the pump to a velocity minimum, but often a new pulsation frequency will be generated such that the pump is again situated near a velocity maximum for a higher mode oscillation. With proper care and detailed analysis of the relative strength of the various pulsation resonance modes of the piping, piping designs can be developed to avoid these strong resonances. Controlling piping stub lengths (i.e., in by-pass piping) so that their quarter-wave stub resonances are far removed from the Strouhal excitation frequencies, will also help in minimizing the potential for resonant pulsations.

This type of instability is more probable at low flows because acoustic damping that is generated by flow friction effects is greater at higher flow rates.

Design/Manufacturing

Dynamic response of the pump components to normal exciting forces within its operating frequency range can result in problems from excessive maintenance to catastrophic failure. Improper manufacture or assembly can cause unbalance resulting in damaging vibrations if the speed approaches natural frequencies of the rotor system. A rotordynamic audit [18] of the pump rotor design is crucial in avoiding speed-related vibrations.

The following section discusses the factors that are important in controlling the rotordynamic responses of a centrifugal pump.

Rotordynamic Analysis

A lateral critical speed is defined by API [7] as the speed at which a peak vibrational response occurs. At the critical speed, the rotor is more sensitive to unbalance than at any other speed. The critical speeds of a pump should be avoided to maintain acceptable vibration amplitudes. This section discusses the techniques involved in calculating the lateral critical speeds of centrifugal pumps.

Lateral Critical Speed Analysis

Pump rotordynamics are dependent on a greater number of design variables than are many other types of rotating equipment. Besides the journal bearing and shaft characteristics, the dynamic characteristics of the seals and the impeller-stationary lip interaction can have significant effects on the critical speed location, rotor unbalance sensitivity, and rotor stability [9, 19]. In this context, a seal is an element having a liquid film within a tight clearance. The liquid film has dynamic characteristics similar to a bearing. There are a variety of seal configurations including floating ring seals, grooved seals, and others. Several seal geometries will be discussed.

For modeling purposes, seals can be treated as bearings in the sense that direct and cross-coupled stiffness and damping properties can be calculated based on the seal's hydrostatic and hydrodynamic properties [20]. Seal clearances, geometry, pressure drop, fluid properties, inlet swirl, surface roughness, and shaft speed are all important in these calculations. The high pressure liquid being pumped also flows (or leaks) through the small annular spaces (clearances) separating the impellers under different pressures, such as wear rings and interstage bushings, and creates a hydrodynamic bearing effect that transforms the pump rotor from a two-bearing system to a multi-bearing system. The additional stiffness generated by the pumped liquid as it lubricates these internal bearings (seals, etc.) is referred to as the "Lomakin effect" [21].

The Lomakin stiffness effect minimizes the shaft deflections when the pump is running, and in some cases, the Lomakin effect can be of sufficient magnitude to prevent the critical speed of the rotor from ever being coincident with the synchronous speed. Since the pressure drop across seals increases approximately with the square of the pump speed, the seal stiffness also increases with the square of the speed.

The amount of support derived from the seals as bearings depends upon (a) the pressure differential, and therefore disappears completely when the pump is at rest, and (b) the clearance that increases significantly as the sealing surfaces wear. Consequently, contact between the rotor and stationary parts may take place each time the pump is started or stopped. In consideration of these facts, the rotordynamic analyses should include the effects of worn seals (loose clearances) as well as new seals (tight clearances).

Analytical techniques [22, 23, 24] have been developed whereby the seal geometry can be specified and the characteristics calculated for specific assumptions with regard to inlet swirl, groove design, etc. A series of grooved seal designs used in commercial pumps has been tested to verify the adequacy of the techniques.

A thorough lateral critical speed analysis is essential for developing a reliable, trouble-free pump system. The design audit [18] should include the following calculations:

- Critical speed map
- Undamped natural frequencies and mode shapes
- Bearing stiffness and damping properties
- Seal stiffness and damping properties
- Rotor response to unbalance
- Pedestal and foundation effects on response
- Rotor stability

The first step in performing a lateral critical speed analysis is to model the shaft with sufficient detail and number of masses to accurately simulate the rotor responses through its speed range. An accurate shaft drawing giving the dimensions, weights, and centers of gravity of all added masses is needed to develop the model. Generally, each significant shaft diameter change is represented by one or more stations. A station is generally located at each added mass or inertia, at each bearing and seal location, and at each potential unbalance location. A typical rotor shaft drawing and the computer model is given in Figure 18-9.

Rotating elements such as impellers are modeled as added masses and inertias at the appropriate locations on the shaft. The polar and transverse mass moments of inertia are included in the analyses to simulate the gyroscopic effects on the rotor. The gyroscopic effects are particularly significant on overhung rotors where the impeller produces a restoring moment when whirling in a deflected position.

Couplings are simulated as concentrated added weights and inertias. Normally half the coupling weight is placed at the center of gravity of the half coupling. When necessary, the entire train, including the driver and driven equipment, can be modeled by utilizing programs that can simulate the shear loading and moment transfer across the coupling. Once the shaft model is completed, the critical speed map can be calculated.

Critical Speed Map. The critical speed map is a logarithmic plot of the undamped lateral critical speeds versus the combined support stiffness, consisting of the bearing and support structure as springs in series. The critical speed map provides the information needed to understand the basic response behavior of rotors; therefore, it is important to understand how the map is developed [25].

For large values of support stiffness, the rotor critical speeds are called the rigid bearing critical speeds. If the bearing stiffness is infinity, the vibrations are zero at the bearings, and the first natural frequency for

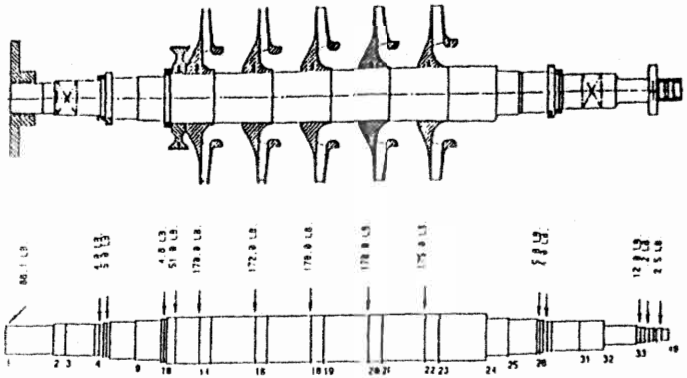


Figure 18-9. Typical shaft drawing and computer model.

shafts that do not have overhung impellers is analogous to a simply supported beam.

A critical speed map, normalized to the first rigid bearing critical speed is given in Figure 18-10 to illustrate the ratios of the various criticals for low and high support stiffness values and to illustrate the mode shapes that the rotor will have at different bearing and support stiffness values. For the rigid bearing critical speeds, the mode shape for the first mode would be a half-sine wave (one loop), the second critical speed would be a two-loop mode and would occur at a frequency of four times the first mode critical, the third critical speed would be a three-loop mode and would be nine times the first critical, etc. For most rotors, the bearing stiffnesses are less than rigid and the second critical will be less than four times (typically two–three times) the first critical.

For low values of support stiffness (shaft stiffness is large compared to support stiffness), the first critical speed is a function of the total rotor weight and the sum of the two support spring stiffnesses. For an ideal long slender beam, the second mode is similar to the rocking of a shaft on two springs and is equal to 1.73 times the first critical speed. Since both the first and second modes are a function of the support stiffness, the slope of the frequency lines for the first and second critical speeds versus support stiffness is proportional to the square root of the stiffness for low values of support stiffness compared to the shaft stiffness.

For a support stiffness of zero, the third and fourth modes would be analogous to the first and second free-free modes of a beam. For an ideal uniform beam, the ratio of the frequencies for these modes compared to the first critical speed for rigid bearings is 2.27 and 6.25.

To aid in the discussion, an example of the critical speed analysis [9] of an eight-stage pump will be presented. The critical speed map for the

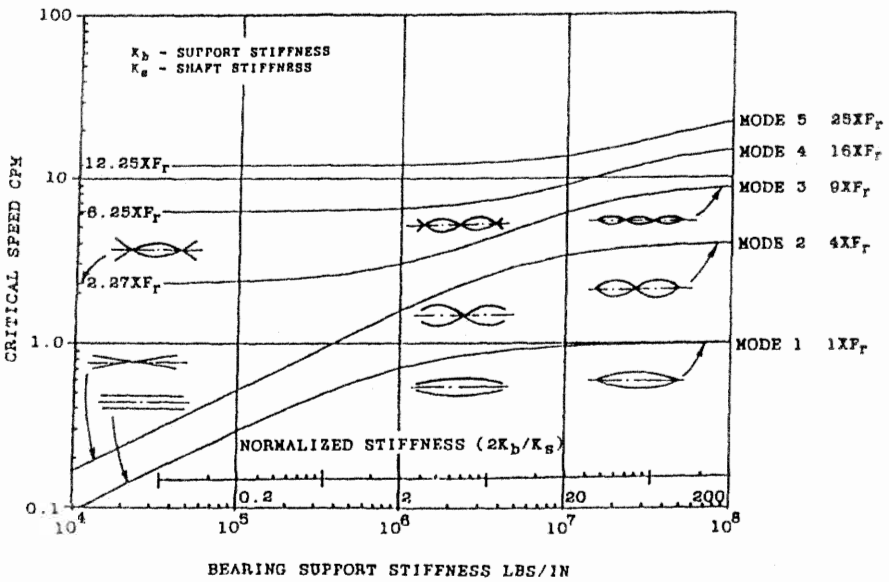


Figure 18-10. Normalized critical speed map.

"dry rotor" model (without the hydrodynamic effect of the seals) is shown in Figure 18-11. The first four undamped lateral critical speeds are plotted versus bearing support stiffness. Horizontal and vertical journal bearing stiffness curves (Kxx and Kyy) for both minimum and maximum assembled clearances are plotted to define the range of critical speeds. Intersections between the bearing stiffness curves and the mode curves represent undamped critical speeds (circled in Figure 18-11). Note that the "Mode 1" curve is fairly flat in the region where the intersections occur. The first two lateral mode shapes were calculated for a nominal bearing stiffness of 500,000 lb/in. and are shown in Figures 18-12 and 18-13. The dry rotor undamped natural frequencies as predicted by the critical speed map for the first, second, and third modes were 1710, 6650, and 8830 cpm, respectively.

Bearing Stiffness and Damping. The dynamic stiffness and damping coefficients of bearings [26] can be adequately simulated using eight linear coefficients (Kxx, Kyy, Kxy, Kyx, Cxx, Cyy, Cxy, Cyx) as shown in Figure 18-14. This information along with the lubricant minimum film thickness, flow, power loss, and temperature rise at operating conditions is needed to evaluate the bearing design. The bearing stiffness and damping coefficients are calculated as functions of the bearing type, length,

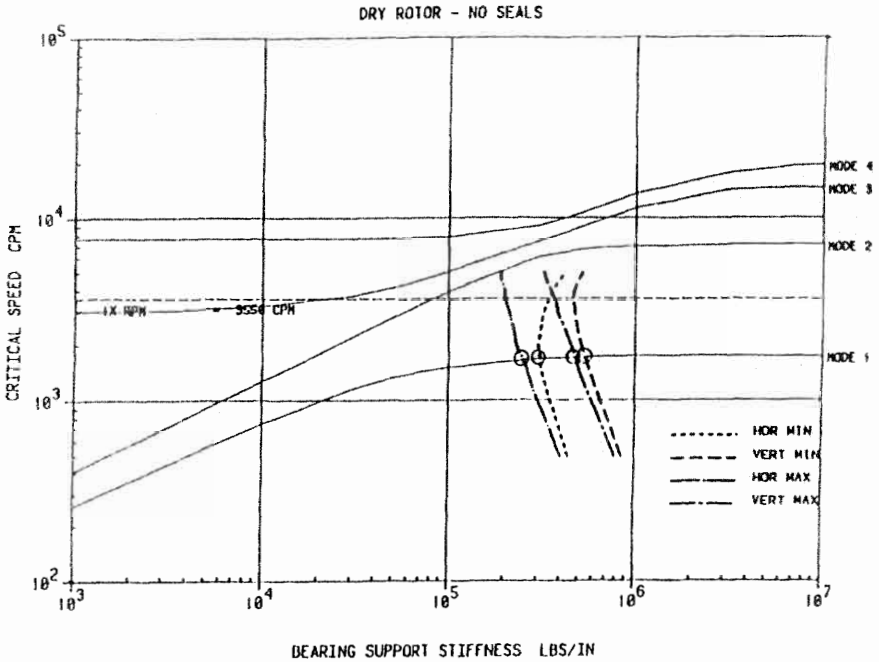


Figure 18-11. Eight-stage pump critical speed map—no seals.

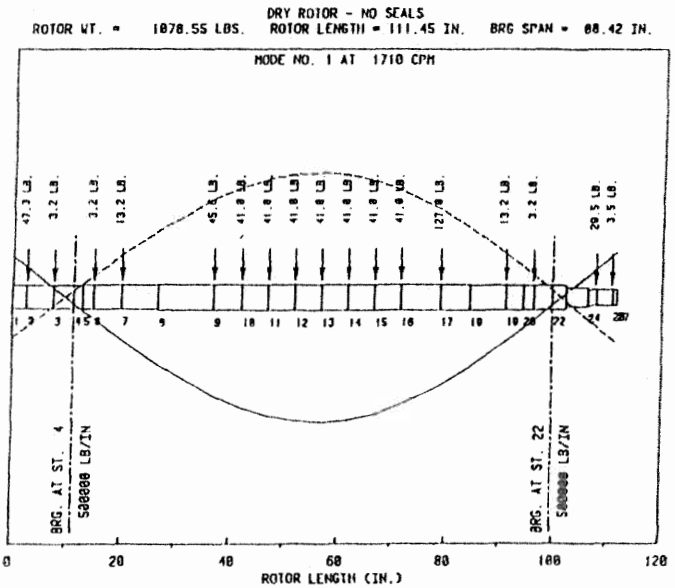


Figure 18-12. Eight-stage pump—first mode response—no seals.

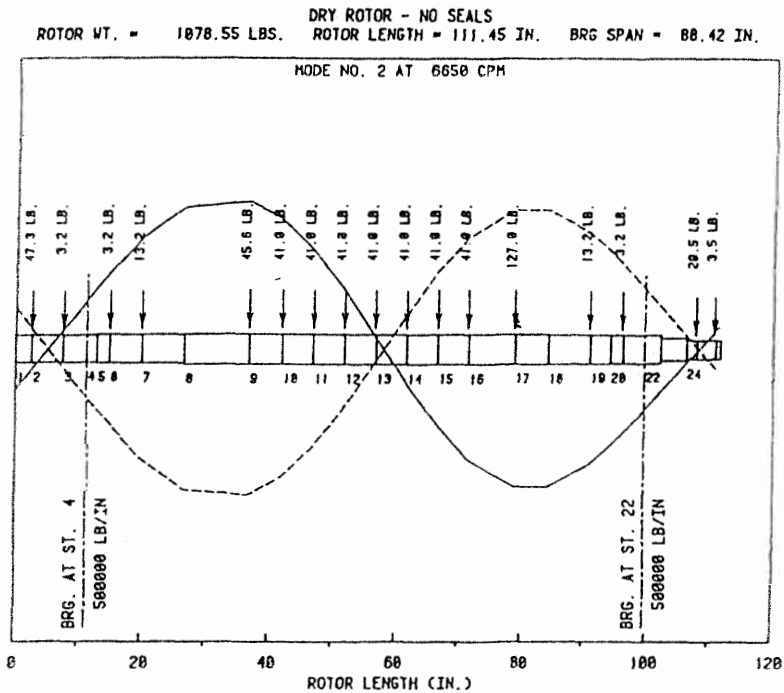


Figure 18-13. Eight-stage pump—second mode response—no seals.

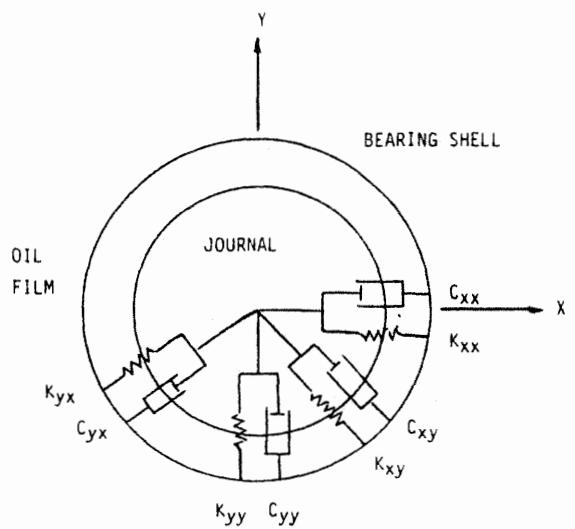


Figure 18-14. Hydrodynamic bearing stiffness and damping coefficients.

diameter, viscosity, load, speed, clearance, and the Sommerfeld number that is defined as:

$$S = \frac{\mu NDL}{W}\left(\frac{R}{C}\right)^2$$

μ = lubricant viscosity, lb-sec/in.
N = rotor speed, rev/sec
D = bearing diameter, in.
L = bearing length, in.
R = bearing radius, in.
W = bearing load, lbs
C = radial machined clearance, in.

The normal procedure in a design audit would be to calculate the bearing characteristics for the range of expected clearances, preload, and oil temperatures. The maximum clearance, minimum preload, and highest oil temperature usually define the minimum stiffness. The other extreme is obtained from the minimum clearance, maximum preload, and the coldest oil temperature. This will typically define the range of expected stiffness and damping coefficients for the bearings.

Preload is a configuration of the bearing clearances to promote a converging wedge of oil that increases the oil pressure and consequently the bearing stiffnesses. A preloaded bearing has its radius of curvature greater than the shaft radius plus the clearance.

Seal Effects

The critical speed map for a eight-stage pump, including the effects of seal and bearing stiffness, is given in Figure 18-15. Even though the bearings and seals add considerable cross-coupling and damping, it is still desirable to generate an undamped critical speed map to establish the range of the undamped (dry) critical speeds.

Adequate experimental data exists that documents that the analytical procedures used for simulating rotor response and stability for compressors and turbines can accurately predict critical speeds and potential instabilities from the design information. This is not true for pumps, especially for pumps that use grooved seals, labyrinth seals, or screw type seals with several leads. The accurate prediction of the stiffness and damping properties of seals for different geometries and operating conditions is a subject of on-going research [22, 27, 28]. The basic theories presented by Black [29] have been modified to account for finite length

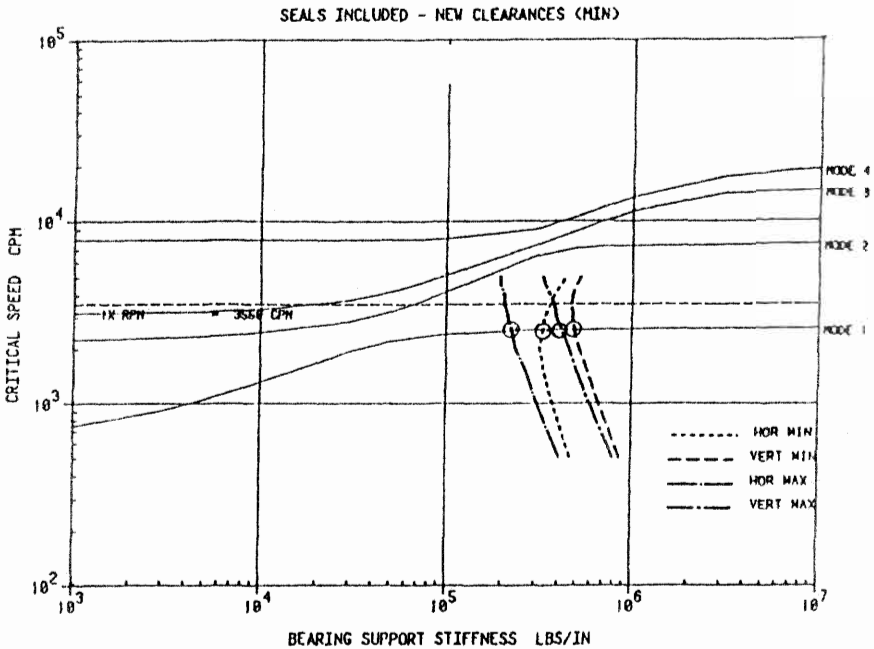


Figure 18-15. Eight-stage pump critical speed map—seals included.

seals, inlet swirl, groove, and other important parameters. However, a universally accepted procedure to accurately predict seal properties is not available for all the seal types that are in use today. If the seal effects are not correctly modeled, calculated critical speeds can be significantly different from actual critical speeds.

The forces in annular pressure seals can have a significant effect on the vibration characteristics of a pump rotor. The hydrodynamic and hydrostatic forces involved can significantly affect unbalanced response characteristics. The fluid film interaction with the shaft and the pressure drop across the seal give rise to a load capacity and a set of dynamic stiffness and damping coefficients similar to those used to represent the oil film in journal bearings.

Unlike hydrodynamic bearings, seals develop significant direct stiffness in the centered, zero-eccentricity position due to the distribution of the axial pressure drop between the inlet losses and an axial pressure gradient due to friction losses. The cross-coupled stiffnesses arise due to fluid rotation (swirl) within the seal. As a fluid element proceeds axially along an annular seal, shear forces at the rotor accelerate or decelerate the fluid tangentially until an asymptotic value is reached. For a seal with the same directionally homogeneous surface-roughness treatment on the

rotor and the housing, the average asymptotic tangential velocity is $R\omega/2$, where R is the seal radius and ω is the rotor running speed. The cross-coupled stiffness coefficient (K) acts in opposition to the direct damping coefficient (C) to destabilize rotors. Hence, steps that can be taken to reduce the net fluid rotation within a seal will improve rotor stability [30] by reducing K.

Childs has defined the dynamic seal coefficients for plain short seals directly from Hirs' lubrication equations [27] and has included the influence of fluid inertia terms and inlet swirl. His assumptions are less restrictive than previous derivations. The derived coefficients are in reasonable agreement with prior results of Black and Jenssen.

Childs [28] has extended the analysis to include finite-length seals. This analysis includes variable inlet swirl conditions (different from $R\omega/2$) and considers variations in the axial and circumferential Reynolds numbers due to changes in clearances.

A combined analytical-computational method has also been developed by Childs [28] to calculate the transient pressure field and dynamic coefficients for interstage and neck ring seals of multistage centrifugal pumps. The solution procedure applies to constant-clearance or convergent-tapered geometries that may have different surface-roughness treatments of the stator or rotor seal elements. The method has been applied to the calculation of "damper" seals as described by von Pragenau [31] and several roughened stator designs, such as knurled-indentation, diamond-grid post pattern, and round-hole pattern, have been tested. These procedures can be used to calculate serrated or grooved seals of various geometries.

Critical Speed Map—Considering Seals. The seal configurations at the balance piston, neck ring, and interstage bushings of the eight-stage pump are shown in Figure 18-16. The critical speed map (Figure 18-15) includes the support stiffnesses of the neck ring seals and interstage bushings combined at each impeller. The seal stiffness and damping coefficients are listed in Table 18-2 for nominal clearances. Note that a negative principal stiffness (K) value is predicted for the balance piston.

For this analysis the pump rotor was analyzed as if it had 11 bearings consisting of two tilted-pad bearings, the balance piston, and eight seals located at the impellers. For the purposes of developing the critical speed map, the seal stiffness values were held constant at their maximum levels (minimum clearances) that represent new seals.

The lateral mode shape of the first critical speed including these seal effects is shown in Figure 18-17. A bearing stiffness value of 500,000 lb/in. was again used for the mode shape calculations. Comparing Figure 18-17 with Figure 18-12, it is seen that the seals increase the frequency of

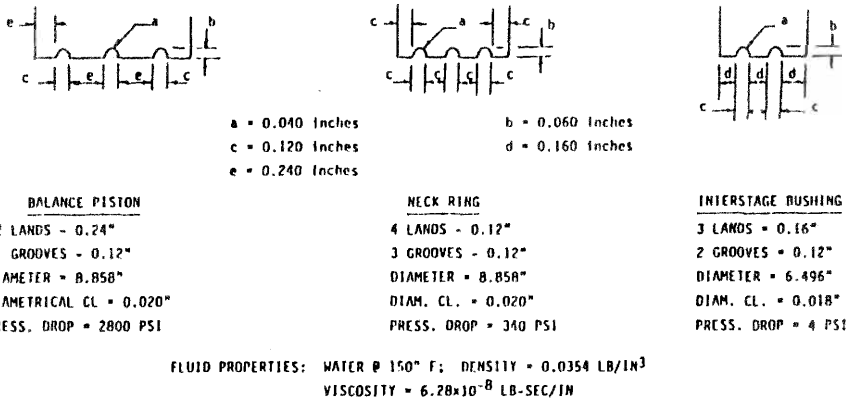


Figure 18-16. Eight-stage pump seal geometries and fluid properties.

Table 18-2
Summary of Seal Coefficients

	Stiffness		Damping	
Seal Type	Principal K–lb/in	Cross-Coupled K–lb/in	Principal C–lb sec/in	Cross-Coupled C–lb sec/in
Neck-Ring	13,400	4900	23	0
Int-Stg Bush	-8	370	3	0
Balance Piston	-271,000	627,000	25,000	3000

the first mode without altering the mode shape significantly. For an assumed bearing stiffness of 500,000 lb/in., the first, second, and third critical speeds from Figure 18-15 are 2570, 7120, and 8830 cpm.

Evaluation of Critical Speed Calculations. To summarize, in the evaluation of the adequacy of the rotor from the critical speed map and the mode shapes, the following items should be examined [18]:

- *The proximity of the critical speed to running speed or speed range.* The undamped lateral critical speeds should not coincide with the running speed. Various codes [7] address the allowable margin between lateral critical speeds and exciting frequencies. To determine if the actual critical speed will cause excessive vibrations, a rotor response to unbalance analysis should be performed.

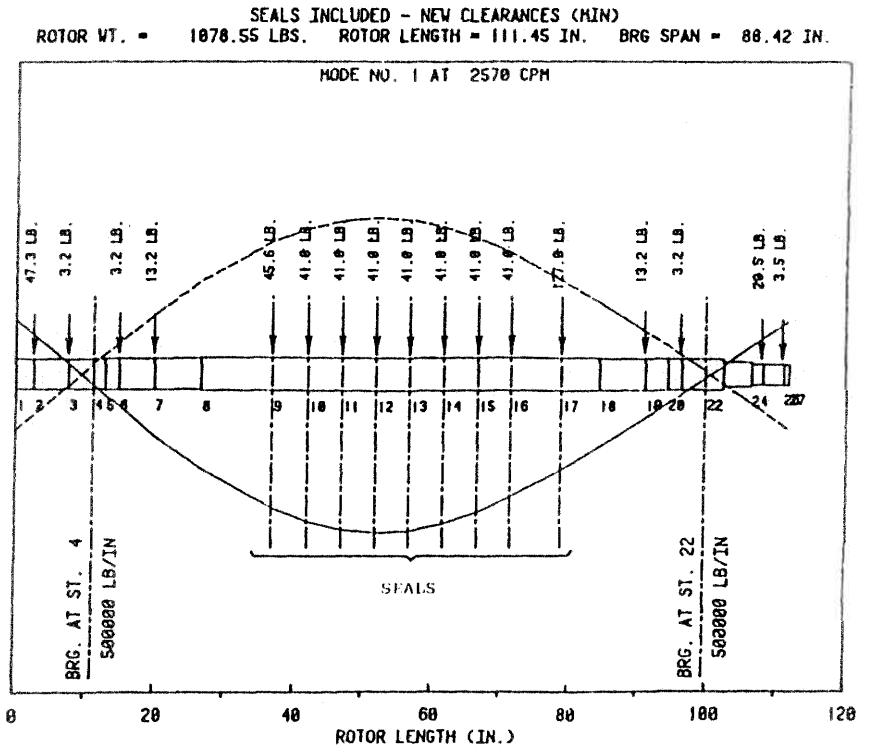


Figure 18-17. Eight-stage pump—first mode response—seals included.

- *The location of the critical speed relative to the support stiffness.* If the critical speed is near the rigid bearing criticals (flexible shaft region), increasing the bearing stiffness will not increase the critical speed because the weaker spring controls the resonant frequency. Vibration amplitudes may be low at the bearings (first mode), and therefore, low damping will be available. This can contribute to rotordynamic instabilities that will be discussed later. If the critical speeds are in the area of low support stiffness (stiff shaft region), the critical speeds are strongly dependent upon the bearing stiffness and damping characteristics and the critical speeds will be dependent upon bearing clearance. Bearing wear could be a significant problem.
- *The mode shape of the critical speed.* The mode shapes are used to assess the response of the rotor to potential unbalances. For example, a rotor that has a conical whirl mode (second critical) would be sensitive to coupling unbalance, but not strongly influenced by midspan unbalance.

Response To Unbalance

The location of a pump critical speed is defined by its response to unbalance. It is important to recognize the difference between critical speeds excited by unbalance and damped eigenvalues that are frequently also called critical speeds [32]. Generally, the effect of damping is to raise the frequency of the critical speed response due to unbalance; however, the effect of damping on the damped eigen values is to lower the frequency. The damped eigenvalues are primarily used for evaluating the stability of the rotor system. For compressors and turbines with tilting pad bearings, the damped eigenvalues are usually comparable to the unbalanced response criticals. However, in a pump with a large number of seals, the added damping to the system can be considerable, resulting in large differences in the unbalanced response critical speeds and the damped eigenvalues.

Rotor unbalance response calculations are the key analysis in the design stage for determining if a pump rotor will be acceptable from a dynamics standpoint. An accurate prediction of rotor unbalanced response is difficult for centrifugal pumps because of the sensitivity to bearing and seal clearances that may be at the tight or loose end of the tolerance range.

Computer programs are available that can calculate the elliptical shaft orbit at any location along the length of a rotor for various types of bearings, pedestal stiffnesses, pedestal masses, seals, labyrinths, unbalance combinations, etc. These programs can be used to determine the response of the installed rotor to unbalance and accurately predict the critical speeds over the entire range of variables. The actual critical speed locations as determined from response peaks caused by unbalance are strongly influenced by the following factors [33]:

- bearing direct stiffness and damping values
- bearing cross-coupled stiffness and damping values
- location of the unbalance
- location of measurement point
- bearing support flexibility

The normal unbalance used in the analysis would produce a force equal to 10% of the rotor weight at operating speed. Usually the rotor response to unbalance calculations are independently made for midspan unbal-

ance, coupling unbalance, and moment unbalance. An unbalance equal to a force of 5% of the rotor weight is usually applied at the coupling to excite the rotor. For moment unbalance, an unbalance equal to 5% of the rotor weight is used at the coupling and another equal unbalance is used out-of-phase on the impeller furthest from the coupling.

The unbalance response of a pump should be analyzed for several cases to bracket the expected range of critical speeds. The first analysis should include minimum seal and bearing clearances that represent the maximum expected support stiffness and therefore, the highest critical speed. The second analysis should consider the maximum bearing clearances and seal clearances of twice the design clearance to simulate the pump condition after long periods of service. The third analysis should simulate the worn condition with no seal effects and maximum bearing clearances that represent the overall minimum expected support stiffness for the rotor (lowest critical speed).

The unbalanced response of the eight-stage pump with maximum bearing clearances and no seals is shown in Figure 18-18. The peak response at 1700 rpm was the minimum calculated critical speed. The unbalance was applied at the rotor midspan to excite the first mode. The response for the intermediate analysis (worn seals) is plotted in Figure 18-19. The worn seals increased the predicted response peak to approximately 1800 cpm. With minimum clearances at the bearings and seals, the response was lower and the frequency increased to 2200 cpm, as shown in Figure 18-20 (note scale changes).

Shop acceptance test data was available for the eight-stage pump which was analyzed. The calculated unbalanced response is compared with the measured vibration data from the test stand in Figure 18-21. For these calculations, maximum bearing clearance and the design values of seal clearance were used. Based on these results, Childs' finite length method [28] provides favorable results compared with measured data. The "shape" of the response curve using Childs' seal values compares closely with the measured results, indicating that the damping contribution of the seals is of the right magnitude.

The anticipated range of rotor response should be calculated with the range of bearing values and various combinations of unbalance. This is important because it is not possible in the design stage to know the exact installed configuration with regard to bearings (clearance, preload) and balance (location of unbalance). Usually a mechanical test will be limited to one configuration (clearance, preload, unbalance) that may not show any problem. Changes introduced later by spare parts during maintenance turnarounds may change sensitive dimensions that may result in a higher response. For this reason, the vibration characteristics of some satisfactorily operating machines may change after an overhaul.

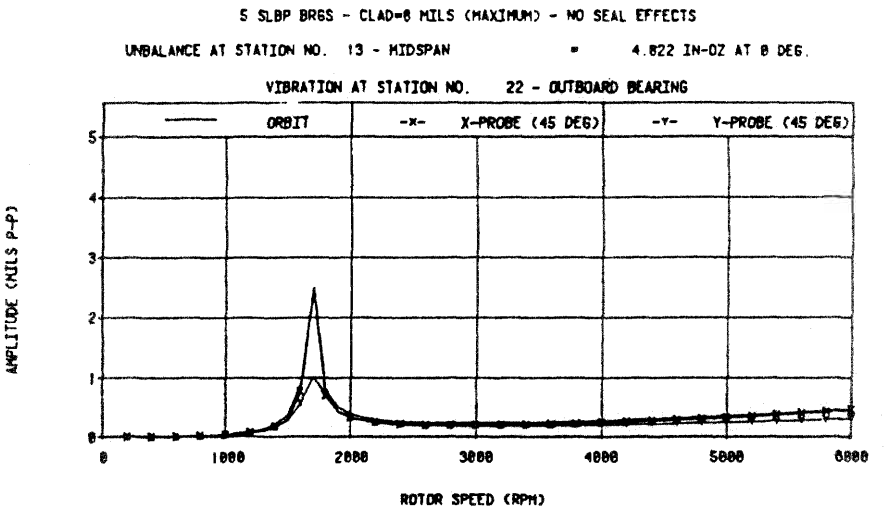


Figure 18-18. Unbalance response at outboard bearing with API unbalance at midspan—no seals.

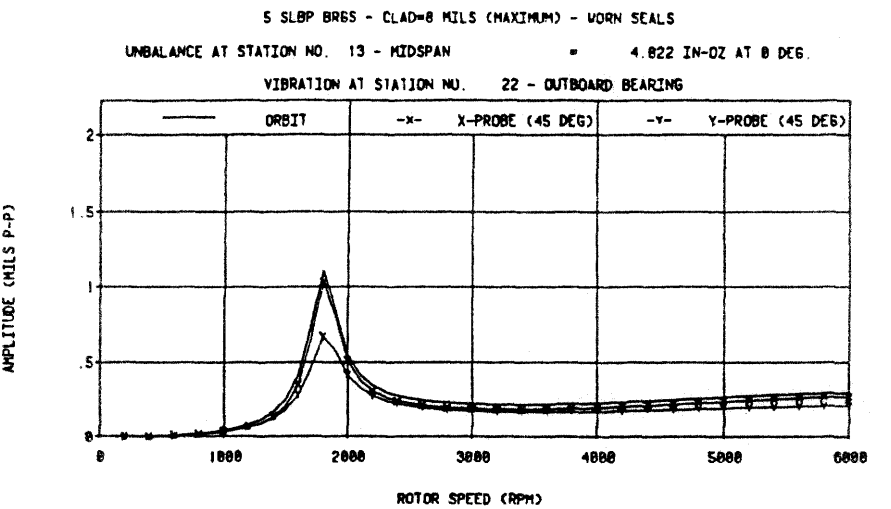


Figure 18-19. Unbalance response at outboard bearing with API unbalance at midspan—maximum clearance seals.

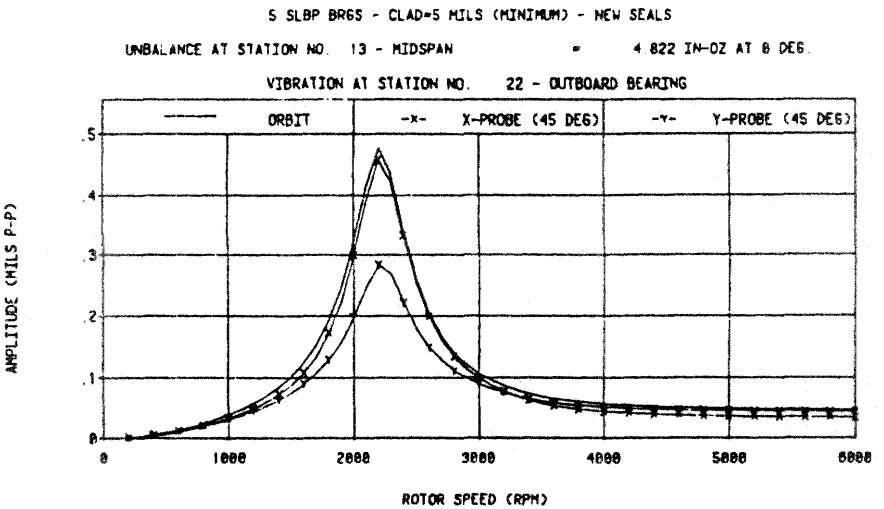


Figure 18-20. Unbalance response at outboard bearing with API unbalance at midspan—minimum clearance seals.

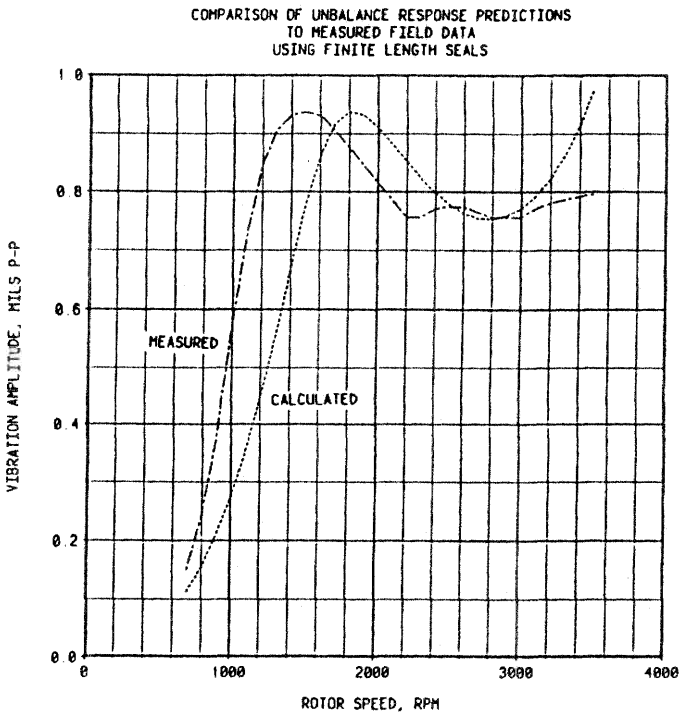


Figure 18-21. Comparison of calculated and measured response with seals.

Acceptable Unbalance Levels

Various engineering organizations have set forth criteria for allowable residual unbalance. The Acoustical Society of America defines balance quality grades for various types of rotors as described in Table 18-3. Pumps may have a range of balance quality grade from G2.5 to G6.3 depending upon size. The ASA Standard 2-1975 defines maximum residual unbalance [34] that is dependent upon speed as shown in Figure 18-22. For example, a 3,600 rpm pump with a rotor weight of 1,000 lbs for balance grade G2.5 would have an allowable residual unbalance of 4.5 in.-oz.

The revised API-610 (seventh edition, 1989) specifies an allowable residual unbalance for centrifugal pumps. The maximum allowable residual unbalance per plane (journal) may be calculated by the following formula:

$$U_b = \frac{4W}{N_{mc}}$$

where U_b = allowable unbalance, inch-ounces
W = journal static weight load, lbs
N_{mc} = maximum continuous speed, rpm

The total allowable unbalance (two planes near the journals) for a 3,600 rpm pump with a 1,000 lb rotor would be 1.1 in.-oz. This calculated residual unbalance from the current edition of the API code is significantly less than the allowable unbalance from earlier editions of the API codes for speeds less than 10,000 rpm. The previous edition of API-610 simply specified dynamic balance for all major rotating components with no specific value for an allowable unbalance. The various balance criteria are compared in Figure 18-23 for a 1,000 lb rotor.

Allowable Vibration Criteria

It is difficult to define the absolute maximum vibration level that can be tolerated without damage to the rotor. Some allowable vibration criteria are based on bearing housing vibrations. With rolling element bearings, the ratio of shaft vibrations to case vibrations is close to unity. The API-610 [7] specifies that the unfiltered vibration measured on the bearing housing should not exceed a velocity of 0.30 ips (inches per second) nor exceed a displacement of 2.5 mils peak-peak at rated speed and capacity $\pm$ 10%.

Table 18-3
Balance Quality Grades for Various Groups of Representative Rigid Rotors

Balance quality grades G	$s\omega$[a,b] (mm/sec)	Rotor types—General examples
G 4 000	4 000	Crankshaft-drives[c] of rigidly mounted slow marine diesel engines with uneven number of cylinders.[d]
G 1 600	1 600	Crankshaft-drives of rigidly mounted large two-cycle engines.
G 630	630	Crankshaft-drives of rigidly mounted large four-cycle engines. Crankshaft-drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft-drives of rigidly mounted fast four-cylinder diesel engines.[d]
G 100	100	Crankshaft-drives of fast diesel engines with six or more cylinders.[d] Complete engines (gasoline or diesel) for cars, trucks, and locomotives.[e]
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts Crankshaft-drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders.[d] Crankshaft-drives for engines of cars, trucks, and locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft-drives of engines with six or more cylinders under special requirements. Slurry or dredge pump impeller.
G 6.3	6.3	Parts or process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembled aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements.
G 2.5	2.5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbo-generator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine-driven pumps.
G 1	1	Tape recorder and phonograph (gramophone) drives. Grinding-machine drives. Small electrical armatures with special requirements.
G 0.4	0.4	Spindles, disks, and armatures of precision grinders. Gyroscopes.

[a] $\omega = 2\pi n/60 \approx n/10$, if n is measured in revolutions per minute and ω in radians per second.
[b] In general, for rigid rotors with two correction planes, one-half of the recommended residual unbalance is to be taken for each plane, these values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. (See Ses. 3.2 and 3.4.) For disk-shaped rotors the full recommended value holds for one plane. (See Sec. 3.)
[c] A crankshaft-drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc. (See Sec. 3.5.)
[d] For purposes of this Standard, slow diesel engines are those with a piston velocity of less than 9 m/sec; fast diesel engines are those with a piston velocity of greater than 9 m/sec.
[e] In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft-drive described in Note c above.

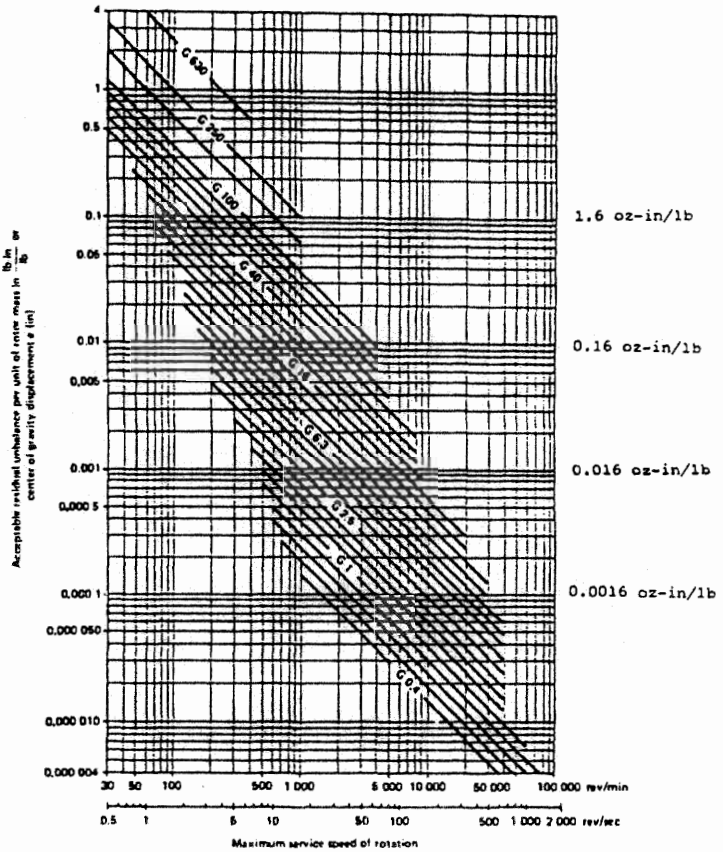


Figure 18-22. Allowable residual unbalance for industrial machinery ASA Standard 2-1975 [34].

However, with oil film bearings, the clearance between the shaft and bearings and the damping of the oil reduces the vibrational force transmitted to the case. For rotors with oil film bearings, shaft vibrations are a better indicator of unbalance conditions. The API-610 allowable vibrations for pumps with sleeve bearings (oil film) are based on shaft vibration measured at rated speed and capacity. The API-610 specifies the allowable unfiltered vibration shall not exceed a velocity of 0.40 ips nor exceed a displacement of 2.0 mils peak-peak including shaft runout.

For critical installations, vibration monitoring equipment should include non-contacting proximity probes (vertical and horizontal) at each

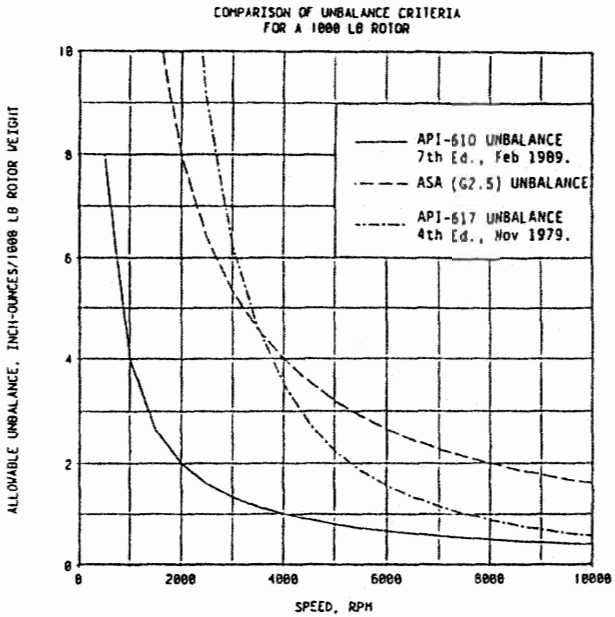


Figure 18-23. Comparison of residual unbalance criterion.

bearing. These vibrations should be continuously monitored with provision made for automatic alarm and shutdown capabilities to protect the installation from damage.

For oil film bearings, the following guidelines for defining maximum acceptable vibration levels are sometimes used when code allowables do not apply. If the vibrations are less than one-fourth of the diametrical bearing clearance, then the vibrations may be considered acceptable. Vibration amplitudes (A, peak-to-peak) greater than one-half the diametrical clearance (C_d) are unacceptable, and steps should be taken to reduce them.

$$A < \frac{C_d}{4} \text{ acceptable}$$

$$\frac{C_d}{4} < A < \frac{C_d}{2} \text{ marginal}$$

$$A > \frac{C_d}{2} \text{ unacceptable}$$

As with most experience-based criteria, these allowable amplitudes are based upon the synchronous vibration component only. Many manufacturers however, still assess acceptable vibrations on their equipment by case or bearing housing vibrations. In plant preventative maintenance programs, hand-held velocity pickups are commonly used to monitor vibrations.

Rotor Stability Analyses

Stability continues to be of major concern for rotors with hydrodynamic bearings, especially for high pressure pumps [36, 37]. In high performance pumps with vaned diffusers, large hydraulic forces can be exerted on the rotor at partial loads. The close internal clearances of pressure retaining seals create hybrid bearings that can produce a destabilizing effect.

Rotor instability occurs when the rotor destabilizing forces are greater than the rotor stabilizing forces. The destabilizing forces can be caused by: the bearings, seals, rotor unbalance, friction in shrink fits, or by loading effects such as inlet flow mismatching the impeller vane angle, turbulence at impeller tips, pressure pulsations, and acoustical resonances.

Instabilities in rotors can cause high vibrations with several different characteristics. They generally can be classified as bearing-related and self-excited. Oil whirl and half-speed whirl are bearing-related instabilities and are caused by the cross-coupling from the bearing stiffness and damping in fixed geometry bearings. Half-speed whirl will result in rotor vibrations at approximately one-half of the running speed frequency. Oil whirl describes a special type of subsynchronous vibration that tracks approximately half-speed up to the point where the speed is two times the first critical. As the speed increases, the subsynchronous vibration will remain near the first critical speed. These types of instabilities can generally be solved by changing the bearing design to a pressure dam, elliptical, offset-half bearing, or a tilting pad bearing.

Self-excited instability vibrations can occur on any rotor, including those with tilted pad bearings. The vibrations will usually occur near the rotor first critical speed or may track running speed at some fractional speed. These types of instability vibrations are sometimes called self-excited vibrations because the motion of the rotor creates the forcing mechanism that causes the instability.

The predominant method used in performing a stability analysis [38] is to calculate the damped (complex) eigenvalues and logarithmic decrement (log dec) of the rotor system including the bearings and seals. The log dec is a measure of the damping capability of the system to reduce

vibrations by absorbing some of the vibrational energy. A positive log dec indicates that a rotor system can damp the vibrations and remain stable, whereas a negative log dec indicates that the vibration may actually increase and become unstable. Experience has shown that due to uncertainties in the calculations, the calculated log dec should be greater than + 0.3 to ensure stability. The damped eigenvalue and log dec are sometimes plotted in a synchronous stability map. The frequency of damped eigenvalues is generally near the shaft critical speeds; however, in some heavily damped rotors it can be significantly different from the unbalanced response.

Rotor stability programs are available that can model the rotor stability for most of the destabilizing mechanisms; however, some of the mechanisms that influence it are not clearly understood. It has been well documented that increased horsepower, speed, discharge pressure, and density can cause a decrease in the rotor stability. Many rotors that are stable at low speed and low pressure become unstable at higher values. To predict the stability of a rotor at the design operating conditions, the rotor system is modeled and the log dec is calculated as a function of aerodynamic loading.

Torsional Critical Speed Analysis

All rotating shaft systems have torsional vibrations to some degree. Operation on a torsional natural frequency can cause shaft failures without noticeable noise or an obvious increase in the lateral vibrations. In geared systems, however, gear noise may occur that can be a warning of large torsional oscillations. Therefore, it is important to ensure that all torsional natural frequencies are sufficiently removed from excitation frequencies.

A torsional analysis should include the following:

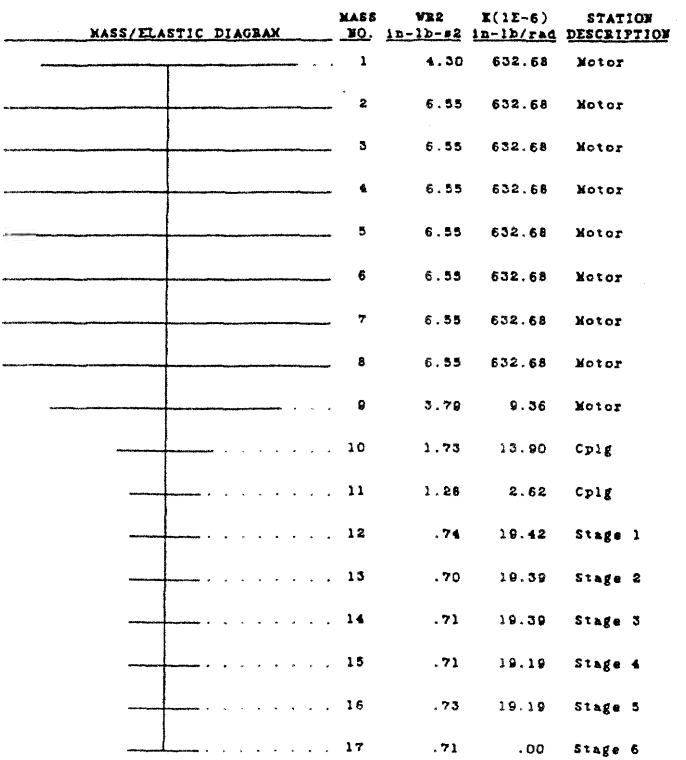
- Calculation of the torsional natural frequencies and associated mode shapes.
- Development of an interference diagram that shows the torsional natural frequencies and the excitation components as a function of speed.
- Calculation of the coupling torques to ensure that the coupling can handle the dynamic loads.
- Calculation of shaft stresses, even if allowable margins are satisfied.
- Calculation of transient torsional stresses [39] and allowable number of starts for synchronous motor drives.

Torsional natural frequencies are a function of the torsional mass inertia and the torsional stiffness between the masses. The natural frequen-

cies and mode shapes are generally calculated by the Holzer method or by eigenvalue-eigenvector procedures [40]. Either of the methods can give accurate results. A good design practice would be to locate the torsional natural frequencies a minimum margin of 10% from all potential excitation frequencies.

An example of the mass-elastic diagram of a torsional system of a 3,600 rpm motor-driven, six-stage pipeline pump is given in Figure 18-24. The natural frequencies and mode shapes associated with the first four natural frequencies are given in Figure 18-25. The mode shapes can be used to determine the most influential springs and masses in the system. This information is important if a resonance is found near the operating speed and system changes must be made to detune the frequencies.

SIX STAGE CENTRIFUGAL PUMP
1750 HP, 3600 RPM
GEAR TYPE COUPLING



MASS/ELASTIC DIAGRAM	MASS NO.	WR2 in-lb-s2	K(1E-6) in-lb/rad	STATION DESCRIPTION
	1	4.30	632.68	Motor
	2	6.55	632.68	Motor
	3	6.55	632.68	Motor
	4	6.55	632.68	Motor
	5	6.55	632.68	Motor
	6	6.55	632.68	Motor
	7	6.55	632.68	Motor
	8	6.55	632.68	Motor
	9	3.79	9.36	Motor
	10	1.73	13.90	Cplg
	11	1.28	2.62	Cplg
	12	.74	19.42	Stage 1
	13	.70	19.39	Stage 2
	14	.71	19.39	Stage 3
	15	.71	19.19	Stage 4
	16	.73	19.19	Stage 5
	17	.71	.00	Stage 6

Figure 18-24. Mass-elastic diagram of six-stage pump train.

Parametric variations of the coupling stiffness should be made if changes are necessary, because most torsional problems can be solved by coupling changes.

An interference diagram for the six-stage pipeline pump is given in Figure 18-26. In this system, excitation by several orders is possible as the pump is started; however, operation at 3,600 rpm has an adequate margin from the critical speeds. Once the system has been modeled and the natural frequencies have been determined, potential forcing functions should be identified. The forcing functions represent dynamic torques applied at locations in the system that are likely to generate torque varia-

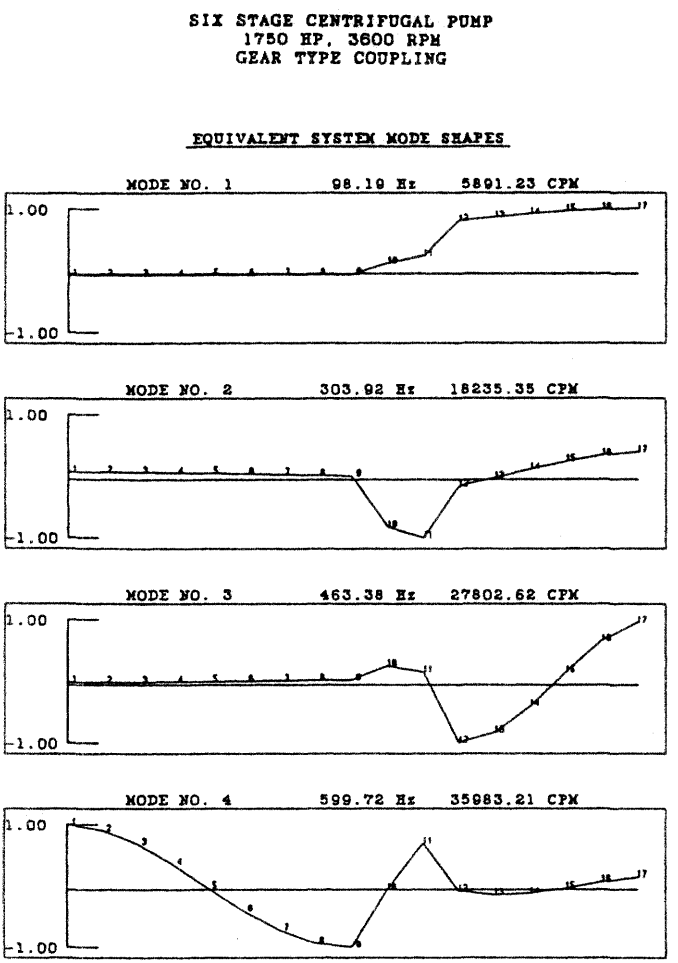


Figure 18-25. Torsional resonant mode shapes of six-stage pump train.

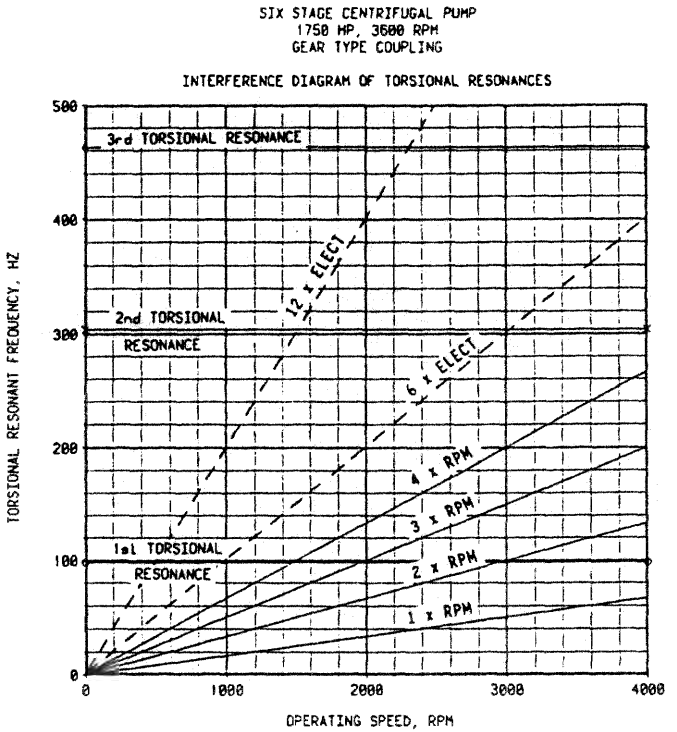


Figure 18-26. Interference diagram of torsional resonances of six-stage pump train.

tions. Identification of all possible sources of dynamic energy is an important step in diagnosing an existing vibration problem or avoiding problems at the design stage.

The most likely sources of dynamic torques include the following:

- Pumps, turbines, and compressors
- Motors (synchronous and induction)
- Couplings
- Gears
- Fluid interaction (pulsations)
- Load variations

The transient torques of a synchronous motor were measured during startup (Figure 18-27) by attaching a strain gauge to the motor shaft and obtaining the signal with an FM telemetry system. A synchronous motor produces a pulsating torque [39] that varies from twice line frequency as

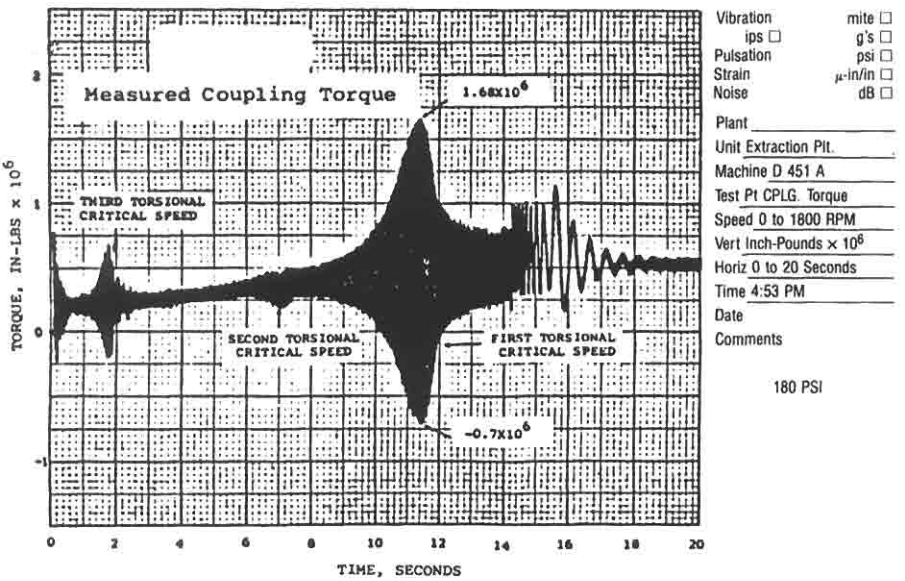


Figure 18-27. Measured coupling torque during synchronous motor startup.

it starts to zero frequency when it is synchronized with the line at operating speed.

Reliability Criteria. The overall system reliability depends upon the location of the torsional natural frequencies with regard to the potential excitation frequencies. An interference diagram generated for each system helps to identify coincidences between expected excitation frequencies and torsional natural frequencies within the operating speed range. Whenever practical, the coupling torsional stiffness and inertia properties should be selected to avoid any interferences within the desired speed range. If resonances cannot be avoided, coupling selection can be optimized based on torsional shaft stress calculations as well as location of critical speeds.

When coupling changes are implemented that have different weights than the vendor originally specified, the lateral critical speeds may be affected. Generally, heavier couplings lower the lateral critical speeds, while lighter couplings raise them. The lateral critical speeds should be reviewed to evaluate the possibility of operating near a lateral response that could result in high radial vibrations.

The acceptability of the torsional system is determined by comparison to typical engineering criteria. Common criteria used for industrial machinery (API Standards) recommend separation of the torsional critical

speed and the frequency of all driving energy by a margin of 10%. The effect of the torque modulation on coupling life should be compared to the manufacturer's vibratory torque criterion.

Allowable Torsional Stresses. For long-term reliability, torsional shaft stresses should be compared to applicable criteria. The allowable shaft stress values given by Military Standard 167 [41] are appropriate for most rotating equipment. The Military Standard 167 defines an allowable endurance limit stress of 4,000 psi zero-peak or 8,000 psi peak-peak, based on 100,000 psi ultimate tensile strength shaft material. The general equation for allowable zero-peak endurance limit is the ultimate tensile strength divided by 25.

When comparing calculated stresses to allowable stress values, the appropriate stress concentration factor and a safety factor must be used. Generally a safety factor of 2 is used for fatigue analysis. The standard keyway (USA Standard ANSI B17.1) has a stress concentration factor of 3 [42]. When these factors are used, it can be shown that fairly low levels of torsional stress can cause failures. A typical torsional stress allowable thus becomes the ultimate tensile strength divided by 150.

To evaluate the stresses at resonance, the expected torsional excitation must be applied to the system. For systems with gear boxes, a torque modulation of 1%, zero-peak is a representative torque value that has proven to be appropriate for most industrial machinery trains. As a rule of thumb, excitations at the higher orders for gears are inversely proportional to the order numbers: the second order excitation is 0.5%, the third is 0.33%, etc.

The torque excitation should be applied at the appropriate location and the torsional stresses calculated on the resonant frequencies and at the running speed. An example of the stress calculations of the first natural frequency resonance for the six-stage pump are given in Table 18-4. The maximum stresses occurred at 2,946 rpm on startup as the second order matched the first critical speed. The stresses at 3,600 rpm were low because there was a margin of approximately 17% from the nearest natural frequency. The torque excitation at the second order would cause a maximum torsional stress of 395 psi peak-peak in shaft 11, which is the pump input shaft between the coupling and the first stage impeller. The dynamic torque modulation across the couplings was calculated for the applied input modulation. For this mode, the maximum torsional vibrations occur across the coupling and the dynamic torque modulation was 48 ft-lbs.

Table 18-4
Six-Stage Centrifugal Pump
1750 HP, 3600 RPM
Gear Type Coupling

Dynamic Torques (1 Percent Zero-Peak) Applied at Motor
Maximum Resultant Torsional Stresses at 2945.62 RPM

Shaft	Stress PSI P-P	SCF	Stress PSI P-P
1	.99	2.00	1.98
2	2.47	2.00	4.93
3	3.87	2.00	7.74
4	5.21	2.00	10.41
5	6.51	2.00	13.02
6	7.82	2.00	15.65
7	9.16	2.00	18.32
8	10.52	2.00	21.03
9	130.49	3.00	391.47
10	*DYNAMIC TORQUE VARIATION*		**48.32***
11	131.77	3.00	395.30
12	72.65	2.00	145.30
13	59.57	2.00	119.13
14	45.51	2.00	91.03
15	30.84	2.00	61.68
16	15.31	2.00	30.63

* —*Values are dynamic torque variation across coupling, ft-lbs*

Variable Speed Drives

Systems that incorporate variable frequency drives [43] require additional considerations in the design stage over conventional constant speed equipment. The wide speed range increases the likelihood that at some operating speed a coincidence between a torsional natural frequency and an expected excitation frequency will exist. In addition to the fundamental mechanical frequency (motor speed), excitation frequencies include the fundamental electrical frequency (number of pole pairs times motor speed) and the sixth and twelfth orders of electrical frequency [44]. The variable frequency power supply introduces a pulsating torque with strong sixth and twelfth order harmonics.

As an example, the same six-stage pump system (described in Figure 18-24) was analyzed for a variable frequency drive motor. The dashed line in Figure 18-26 represents the sixth harmonic speed line that shows that the sixth order would excite the first torsional natural frequency at 982 rpm.

Because of the strong sixth order electrical torques produced by the variable frequency drive, the stresses (Table 18-5) are significantly greater than the stresses for the same system with a constant speed motor (compare with Table 18-4). Even though the natural frequencies of the rotor system remain constant, the exciting torques are greater and occur at a different speed.

It is dffficult to remove all coincidence of resonances with the excitation sources over a wide speed range; therefore, stress calculations must be made to evaluate the adequacy of the system response. The input shaft stresses for the six-stage pump with a variable frequency drive are shown in Figure 18-28. They are significantly different stresses than for the constant speed motor. This example demonstrates that converting a constant speed system (motor or turbine) to a variable frequency drive motor must be considered carefully. Although the variable speed provides greater efficiency; adjustments may be required (i.e., proper choice of couplings) to obtain acceptable torsional response.

Some variable speed systems use couplings with flexible elements to increase the damping as critical speeds are passed. Several couplings in common use have rubber elements to add damping that increases stiffness with higher transmitted torque. Other couplings use flexible grids or springs that again have an increased stiffness with load. These couplings are sometimes necessary when several torsional frequencies occur within the speed range. The torsional stiffness of these nonlinear couplings changes with load (transmitted torque) and speed. This nonlinearity adds complexity to the analysis. The pump load must be considered which also varies as a function of speed squared.

Diagnosis of Pump Vibration Problems

Large plants that handle liquids (i.e., chemical plants) may have hundreds or more small pumps, making it almost impossible to measure each pump in detail on a regular basis. The cost of a detailed analysis with several transducers, spectrum analyses, and the necessary study of the data can quickly exceed the cost of repair or even replacement of some of the small pumps. Only the pumps in a critical system, which could affect production if a failure occurred, can afford the expense of a detailed analysis or a permanent monitoring system. However, the hundreds of small pumps need regular attention to keep plant efficiency (and flow rates) from dropping.

An effective preventative maintenance program should periodically measure and record the vibrations on each bearing of the pump and its driver. Measurements with a hand-held velocity transducer or data collector system are usually adequate to obtain periodic data to evaluate the

Table 18-5
Six-Stage Centrifugal Pump
1750 HP, 3600 RPM Variable Frequency Drive Motor
Gear Type Coupling

Dynamic Torques (1 Percent Zero-Peak) Applied at Motor
Maximum Resultant Torsional Stresses at 981.87 RPM

Shaft	Stress PSI P-P	SCF	Stress PSI P-P
1	48.12	2.00	96.25
2	121.31	2.00	242.62
3	193.97	2.00	387.93
4	265.92	2.00	531.84
5	336.94	2.00	673.89
6	406.63	2.00	813.26
7	474.66	2.00	949.32
8	540.76	2.00	1081.52
9	6676.68	3.00	20030.04
10	*DYNAMIC TORQUE VARIATION*		**2489.59***
11	6822.28	3.00	20466.84
12	3763.91	2.00	7527.81
13	3087.43	2.00	6174.85
14	2359.91	2.00	4719.81
15	1599.41	2.00	3198.81
16	794.30	2.00	1588.60

* — *Values are dynamic torque variation across coupling, ft-lbs*

vibrational trend. If the vibrations show an increasing trend, the unit should be monitored more frequently. Vibrational guidelines in common usage can aid in determining the severity or extent of damage so that maintenance can be scheduled. A vibrational velocity of less than 0.3 ips (inches per second) is generally accepted as satisfactory operation for pumps [7] and motors. Velocity levels above 0.3 ips are warnings of potential problems. Velocity levels of 0.5 ips and above may be indicative of significant damage to the bearings, seals, or rotor.

The basis of troubleshooting is obtaining test data on troublesome pumps during normal operation and comparing the data to the purchase specifications, vendor guarantees, or applicable vibration criteria. It is crucial to the success of the troubleshooting effort to have adequate instrumentation on the pump system. In many cases, the instrumentation can be temporarily added to obtain the required measurements.

Measurement Techniques

Typically pumps are installed with a minimum of vibration monitoring equipment. Some pumps may have a velocity pickup on each bearing

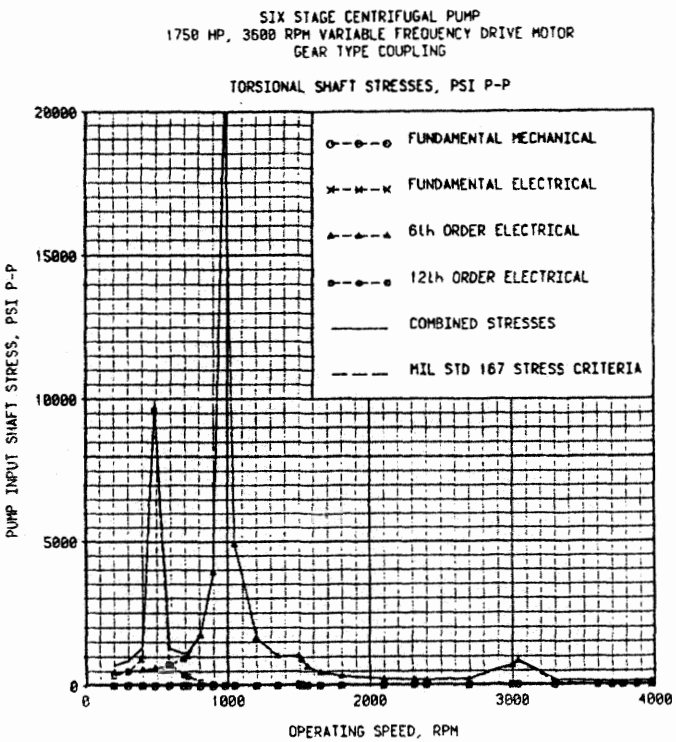


Figure 18-28. Pump shaft stress with variable frequency drive motor.

housing. These systems are satisfactory for pumps with rolling element bearings, because the bearings transmit the rotor forces directly to the case. However, additional instrumentation is often required to define difficult problems. For pumps with oil film bearings, it is desirable to have two proximity probes 90° apart near each bearing and an axial probe. Accelerometers and velocity probes attached to the bearing housings or case are often used to measure pump vibration; however, if the proximity probes are not installed, the data will be limited and may be a detriment to the diagnosis of some types of problems. A pressure transducer is vital to diagnostic work for measuring dynamic pulsations in the piping, impeller eye, diffuser, and across flow meters. Accurate flow measurements are necessary to define the pump vibration characteristics as a function of its location on the head-flow performance map.

Accelerometers or velocity transducers can be used to determine frequencies and amplitudes of the pump case or support structure during normal operation [45]. Two accelerometers and a dual channel oscilloscope displaying their complex vibration waveforms can be used to de-

fine the phase relationship between the signals. By keeping one accelerometer stationary as a reference, subsequent moves of the other accelerometer to measure amplitudes at various points on the structure can define the mode shape. This method requires that the speed remain constant while the measurements are being made. The speed should be set at the resonant mode to be identified.

As a general technique, this type of measurement should be taken at the resonant frequencies near the operating speed to define the components that control the resonance. Measurements can also be made by using a reference accelerometer on the structure to trigger the data loading sequence of a real time analyzer to enable more accurate amplitude/phase data to be taken. The reference signal may be from a key phase signal that relates the vibrational maximum to the actual shaft position.

Amplitude/phase data for a feedwater pump (Figure 18-29) is tabulated directly on the figure to aid in interpreting the mode shape. The vertical vibrations at the pump centerline are plotted in Figure 18-29a. Although the phase difference from the outboard and inboard ends was 165° (not 180°), the mode was characterized by a rocking motion about the pivotal axis. The vertical vibrations on the pump inboard (Figure 18-29b) indicate that the horizontal support beam was one of the main flexible elements.

Impact Tests

An impact test is a simple technique that can be used to excite resonances of flexible substructures. A significant advantage of the method is that mechanical natural frequencies can be measured while units are down. Typical instrumentation includes an accelerometer attached to the head of a rubber-tipped hammer to measure the impact force and a second accelerometer used to measure the response of the test structure. In modal analysis testing, the accelerometer remains at one location and an impact hammer is used to excite the structure at selected locations. The response signal can be automatically divided by the impact signal using a fast fourier transform (FFT) spectrum analyzer with transfer function (XFR) capabilities.

For best results, the impact velocity should not be greatly different from the vibrational velocity of the vibrating member. For example, the lowest beam mode of a piping span can be excited with a rubber mallet by applying the impact with a forceful, medium-speed swing. However, a sharp rap with a steel-faced hammer could produce higher modes of beam vibration or possibly a pipe shell wall resonance. These higher energy modes will be quickly damped and difficult to identify. Using modal

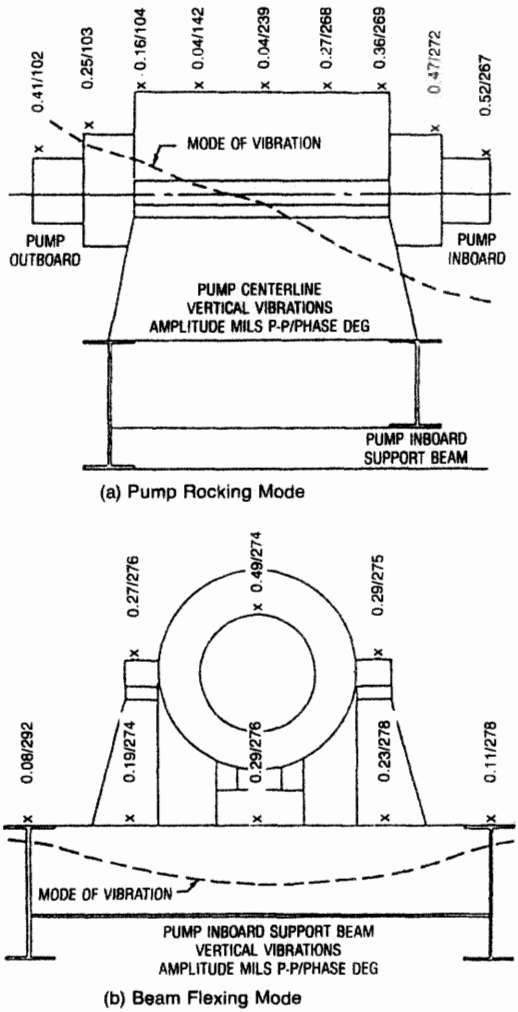


Figure 18-29. Pump vibrational mode shapes during operation.

analysis techniques, the input spectrum can be evaluated and the impact can be adjusted to obtain the best results for the structure being tested.

In nonsymmetrical members or structures, a resonant mode has a preferred direction of motion. Impact testing should be approached on an experimental basis by trying variations of impact velocity, direction, etc. Numerous frequencies may occur that make the mode shapes difficult to identify. Generally, the lower modes of vibration will be predominate. The mode of vibration is strongly influenced by the following:

- Direction of impact
- Interface material
- Impact velocity
- Contact time
- Impact location

The objective of the testing should be to identify structural resonant frequencies that occur within the machinery operating frequency range. In a complex system, there are many natural frequencies. Consequently, it may be difficult to identify the individual natural frequencies because the vibrations can be transmitted between the different elements (pump rotor, case, and support structure). The impact technique is useful for separating the structural natural frequencies because each element can be impacted individually. Individual resonances can generally be identified by analyzing the vibrational mode shapes and comparing the transfer function amplitudes.

The transfer function amplitude is an indication of the relative stiffness of a structure. For example, a rigid structure that is difficult to excite or resonate would have a low amplitude transfer function (response/force). A highly responsive structure would have a higher amplitude transfer function because less force is required to obtain the response. This technique was used to determine the bearing housing resonance of a pump that is described in the case histories that follow.

Various types of data presentation formats are in common use and are described in the case histories. The data format is dependent upon the type of instrumentation available; however, for convenience of comparison to criteria or specifications, some formats may be more convenient than others.

Troubleshooting

As examples of the types of diagnostic procedures that are used to identify the causes of some typical pump problems, several field case histories will be presented.

Bearing Housing Resonance of Feed Charge Pumps. High vibration amplitudes and seal failures occurred on one of a pair of centrifugal pumps that are the feed charge pumps for a cat feed hydro-treater plant. The pumps were driven by electric motors at a speed of approximately 3,587 rpm (59.8 Hz). The pumps have five impellers; the first two impellers have six vanes and the last three impellers have seven vanes.

Vibration data recorded on both pumps indicated that the vibration levels were significantly higher on the inboard bearing of Pump A. The maximum vibration occurred at seven times running speed (418 Hz) which is the vane passing frequency of the last three impellers.

The high vibration levels and seal failures on the inboard bearing of the pump could be caused by several problems, including:

- Pulsation at the vane passing frequency that can increase the forces on the impeller and shaft.
- Mechanical or structural natural frequencies that amplify the vibration levels.
- Misalignment between the motor and the pump or internal misalignment between the pump bearings.

A field test was performed to measure the vibrations and pulsations in the system. The vibration amplitudes were considerably higher on the inboard bearing of Pump A which experienced the seal failures. The vibration amplitude increased significantly between the case and the bearing housing. This differential vibration could be the cause of the seal failures. The bearing housing vibrations of Pump A were primarily at seven times running speed (418 Hz).

Vibration measurements were taken at four locations between the end of the case and the end of the bearing to illustrate the vibration mode shapes. The vibration amplitudes on the inboard bearing were considerably higher near the end of the bearing housing compared to near the case. The horizontal vibrational mode can be defined from the vibration spectra at the four points shown in Figure 18-30. As shown, there was significant differential motion between the case and the end of the flange and across the bolted flange. The vibration amplitude on the bearing housing was approximately 6 g peak-peak at 418 Hz (0.44 in./sec peak). Most allowable vibration criteria would consider these amplitudes to be excessive. However, the actual differential displacement was only 0.13 mils peak-peak. The amplitudes at the outboard bearing were lower compared to the amplitudes on the inboard bearing.

The discharge pulsation amplitudes were 12 psi at seven times running speed and 3 psi at fourteen times running speed. The discharge pulsations were higher on Pump B (without any seal failures), which indicated that the pulsations were not the cause of the increased vibrations.

Impact Tests

Impact tests on the bearing housings indicated that the inboard bearing housings were very responsive compared to the outboard bearing hous-

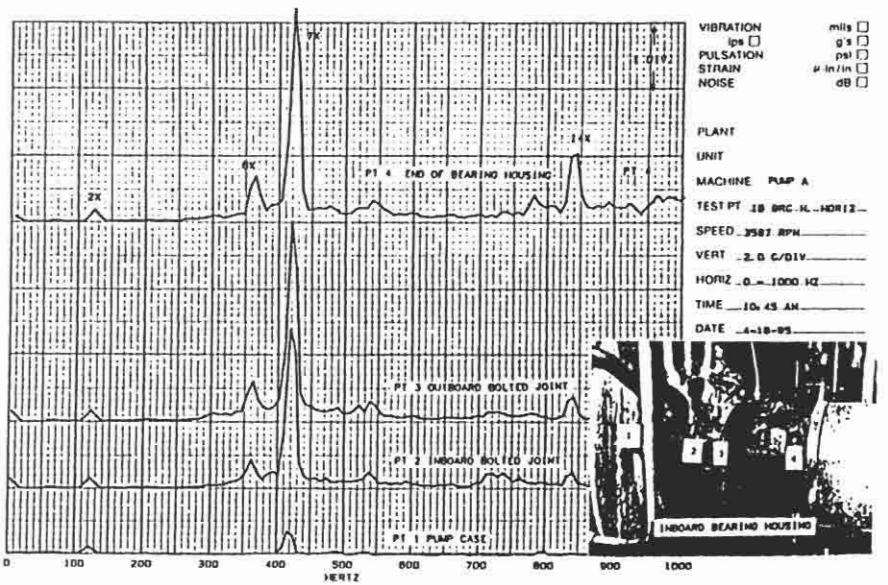


Figure 18-30. Vibrational response of inboard bearing housing.

ings. The transfer function (Figure 18-31) was plotted with a full scale amplitude equal to 0.4 (0.05 per major division on the graph paper).

The inboard bearing of Pump A had a major response at 420 Hz with a transfer function amplitude of 0.24. This frequency was almost coincident with seven times running speed (418 Hz) which would amplify the vibration levels at seven times running speed and appeared to be the primary cause of the high vibration at seven times running speed. The flange bolts were tightened and the frequency increased to 430 Hz which indicated that the response near 420 Hz was primarily associated with the bearing housing and its attachment stiffness to the case.

A similar major response near 432 Hz occurred in the vertical direction. The transfer function amplitude was lower and the frequency was higher than measured in the horizontal direction because the bearing was slightly stiffer in the vertical direction compared to the horizontal direction. When the flange bolts were tightened, the response near 432 Hz was increased to 440 Hz.

The outboard bearing was more massive and stiffer and not as responsive as the inboard bearing. There were no major responses in the horizontal direction near the excitation frequencies of interest. A vertical response was measured near 580 Hz; however, because it was above the frequencies of interest, it should not cause an increase in vibration on the bearing housings.

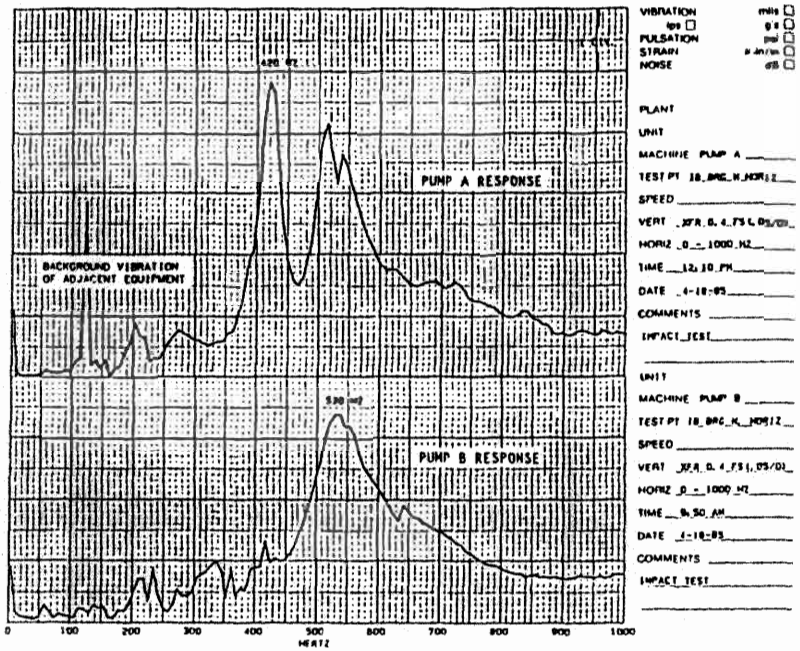


Figure 18-31. Structural natural frequency of inboard bearing housing.

Impact tests were performed on Pump B (no seal failures). The responses on the inboard bearing were 530 Hz in the horizontal direction (Figure 18-31) and 600 Hz in the vertical direction. Although visually the two pumps appeared to be identical, detailed measurements revealed slight differences in the cross-sectional thickness and lengths of the inboard support between the flange and the end of the cases. Apparently these slight dimensional differences caused the mechanical natural frequencies to vary considerably between the two pumps.

The obvious solution to the problem would be to replace the bearing housing casting or modify the inboard bearing housing support of Pump A. The mechanical natural frequency could be increased by stiffening the support with gussets. For improved reliability, the mechanical natural frequency should be increased approximately 10% above the primary excitation frequency at seven times running speed.

Pump Critical Speed Problem [46]. A critical speed analysis was performed on a centrifugal pump used as a jetting pump on a pipe laying barge. The predicted unbalanced vibration response of the centrifugal pump showing the effects of the seals is shown in Figure 18-32. Note that

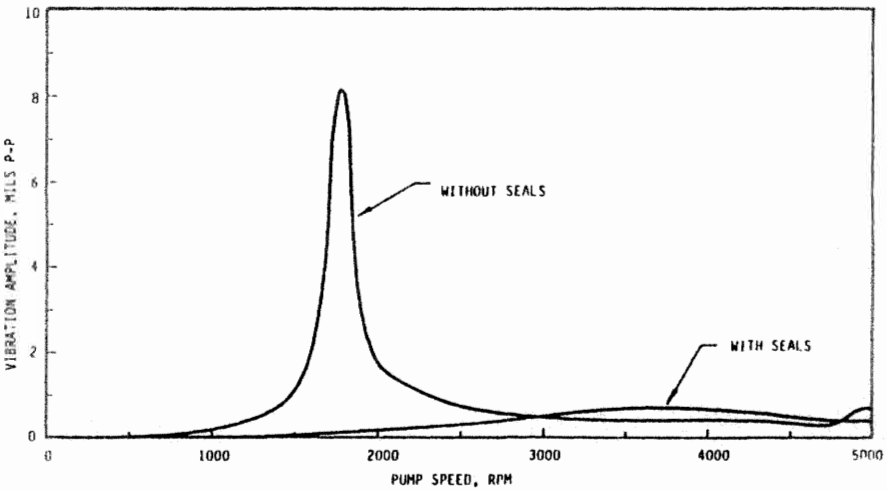


Figure 18-32. Predicted vibration of three-stage centrifugal pump showing effect of seals.

the seal effects shifted the critical speed from 1,800 to 3,700 rpm and significantly increased the damping, which illustrates that the pump critical speed should be sensitive to seal stiffness effects. When the seals were considered, the predicted amplitudes were reduced by a factor of more than 10 to 1.

The vibrations on the pump were measured (Figure 18-33) using the peak-store capabilities of a real-time analyzer. The vibrations were low until the pump reached 3,300 rpm and then sharply increased to 6 mils peak-peak at 3,600 rpm. The design speed was 3,600 rpm; however, the pump could not run continuously at that speed. The pump speed was kept below 3,400 rpm so that the vibrations were less than 2 mils peak-peak. Even with the reduced pump speed, the pipe-laying barge was able to set pipe-laying records.

After a year of operation, the pump vibrations began to increase until the vibrations at 3,400 rpm were unacceptable. The pressure breakdown bushing had worn which reduced its effective stiffness and the critical speed had dropped to 3,400 rpm. It was recommended that the pump be operated at a speed above the critical speed. This was tried and the pump operated at 3,600 rpm with vibration levels less than 2 mils peak-peak. The barge remained in service and reset the pipe-laying records during the next season.

The data analysis technique used to determine this critical speed was to use the peak-store capabilities of the real-time analyzer. An alternate method would be to use a tracking filter to determine the critical speed

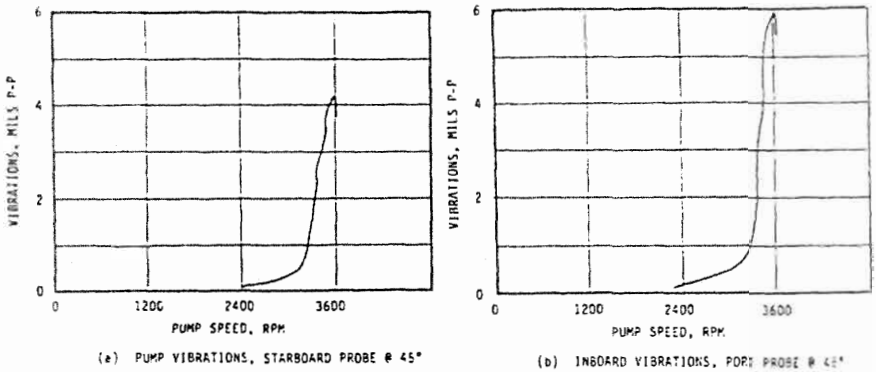


Figure 18-33. Jetting pump measured vibrations.

response because both the amplitude and phase data could be made available. Although it is generally better to have both the Bode and Nyquist plots, for this case they were not required to define the cause of the vibration plot.

High Vibrations of a Centrifugal Pump. Critical speed calculations were performed on a three-stage centrifugal pump to determine if a critical speed existed near running speed. The mode shape calculations for the first and second critical speeds are summarized in Figures 18-34 and 18-35. As discussed, the liquid seals can significantly affect the critical speeds of a pump; however, for this rotor, the seals only increased the critical speed by about 15%. This critical speed analysis considered a shaft with two bearings and the eight seals, or ten sets of stiffness and damping coefficients. Each of these coefficients varied as a function of speed and was included in the unbalanced response analysis (Figure 18-36). A comparison of predicted responses to measured test stand data is given in Figure 18-37. The agreement with the measured data was considered good, indicating that the rotordynamic model of the rotor and the bearing and seal clearances was acceptable.

A data acquisition system with a tracking filter was used to plot the amplitude and phase angle versus speed (Bode Plot). The lack of a significant phase shift through the critical speed could not be explained without a more detailed analysis. As in many field studies, other priorities prevailed, and additional tests were not possible. Although the critical speed was in the running speed range, the vibration amplitudes were low and any wear of the seals should move it further away from the rated speed of 3,600 rpm.

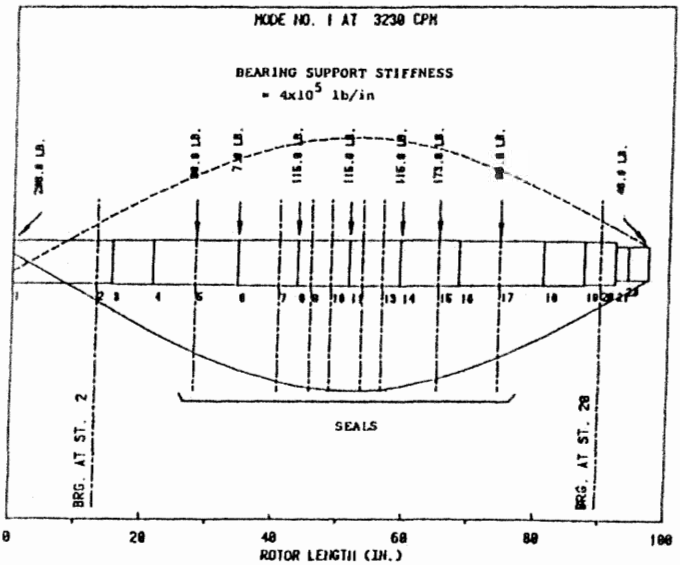


Figure 18-34. Three-stage pump—first mode response—with seals.

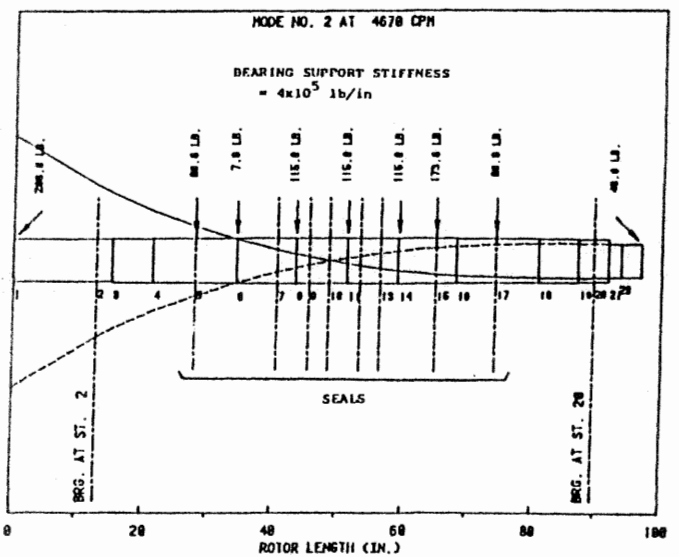
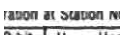
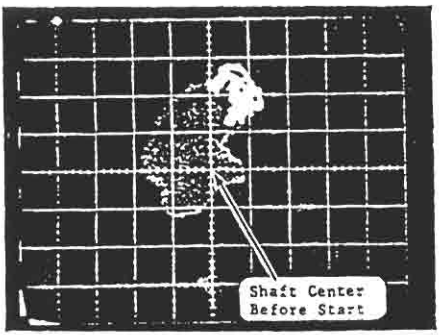
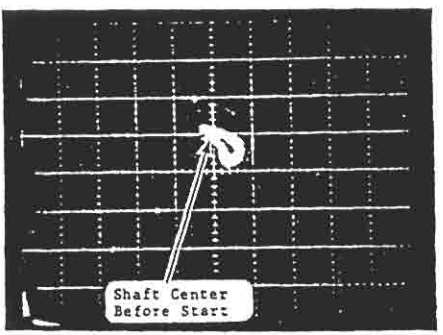


Figure 18-35. Three-stage pump—second mode response—with seals.





(a) PUMP SHAFT MIDSPAN SPATIAL POSITION



(b) PUMP SHAFT INBOARD SPATIAL POSITION

Figure 18-40. Pump shaft vibration orbits.

times as long. A cascade plot of the vibration data (Figure 18-41) showed that just before trip of the unit at 21,960 rpm, an instability vibration component occurred near 15,000 cpm. From this and other data, it was determined that the high vibrations were caused by:

- A sudden increase in nonsynchronous vibration as the unit approached full speed resulting in shaft bow.
- A sudden increase in unbalance due to the shaft bow and as a result, a rapid increase in synchronous vibration levels as the nonsynchronous components disappeared.

After the problem source was identified using the above data analysis technique, computer simulation of the rotor led to a solution consisting of bearing modifications. The stability analysis of the pump rotor predicted that the pump had an unstable mode at 15,000 cpm with a negative loga-

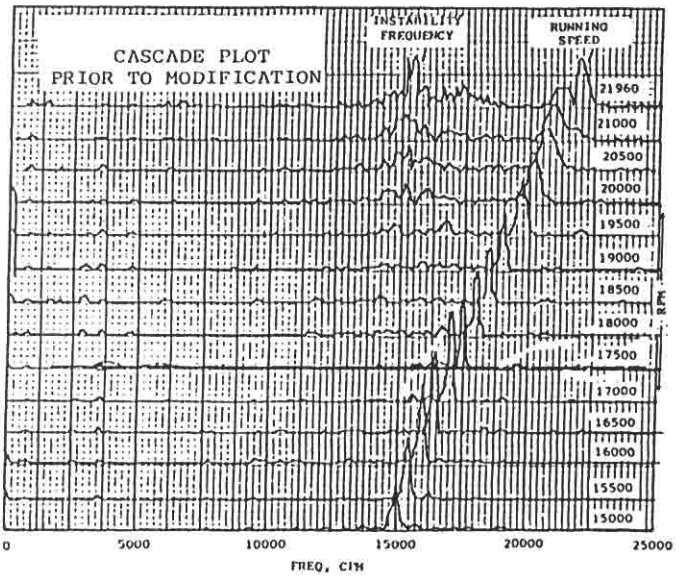


Figure 18-41. Nonsynchronous instability vibration of high speed pump.

rithmic decrement of 0.01 for a simulated fluid aerodynamic loading of 1,000 lb/in. at the impellers [36, 47]. The pump rotor with the modified bearings was predicted to have a positive logarithmic decrement of 0.10. The rotor vibrations after the bearing modifications were made are shown in Figure 18-42. The nonsynchronous vibration component was no longer present and the unit has since operated successfully.

Appendix
Acoustic Velocity of Liquids

The acoustic velocity of liquids can be written as a function of the isentropic bulk modulus, K_s and the specific gravity:

$$c = 8.615 \sqrt{\frac{K_s}{\text{sp gr}}} \tag{1}$$

where c = acoustic velocity, ft/sec
sp gr = specific gravity
K_s = isentropic (tangent) bulk modulus, psi

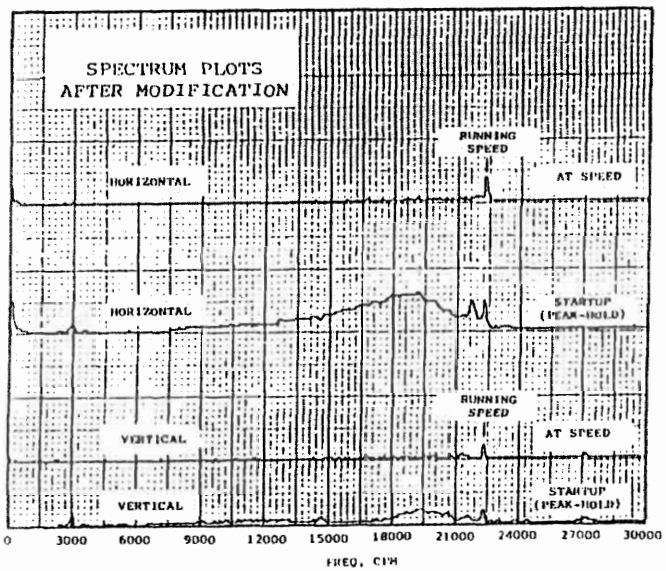


Figure 18-42. High speed pump vibrations after bearing modification.

At low pressure, water can be considered to be incompressible and the acoustic velocity is primarily dependent upon the change of density with respect to temperature. The acoustic velocity of water at low pressures [48] is given for various temperatures from 32° to 212°F in Figure 18-43. For elevated pressures, the acoustic velocity must be adjusted for pressure effects and the function using the bulk modulus is convenient.

The bulk modulus of water [49] can be calculated with the following equation for pressures up to 45,000 psig and various temperatures.

$$K_s = 1000\ K_o + 3.4P \qquad (2)$$

where K_o = constant from Table 18-6
K_s = isentropic tangent bulk modulus, psi
P = pressure psia

The calculation accuracy is $\pm 0.5\%$ for the isentropic bulk modulus of water at 68°F and lower pressures. The error should not exceed $\pm 3\%$ at elevated pressures (greater than 44,000 psi) and temperatures (greater than 212°F).

The bulk modulus for petroleum oils [50] (hydraulic fluids) can be determined for various temperatures and pressures by using Figures 18-44 and 18-45 which were developed by the API. The isothermal secant bulk modulus for petroleum oils at 20,000 psig is related to density and tem-

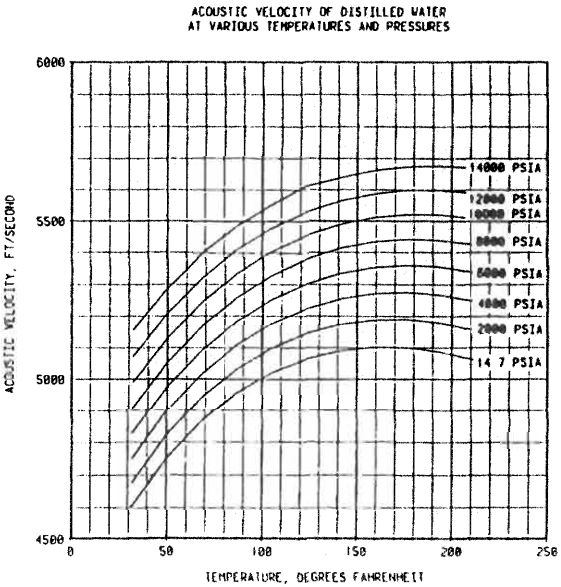


Figure 18-43. Acoustic velocity of distilled water at various temperatures and pressures.

Table 18-6 [49]
Constant K_o for Pressure Correction for Bulk Modulus of Water

$K_s = 1000\ K_o + 3.4P$
P = pressure, psia
(valid up to 45000 psia)

Temperature (°F)	Constant K_o
32	289
50	308
68	323
86	333
104	340
122	345
140	348
158	348
176	341
194	342
212	336

perature in Figure 18-44. The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure within ±1%. Pressure compensation for the isothermal secant bulk modulus can be made by using Figure 18-45.

The isothermal bulk modulus, K_T, and isentropic bulk modulus, K_s, are related by the following equation:

$$K_s = K_T \frac{c_p}{c_v} \tag{3}$$

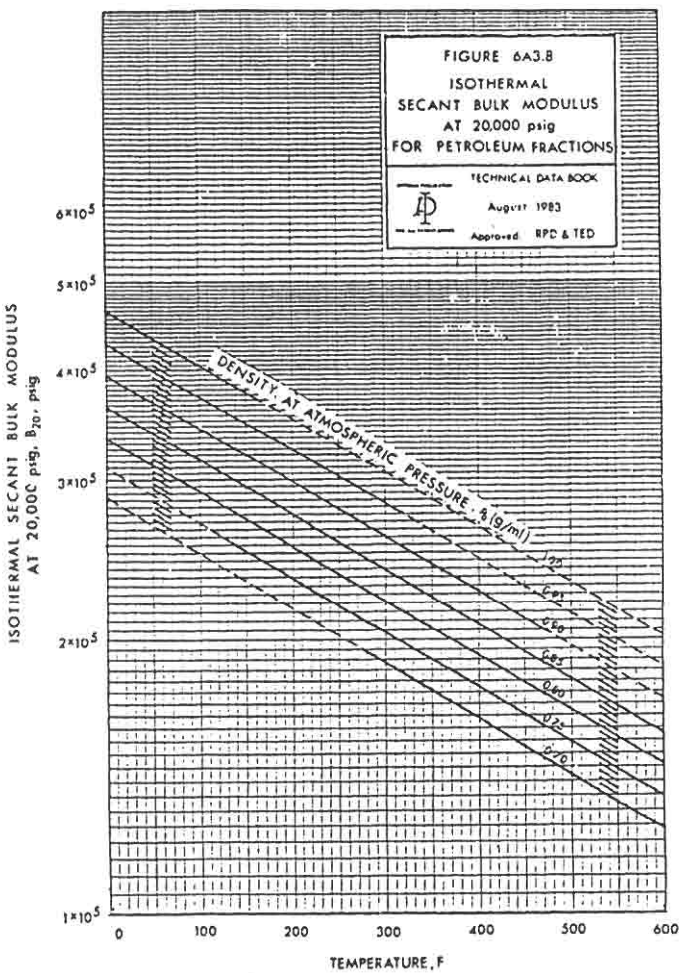


Figure 18-44. Isothermal secant bulk modulus at 20,000 psig for petroleum fractions [50].

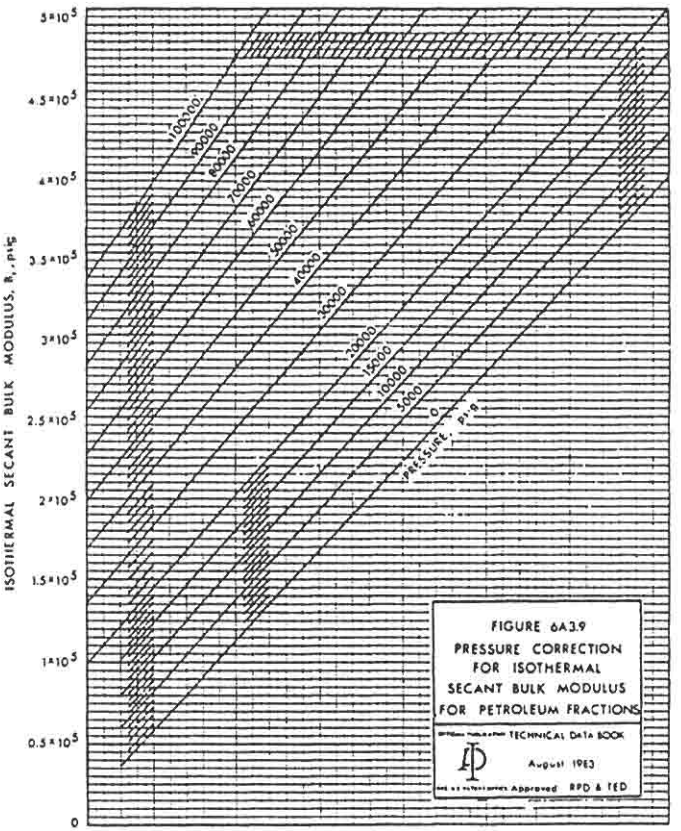


Figure 18-45. Pressure correction for isothermal secant bulk modulus for petroleum fractions [50].

The value of c_P/c_V for most hydraulic fluids is approximately 1.15.

The isentropic tangent bulk modulus for common petroleum oils can be determined from the Figures 18-44 and 18-45 as follows:

1. On Figure 18-44, enter the desired temperature on the horizontal scale and go to the proper specific gravity line to read the isothermal secant bulk modulus at the reference pressure of 20,000 psig.

2. Enter the isothermal secant bulk modulus (value from Step 1) on Figure 18-45 on the vertical scale; proceed horizontally to intersect the 20,000 psi reference pressure line; move vertically to the line corresponding to twice the desired pressure. Move horizontally to

the left and read the equivalent isothermal tangent bulk modulus on the scale.

3. Multiply the isothermal tangent bulk modulus by 1.15 (to compensate for ratio of specific heats) to determine the value of the isentropic tangent bulk modulus.

The value of the isentropic tangent bulk modulus obtained in Step 3 can be used in Equation 1 to calculate the acoustic velocity.

Piping systems with incompressible fluids (liquids) have an apparent effect on the acoustic velocity because of the pipe wall flexibility. The classical Korteweg correction can be used for thin wall pipe (wall thickness < 10% diameter) to adjust the acoustic velocity.

$$c_{adjusted} = c \sqrt{\frac{1}{1 + \frac{DK_s}{tE}}} \tag{4}$$

where D = pipe diameter, in.
t = pipe wall thickness, in.
E = elastic modulus of pipe material, psi

References

1. Florjancic, D., Schoffler, W. and Zogg, H., "Primary Noise Abatement on Centrifugal Pumps," Sulzer Tech., Rev. 1:24, 1980.
2. Szenasi, F. R. and Wachel, J. C., Section 8.4, "Pump Noise," *Pump Handbook,* 2nd ed., McGraw-Hill, 1986, pp. 8.101–8.118.
3. Jones, A. B., "The Mathematical Theory of Rolling-Element Bearings," *Mechanical Design and Systems Handbook,* McGraw-Hill, 1964, pp. 13-1 to 13-76.
4. *Centrifugal Pump Hydraulic Instability,* CS-1445 Research Project 1266-18, Electric Power Research Institute, 3412 Hillview Ave., Palo Alto, California, 1980.
5. Wachel, J. C. and Tison, J. D., *Field Instrumentation and Diagnostics of Pump Vibration Problems,* Presented at Rotating Machinery and Controls Short Course, University of Virginia, June 6–7, 1983.
6. Mancuso, J. R., *Couplings and Joints,* Marcel Dekker, Inc., New York, 1986, pp. 340–350.
7. API Standard 610 "Centrifugal Pumps for General Refinery Services," Section 2.8, 7th ed., American Petroleum Institute, Washington, D.C., 1989.

8. Bloch, H. P., "Improving Machinery Reliability," *Practical Machinery Management for Process Plants,* Vol. I, Gulf Publishing Company, Houston, Texas, 1982, pp. 72–162.
9. Atkins, K. E., Tison, J. D. and Wachel, J. C., "Critical Speed Analysis of an Eight Stage Centrifugal Pump," *Proceedings of the Second International Pump Symposium,* April 1985, pp. 59–65.
10. Wood, D. J. and Chao, S. P. "Effect of Pipeline Junctions on Waterhammer Surges," *Proceedings of the ASCE, Transportation Engineering Journal,* August 1978, pp. 441–457.
11. Fraser, W. H., "Flow Recirculation in Centrifugal Pumps," in *Proceedings of the Tenth Turbomachinery Symposium,* 1981.
12. Fraser, W. H., "Recirculation in Centrifugal Pumps: Materials of Construction of Fluid Machinery and Their Relationship to Design and Performance," paper presented at winter annual meeting of ASME, Washington, D.C., November 16, 1982.
13. Wachel, J. C. and Szenasi, F.R., et al., "Vibrations in Reciprocating Machinery and Piping," *EDI Report 85–305,* Chapter 2, October, 1985.
14. Sparks, C. R. and Wachel, J. C., "Pulsations in Liquid Pump and Piping Systems," *Proceedings of the Fifth Turbomachinery Symposium,* October 1976, pp. 55–61.
15. Wylie, E. B. and Streeter, V. L., *Fluid Transients,* FEB Press, Ann Arbor, Michigan, 2nd Ed., pp. 213–224, 1983.
16. *Technical Data Handbook-Petroleum Refining,* Vols. I, II, III, 4th Ed., American Petroleum Institute, Washington, D.C., 1983.
17. Kinsler, L. E. and Frey, A. R., *Fundamentals of Acoustics,* John Wiley & Sons, New York, 3rd Ed., 1982, Chapters 9–10.
18. Wachel, J. C., "Design Audits," *Proceedings of the Fifteenth Turbomachinery Symposium,* November 1986, pp. 153–168.
19. Massey, I. C., "Subsynchronous Vibration Problems in High-Speed Multistage Centrifugal Pumps," *Proceedings of the Fourteenth Turbomachinery Symposium,* October 1985, pp. 11–16.
20. Childs, D. W. "Finite Length Solution for Rotordynamic Coefficients of Turbulent Annular Seals," ASLE Transactions, *Journal of Lubrication Technology,* 105, July 1983, pp. 437–445.
21. Karassik, Igor, "Centrifugal Pump Construction," Section 2.2, *Pump Handbook,* 2nd Ed., McGraw-Hill, 1986, pp. 2–69.
22. Childs, D. W. and Scharrer, J. K., "Experimental Rotordynamic Coefficient Results for Teeth-on-Rotor and Teeth-on-Stator Labyrinth Gas Seals," ASME Paper No. 86-GT-12, 1986.
23. Iwatsubo, T., Yang, B. and Ibaraki, R., "An Investigation of the Static and Dynamic Characteristics of Parallel Grooved Seals," The Fourth Workshop of Rotordynamic Instability Problems in High-Per-

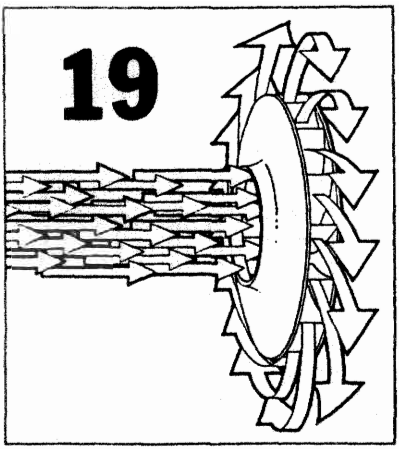
formance Machinery, Turbomachinery Laboratories, Texas A&M University, College Station, Texas, June 1986.
24. Black, H. F. and Jenssen, D. N., "Dynamic Hybrid Properties of Annular Pressure Seals," ASME Paper 71-WA/FF-38, 1971.
25. Wachel, J. C., and Szenasi, F. R., et al, "Rotordynamics of Machinery," *EDI Report 86-334,* April 1986, Chapters 3-5.
26. Allaire, P. E. and Flach, R. D., "Design of Journal Bearings for Rotating Machinery," *Proceedings of 10th Turbomachinery Symposium,* December, 1981.
27. Childs, D. W., "Dynamic Analysis of Turbulent Annular Seals based on Hirs' Lubrication Equation," ASME Transaction, *Journal of Lubrication Technology,* 105, July 1983, pp. 429-436.
28. Childs, D. W. and Kim, C. H., "Analysis and Testing for Rotordynamic Coefficients of Turbulent Annular Seals with Different, Directionally-Homogeneous Surface-Roughness Treatment for Rotor and Stator Elements," ASME paper, ASME-ASLE Joint Lubrication Conference, October 1984.
29. Black, H. F., "Effects of Hydraulic Forces in Annular Pressure Seals on the Vibrations of Centrifugal Pump Rotors," *Journal of Mechanical Engineering Science,* 11, (2), 1969.
30. Childs, D. W. and Moyer, D. S., "Vibration Characteristics of the HPOTP of the SSME," ASME Paper 84-GT-31,1984.
31. Von Pragenau, G. L., "Damping Seals for Turbomachinery," NASA TP-1987, March 1982.
32. Bolleter, U., Frei, A. and Florjancic, D., "Predicting and Improving the Dynamic Behavior of Multistage High Performance Pumps," *Proceedings of the First International Pump Symposium,* May 1984, pp. 1-8.
33. Wachel, J. C., "Rotor Response and Sensitivity," *Proceedings Machinery Vibration Monitoring and Analyses,* Vibration Institute, Houston, Texas, April 1983, pp. 1-12.
34. ASA STD2-1975 "Balance Quality of Rotating Rigid Bodies," Acoustical Society of American Standard, ANSI S2, 1975.
35. API Standard 617 "Centrifugal Compressors for General Refinery Services," Section 2.8, 5th Ed., American Petroleum Institute, Washington, D.C., 1988.
36. Wachel, J. C., "Rotordynamics Instability Field Problems," Second Workshop on Rotordynamic Instability of High Performance Turbomachinery, NASA Publication 2250, Texas A&M University, May 1982.
37. Black, H. F., "Calculation of Forced Whirling and Stability of Centrifugal Pump Rotor Systems," ASME Paper No. 73-DET-131, 1973.

38. Barrett, L. E., Gunter, E. J. and Nicholas, J. C., "The Influence of Tilting Pad Bearing Characteristics on the Stability of High Speed Rotor-Bearing Systems," *Topics in Fluid Film Bearing and Rotor Bearing Systems, Design and Optimization,* ASME Publication Book No. 100118, 1978, pp. 55–78.
39. Szenasi, F. R. and Von Nimitz, W., "Transient Analysis of Synchronous Motor Trains," *Proceedings of the Seventh Turbomachinery Symposium,* December 1978, pp. 111–117.
40. Szenasi, F. R. and Blodgett, L. E., "Isolation of Torsional Vibrations in Rotating Machinery," *National Conference on Power Transmissions,* 1975, pp. 262–286.
41. Military Standard 167, Bureau of Ships, 1964.
42. Peterson, R. E. *Stress Concentration Factors,* John Wiley and Sons, 1974, pp. 245– 253.
43. Frej, A., Grgic, A., Heil, W. and Luzi, A., "Design of Pump Shaft Trains Having Variable-Speed Electric Motors," *Proceedings of the Third International Pump Symposium,* May 1986, pp. 33–44.
44. Frank, K., "Frequency Converters for Starting and Speed Regulations of AC Motors for Turbomachine Applications," *Proceedings of the Tenth Turbomachinery Symposium,* December 1981, pp. 87–94.
45. Szenasi, F. R., "Diagnosing Machinery-Induced Vibrations of Structures and Foundations," ASCE Spring Meeting, April 29, 1986.
46. Wachel, J. C., "Prevention of Pump Problems," Worthington 2nd Technical Pump Conference, October 1981.
47. Allaire, P. E., et al., "Aerodynamic Stiffness of an Unbounded Whirling Centrifugal Impeller with an Infinite Number of Blades," *Workshop on Rotordynamic Instability,* Texas A&M University, May 1982.
48. Wilson, W. D., "Speed of Sound in Distilled Water as a Function of Temperature and Pressure," *Journal of the Acoustical Society of America,* Vol. 31, No. 8, August 1959, pp. 1067–1072.
49. Hayward, A. T. J., "How to Estimate the Bulk Modulus of Hydraulic Fluids," *Hydraulic Pneumatic Power,* January 1970, p. 28.
50. *Technical Data Handbook-Petroleum Refining,* Vol. I, 4th Ed., 8th Revision, American Petroleum Institute, Washington, D.C., 1983, pp. 6-65 to 6-82.

Part 4

Extending Pump Life

This page intentionally left blank



ALIGNMENT

by **Malcolm G. Murray, Jr.**
Murray and Garig Tool Works

Definitions

Internal Alignment. This type of alignment refers to machined fits and precise assembly that ensure that the pump incurs no excessive deviation from intended tolerances *prior to* installation on its pedestal and connection of piping and shaft coupling. Although of obvious importance, this type of alignment will not be emphasized here. It is covered in numerous general and specific manufacturers' manuals and shop practice standards. Further reference to it will be brief, and instructions for its accomplishment will not be given in this section.

External Alignment. This type of alignment refers to a procedure or condition that ensures the accuracy with which the centerline of one machine shaft coincides with the extension of the centerline of an adjacent machine shaft to which it is to be coupled. For horizontal machine elements, alignment figures are customarily given in vertical and horizontal planes along the shaft axis, both necessary for the complete picture. For vertical machine elements, similar information is given in two convenient planes perpendicular to each other. *Alignment* in this chapter will usually refer to external alignment. Other terms often used are *shaft alignment, coupling alignment, angular,* and *parallel alignment.* See Figure 19-1.

Why Bother With Precise Alignment?

This is a logical and legitimate question. Certainly, if pumps would run as well with poor alignment as with good alignment, this section of the

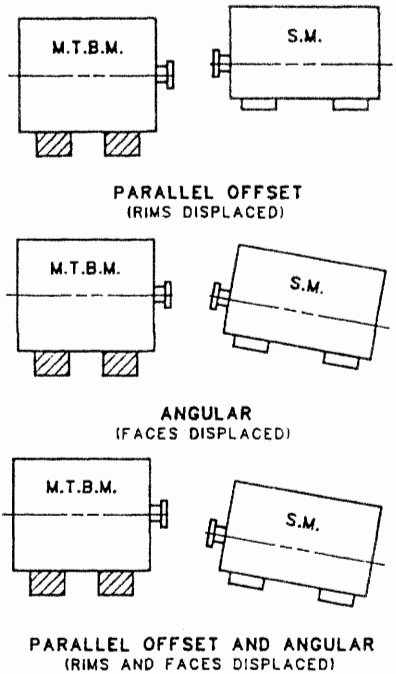


Figure 19-1. Basic conditions of misalignment.

book could be omitted. Experience has shown though, rather painfully at times, that excessive misalignment can have undesirable consequences. These include internal rubs, excessive vibration, increased static and cyclical stress. These factors can lead to premature failure of bearings, seals, couplings, and other parts. Service interruptions, fire, and personnel injury can result. Figure 19-2 shows a catastrophic pump failure caused by misalignment. Costs of such a failure will seldom be less than several thousand dollars and can easily run into the millions if the failure consequences go beyond the original pump.

Causes of Misalignment

Internal misalignment is generally caused by defective parts, poor machining, and/or improper fits and assembly. One other occasional cause is an improper field practice—varying the bolt torque or the gap at the pump head joint on a vertically split pump to make external alignment adjustments.

External (shaft/coupling) misalignment can be caused by a host of factors. These factors are described in the following pages.

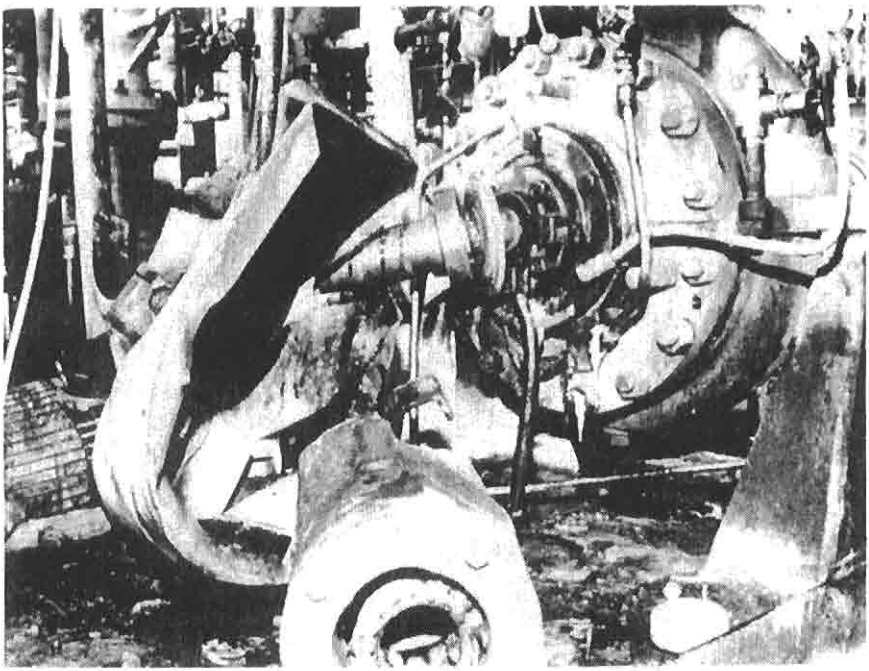


Figure 19-2. Pump damage caused by bearing seizure. Bearing seizure was result of gear coupling damage from excessive misalignment caused by piping forces.

Management and Human Factors

Standards and Tolerances. Lack of suitable standards and tolerances, or lack of their enforcement, can make a mockery of any attempt to accomplish meaningful alignment. More specific recommendations will be given later.

Training. Inadequately trained alignment personnel are likely to do poor alignment, quickly, or good alignment, slowly. Well trained and equipped personnel, by contrast, can do good alignment systematically and efficiently without wasting time in a trial-and-error approach.

Time Skimping. In training people to do alignment systematically and efficiently, the most frequent comment from students goes something like this:

> I like the new tools and methods you have shown us. I would like to use them to do a better job. But, when I am given an alignment job to do, my supervisor wants a *fast* job, and quality is secondary. So,

I use a straightedge and take 30 minutes. If I use dial indicators, they are just for show, and nobody checks their readings.

Our answer to this is to suggest that the student politely tell his supervisor that a proper pump alignment job will typically require a half day rather than a half hour. Then ask, "Do you want me to do a sloppy job?" Few supervisors will answer this question in the affirmative. It all gets back, though, to standards and tolerances, and whether management is serious about their enforcement. If they are not, the training is a waste of time, and the only alignment tool needed is a crooked straightedge.

Tooling and Supplies. Even a well-trained field machinist or millwright, if inadequately equipped, will be severely handicapped in doing a good alignment job efficiently. For small machines, inexpensive tooling is available that can give adequate results if used properly. More typically, though, a process plant having a number of pumps of various sizes, can benefit greatly by acquiring several thousand dollars worth of alignment tools and supplies. Skimping here is false economy. If funds are tight for tool purchase, good alignment tools and tool kits can sometimes be rented. (See Murray [6] and [10] for more specific details on tools and examples showing return on investment for alignment tools.)

Physical Factors

Foundation. Occasionally, the foundation mass will prove insufficient for stability, resulting in an insufficient damping ability. To be assured of having adequate foundation mass, a rule of thumb calls for three times the machine weight for rotating machines and five times the machine weight for reciprocating machines, assuming a density of 150 lb/ft^3 for reinforced concrete. If soft soil conditions are present, spread footings should be considered. A flexible isolation joint between the foundation and the pavement slab can be helpful in avoiding undesired vibration transmission between adjacent foundations.

A common deficiency in pump foundation design is the insufficient extension of the foundation's perimeter beyond the outline of the pump baseplate. This can cause difficulties in leveling and grouting. Foundation extension three inches beyond the baseplate perimeter is recommended.

Foundation strength is related to the mix composition and rebar placement. The mix should be workable, but not soupy, giving at least 2,500 psi compressive strength when tested following a 28-day cure. It should not have shrinkage cracks. In addition to proper engineering design of the rebar size and layout, care should be taken to ensure at least two inches concrete cover to resist spalling.

The best designs fail if the executions are faulty. Taking pour samples for later testing (and making all concerned aware of this) helps dull the temptation to add excessive water to the mix. The actual rebar layout should be inspected for design compliance *before* the pour is made, for obvious reasons. A minimum three-day wet cure is desirable, although additives and/or plastic sheeting are sometimes used instead for moisture retention. Regardless of method, responsible followup is essential to be sure that what is called for really does get done.

Baseplate—Common Problems and Suggested Solutions. The baseplate may be excessively flexible. Another point to watch is shipping and in-plant transportation of baseplates. Electric motors are usually much heavier than the pumps they drive. For this reason, it is good to have motors over 100 horsepower shipped unmounted to avoid baseplate distortion caused by the motor weight acting on the ungrouted baseplate.

For welded steel baseplates, request oven stress relieving after welding and before final machining of the pad surfaces and before protective coating is applied. This will help avoid distortion due to gradual in-service stress relief.

High strength machine-to-pedestal hold-down bolts, such as SAE Grade 8, can be used as replacements, but should not be used as original equipment in which their high strength is taken advantage of in determining bolt size and/or quantity. Sooner or later, somebody in the field will substitute one or more Grade 2 mild steel bolt(s), which will be vulnerable to overstressing.

Specify 3/16 in. shim space beneath driver feet, rather than the 1/16 in. or 1/8 in. called for by API610. It is often necessary to remove more than 1/8 in. of shim thickness, but seldom necessary to remove more than 3/16 in.

To avoid restricting necessary horizontal alignment movement capability, specify a minimum of 1/8 in. annular clearance between full-size (not undercut) SAE Grade 2 bolts and their boltholes after the driver has been aligned to the pump at the factory. Once satisfactory running alignment has been achieved in the field, install two Number 8 taper dowel pins at the pump for easy return of the pump to its pedestal to a *nearly aligned* position. Use an anti-seize compound when installing these pins. An exception would apply for a row of identical pumps with a *floating spare.* Here, doweling is useless. The driver should not be doweled because this will normally require some movement for aligning it to the pump each time the latter is reinstalled following shop repair. Three pairs of foot/pad punch marks can be used instead.

Specify rounded baseplate flange corners with approximately 2-inch radius. This will reduce the tendency for grout cracking at corners and will also improve safety in handling.

Baseplate surface protection should be specified for the dual purpose of corrosion protection and for a good grout bonding surface. One good system is an inorganic zinc silicate coating applied immediately after sandblasting to white metal per SSPC SP-5. This leaves a rough, dull surface with the coating having a molecular bond to the metal, and works well with both inorganic and epoxy grouts.

Preferably, one 4-inch grout filling hole should be provided at the center of each bulkhead section, with one 1/2-inch vent hole near each corner of the section. This allows controlled grout placement and verification that each section is filled with grout.

Vertical leveling screws should be specified. Shims and wedges should be avoided. If left in place after grouting, these may cause "hard spots" rather than the uniform support desired from the grout. Also, they may allow moisture penetration, followed by corrosion and grout spalling.

Mounting pads should be machined flat, parallel, and coplanar. This should be done after all welding and stress relieving are complete.

Individual requirements for the installation of alignment jackscrews should be examined closely. At times, as with an axially long boiler feed pump, it may be worthwhile to have jackscrews at the pump rather than or in addition to those at the driver. In the majority of cases, jackscrews can be omitted at electric motors if adequate access is available to allow the use of a soft-face deadblow hammer. This is a surprisingly accurate method for making horizontal alignment moves without damaging the motor. It will work even on large motors of several thousand horsepower and weighing several tons. For motors 500 horsepower and larger, it is worthwhile to ask for API 610's optional 2-inch minimum clearance beneath the motor center at each end of the motor. This will permit the use of hydraulic jacks for lifting or a pair of portable alignment positioners for easy adjustment in any direction, vertical and horizontal [9]. For turbine drivers, alignment jackscrews are nearly always worth having. Jackscrews and their mounting lugs should be resistant to corrosion and sized to avoid bending or breaking in use. For most installations 5/8-in. screws installed in 2-in. × 3/4-in. lugs are strong enough. Zinc plating protects against corrosion at low cost, or upgrading to 316L stainless steel may be considered. If the lugs are installed by welding, they should be placed in such a way to allow access for machine and shim installation and removal. If this is not feasible, they should be attached by drilling, tapping, and bolting. Tables 19-1 and 19-2 give further information on jackscrews and other alignment movement methods.

Grouting. This subject is too extensive to cover adequately here. We will instead make a few general remarks and mention some references having more information.

Table 19-1
Vertical Alignment Movement Methods

Method	Advantages	Disadvantages
Integral Jackscrews, Threaded into Machine Feet	No problem locating jacking surfaces. Inexpensive if included when the machine is purchased.	Shims require an extra hole or slot. Threads sometimes rust and cause difficulty in turning the jackscrew. If the screw breaks, removal of the remaining fragment may be difficult. Difficult and expensive to retrofit after machine is installed.
Prybar	Inexpensive. Readily available. Can get into low-clearance locations.	Access and clearance are sometimes a problem, limiting usefulness and load capacity. Safety can be compromised if the bar slips while someone is reaching beneath the machine feet to install or remove shims.
Portable Jacks and Wedges	Available in a variety of types and load ratings, most fairly inexpensive, with collapsed heights down to 3/16 in.	Sometimes the one needed is somewhere else, and the one available is not suitable. This is not really a disadvantage for the method, however, but merely an indication of poor planning.
Crane or Hoist	The obvious solution if bottom clearance, access, or load bearing surfaces are unsuitable for jacking.	Expensive and bulky. Sometimes have access problems. Poor small movement control.
Portable Alignment Positioners	In a single device, they give vertical, precise horizontal transverse, and precise horizontal axial movement capability. Capacities up to 20 tons per pair are available. Used in groups of four, the maximum capacity increases to 40 tons. (For more details, see References 6, 9, and 14.)	Two inch minimum vertical clearance required. More expensive than regular jacks. Not usually worth bothering with for machines smaller than 500 hp.

(table continued on next page)

Table 19-1 Continued
Vertical Alignment Movement Methods

Method	Advantages	Disadvantages
Portable Alignment Positioners	Easy to use for movement of large electric motors. Can also be used on some turbines and gearboxes.	

Table 19-2
Horizontal Alignment Movement Methods

Method	Advantages	Disadvantages
Steel Hammers and Sledges	Readily available and inexpensive. No interference with shim installation and removal.	Poor movement control. Likely to damage machines being moved.
Soft-face, Hollow Head, Shot-loaded Deadblow (No-bounce) Hammers and Sledges	Fairly inexpensive. Good movement control (no bounce). Works similar to lead hammer, but far more durable. Does not cause machine damage. Available in a range of sizes and weights. Short handle 6 lb. and long handle 11 lb. will cover most alignment requirements.	Access sometimes insufficient for use.
Jackscrews	Good movement control if screws and mounting lugs are sufficiently large and rigid. Durable, if materials are chosen intelligently. Jackscrews do not cause machine damage. Easy to specify or install on new machines.	If mounting lugs are too flimsy, stick-slip action causes poor movement control. If not of suitable corrosion resisting material, screws may corrode and cannot be turned. Often interfere with direct shim installation and removal, requiring shims to be inserted and removed from foot ends or inside edges. This can become awkward at times. Bolting, rather than welding the jackscrew lugs in place, can solve this problem.

Table 19-2 Continued
Horizontal Alignment Movement Methods

Method	Advantages	Disadvantages
		Difficult to retrofit to existing machines. Not usually portable. Individual lugs and screws are normally needed at each machine. Portable clamp-on jackscrews are sometimes used, but usually it is difficult to find a good place and way to clamp them on.
Portable Jacks	Easy to use, where applicable. Fairly good movement control.	A nearby fixed point to jack against is required, and often this is not present.
Comealong or Horizontal Hoist	Can attach to fixed anchor point some distance away.	Poor control. Stick-slip action is likely.
Portable Alignment Positioners	(See previous description in Table 19-1.) These devices handle vertical and horizontal movement in the same overall step.	
Doweling	Returns a machine to its previous position on its support.	Dowels sometimes stick and break off. Access for drilling and reaming is sometimes insufficient.
Matching Punchmarks	Easy to apply, using automatic centerpunch, in three foot locations. Can be used on both machine elements, whereas doweling is normally used only on one.	Less precise than doweling. Does not move the machine, but gives an indication of where to put it to restore its previous position.

The purpose of grouting is to fill the baseplate cavity without leaving voids and to stiffen the baseplate, providing an even support to transmit load into the foundation.

Grout material is normally either organic (usually epoxy plus aggregate) or nonmetallic inorganic (Portland cement, sand, water, and nonmetallic anti-shrink additives). The choice is based on economics and en-

vironmental factors. More specifically, epoxy is more expensive and less resistant to high temperatures, but has higher early and ultimate strength and greater chemical and impact resistance than inorganics. It is also easier to place properly and to cure.

With either category of grout material, success depends on careful planning and preparation and attention to numerous details such as site preparation, equipment selection, and a stepwise procedure for proportioning and mixing, placement, curing, and protective coating. (For more thorough coverage of this subject, see Bloch [1] and Murray [12].)

Piping Fit and Support. Here, the object is to avoid excessive pump nozzle loading. Piping fit is a function of installation workmanship and effective construction inspection/quality control. Piping flexibility and support are functions of design, with due consideration given to weights, stiffnesses, and thermal effects. If not carefully designed to avoid such problems, heavy, hot piping can become "the tail that wags the dog," with the dog being the pump.

Flexible Coupling. Choice of flexible coupling can be important. Tradition and price are the two most often used criteria, and either can result in a poor choice. For most pumps, it is desirable to use a spacer coupling. This gives more parallel misalignment capability than a close coupling of the same type and also provides better access for maintenance.

If radial space is limited, a gear coupling may be required. This will transmit more horsepower at a given speed than any other type occupying the same space. It is, however, more sensitive to misalignment than other types and also requires lubrication. The latter factor is no longer as much of a disadvantage as it once was though, because non-separating coupling greases are now available. A gear coupling can handle limited end float applications for sleeve bearing motors, but doing this correctly can get complicated.

Gridmember couplings are another relatively inexpensive lubricated design used mostly for the smaller pumps.

Elastomeric couplings are popular for pumps in certain size and speed ranges. These accommodate fairly large misalignments, but are sensitive to high temperatures (including heat caused by rubbing against a coupling guard) and certain chemicals. This type of coupling may be unable to handle sleeve bearing motor limited end float applications well.

Dry metallic flexing element couplings are available in several designs—disk, diaphragm, and cantilever link. Chloride stress corrosion cracking is sometimes a problem, but can be minimized by judicious choice of element material, for example, type 316 stainless steel in preference to type 304. These couplings are a good choice for use with sleeve

bearing motors because they have an elastic centering characteristic that is easy to apply to end float limitation and geometric centering of the motor rotating element.

Alignment Access. As in many fields, forethought is helpful when it comes to alignment access. A common problem, easily avoided in the specifying stage, is insufficient exposed shaft between front of bearing housing and back of coupling for concealed hub couplings. At least one inch of exposed shaft is desirable to allow alignment bracket attachment. It is helpful to visualize the radial space requirements of alignment brackets for full shaft rotation and to avoid placing piping or other interferences in this circular path. Access for moving and monitoring moves on the machine element to be adjusted for alignment (usually the driver) should also be considered. As mentioned earlier, a 2-inch vertical jacking access beneath each end of a motor will allow insertion of hydraulic jacks or alignment positioners [9]. Access to install and remove shims and place horizontal move monitoring dial indicators is also important. Access for applying and turning wrenches on hold-down bolts should not be overlooked. Aligning a machine that has one side immediately adjacent to a wall, or blocked by a maze of pipes, can require acrobatic maneuvers with a degree of difficulty directly proportional to the age, size, weight, and arthritic tendencies of the aligner.

Pre-Alignment Steps

It may seem to some readers that we are taking a long time to get into the ostensible subject—shaft/coupling alignment. This may be true, but there is good reason for it. One thing builds upon another. Without certain prealignment steps, the alignment itself is likely to be dangerous, difficult, or ineffective. Nobody knows exactly how the pyramids were built, but it is a pretty sure bet that they were not done from the top down. Pre-alignment steps will now be described.

Work Permit and Power Lockout. This is basic and is usually the first step. It is essential that all concerned agree on *which* pump/driver combination is to be aligned, and that it will be effectively prevented from starting under power during the alignment.

Personnel Coordination. Sometimes the work area is crowded, and several categories of work are to be done. Alignment is the type of work that suffers if non-alignment personnel, such as electricians and pipefitters, are doing their own jobs in the same vicinity and inadvertently lean against a jacked-up machine, kick a foot-move-monitoring dial indicator,

etc. It is best in such cases to arrange for the various types of work to be done separately, either sequentially or on separate shifts.

Heaters and Circulation. Some electric motors have base or winding heaters that come on when the main power is deactivated. These can cause undesired motor growth and can also magnetize dial indicators and cause them to stick. It is usually desirable to get such heaters turned off before starting the alignment. The same is true for any hot fluid or steam circulation through pump or turbine.

Shading or Night Schedule. These are precautions more likely to be needed with large elevated compressor trains than with typical pumps. Nevertheless, it is worth mentioning that uneven sun heating can sometimes affect alignment and cause inconsistencies during the job. The usual ways of avoiding such problems are to erect a tarp sunshade or do the work at night.

Installation of Pump on Pedestal. The pump foot and pedestal surfaces should be clean and free of burrs.

Connection of Main Piping. Prior to connection of piping, stationary-mount vertical and horizontal dial indicators should be applied to the pump at its coupling end, set to mid range, and zeroed. The pump suction piping should then be connected, followed by the discharge piping. During flange bolt tightening, the indicators should be watched, and any indicated distortion should be kept less than 0.003 in. for each flange by selective tightening of flange bolts. If this cannot be achieved, it may be necessary to refit the pipe or use a dutchman. The latter is a tapered filler ring inserted between the flanges, with a gasket on each side to occupy the uneven gap. (See Murray [7] for more information on dutchmen.)

Coupling Surface Runout Check. This is sometimes necessary, but not always. It should be done at face and rim measurement surfaces of any coupling that will remain stationary while the dial indicators rotate, contacting these surfaces. If both coupling halves will be rotated in coordination or while coupled, the surface runout check can be omitted. To do the runout check, apply indicators mounted from a stationary point and rotate the coupling while watching for deviations from zero start. If these appear, mark their locations and set them away from quandrantal measurement locations (top-bottom-side-side).

Soft Foot Check and Shim Recommendations. Soft foot should be minimized to prevent machine frame distortion and inconsistent align-

ment results. It should be checked at electric motor drivers, which usually have four feet. Checking can often be omitted at turbines and pumps, which are frequently supported at two or three feet. To do the check, apply a vertical indicator mounted from a stationary base to each foot in turn, and measure the upspring when the hold-down bolt is loosened. If this exceeds 0.002 in., try shimming the indicated amount beneath the foot with the largest upspring. This should help, if the cause was improper shimming. Other causes may not be helped by this remedy though. These include heel-and-toe effect, tilting feet or support pads, and distorted or dirty shims. Remedies for these problems include remachining supports, using tapered or liquid epoxy shims, and replacement of bad shims with clean, flat shims. The best shim pack is a "sandwich" having thick shims top and bottom protecting a few thin shims in between. Measure the thickness of each shim. Markings are not always accurate, and thin shims often stick together, doubling their marked thickness. Stainless steel is generally the best material, and pre-cut shims are desirable if available in the outline size required. Pre-cut shims 0.050 inch and thicker should be checked with a straightedge for excessive edge distortion.

Axial Gap and End Float Check. For ball bearing motors, end float is zero and axial gap depends on coupling geometry. For elastic centering couplings (for example, metal disk), the coupling should be at zero axial deflection with the motor in the center of its float, or manufacturer's cold-spring recommendations should be followed. These are the easy ones.

For those who enjoy messy mathematics, Murray [6] has some examples showing how to calculate dimensions for achieving desired axial gaps and end floats with sleeve bearing motors having gear couplings. For most situations, however, it is easier to get these by intelligent trial and error. By knowing our objectives, the task becomes fairly easy. We wish to accomplish three things:

1. The motor shaft, when coupled, must be capable of passing through its float center.
2. It must be restrained by the connected coupling so that it cannot rub the outboard bearing stop.
3. It must be restrained by shaft contact, coupling thrust plates, or thrust buttons from rubbing the inboard bearing stop.

In effect, the last two transfer thrust forces to the pump thrust bearing through the coupling. Unlike the relatively weak motor thrust surfaces, it is capable of handling these forces.

By marking float extremities and float center on the shaft of the uncoupled motor, then coupling it and applying thrust in both directions while rotating on the oil film, it becomes easy to see whether our objectives have been met. If they have not, the solution becomes fairly obvious—move the coupling in or out on the shaft, move the motor frame in or out, add thrust buttons of necessary thickness, or a combination of these.

Bracket Sag Check. In most cases, particularly with spacer couplings, a sag check should be made on the indicator bracket to be used for the alignment. In doing this, rim sag should always be checked. Face sag, if face readings are to be taken, is usually negligible and can be ignored. An exception could occur with a long dogleg face mount. Procedures for handling this are shown in Murray [6], but will not be repeated here because this situation does not arise often with pumps. For vertical pumps, bracket sag may be ignored. It may exist, but it is constant everywhere and does not require correction. Vertical pumps seldom require field alignment because they normally achieve their alignment by machined fits.

To do a rim sag check, mount the bracket on a stiff pipe, zero the indicator at mid range in the top position, and roll the pipe from top to bottom on sawhorses, and note the bottom reading. This is "total sag," which is twice the bracket sag. Do this several times and "jog" or depress the indicator contact point and let it reseat at top and bottom reading.

To keep less than 0.001 in. error due to calibration pipe sag, using Schedule 40 calibration pipe, keep the following span limits:

Pipe	Span Between Supports
2 in.	2 ft. 6 in.
3 in.	3 ft.
4 in.	3 ft. 6 in.
6 in.	4 ft. 4 in.

For a face mounted bracket, do the same type of check between lathe centers or mount inside a pipe capped at one end. The latter method will usually require a sign reversal, but the numbers will be correct.

Measurement of Linear and Face Dimensions. This should be done to the nearest 1/8 in. for small machines and the nearest 1/4 in. for large machines, and the dimensions entered into a prepared alignment data form. Examples of such forms can be found in Murray [6]. Measurements should be based on indicator contact point centers and foot support centers.

Determination of Tolerances. This is necessary in order to know when the alignment is adequate and work can be stopped. Murray [6] and Bloch [1] contain a number of criteria for alignment tolerances based on coupling and machine type, distance between coupling flex planes, and rotating speed. Results vary with the different approaches, from very sloppy to very precise. There is general agreement, however, that coupling manufacturers' advertised tolerances are too lax for the machines being coupled, even though they may be all right for the couplings themselves. A simple rule of thumb that works well for most horizontal pumps is to allow 0.003 in. maximum centerline offset at measurement planes and the same amount for maximum face gap difference. For vertical pumps, in the infrequent cases requiring their manual alignment, 0.001 in. maximum face gap difference is recommended.

Methods of Primary Alignment Measurement

By "primary" measurement, we mean that thermal growth is not determined thereby, although it may be incorporated into the alignment data if known from some other source. Table 19-3 compares various primary alignment measurement methods. Figures 19-3A and B and 19-4 illustrate the two most common methods—face and rim and reverse indicator (rim and rim).

Table 19-3
Primary Alignment Measurement Methods

Method	Advantages	Disadvantages
Straightedge, with or without feeler gauge	Simple, inexpensive, easy to use if good coupling rim surfaces are available. Commonly used for preliminary alignment. Also used for final alignment of low speed and/or intermittently operated machines.	Less precise than most other methods.
Taper gauge; small hole gauge and outside micrometer; inside dial caliper	Simple and inexpensive. No bracket required. Fairly accurate with care. Used for measuring face gaps of close-coupled machines.	Limited application. Does only part of the alignment job.

(table continued on next page)

Table 19-3 Continued
Primary Alignment Measurement Methods

Method	Advantages	Disadvantages
Inside micrometer	Simple and inexpensive. No bracket required. Used for measuring face gaps. Not limited to close-coupled machines. Fairly accurate with care.	Limited application. Does only part of the alignment job.
Dial indicators, in any of several arrangements, as follows:	Precise measurement capability with relatively inexpensive equipment.	Costs more than straightedge and requires more careful handling.
Face and Rim (see Figure 19-3A)	Traditional arrangement familiar to most alignment personnel. Usable with large, heavy machines having one shaft difficult to rotate for alignment measurement. Geometrically more accurate than other methods for machines with large measurement diameters and short axial spans. Usually the easiest arrangement to apply to small, close-coupled machines.	If used with a sleeve bearing machine, shaft float can cause errors in face readings. There are at least three ways to eliminate these errors, but they make the procedure more complex. Reference 6 has further details. Geometrically less accurate than reverse-indicator arrangement for situations in which axial span exceeds face measurement diameter. This will usually be true for pumps having spacer couplings. Brackets are often face mounted, requiring coupling spacer removal, and are more difficult to calibrate for sag. Brackets are often custom built for a particular machine rather than being "universal" variable geometry type.
Reverse-Indicator (Rim and Rim). (See Figures 19-3B and 19-4)	Sleeve bearing shaft float does not cause error or require special procedure to avoid error. Geometrically more accurate	Inapplicable to machines having one or both shafts impractical to rotate for alignment measurement. Often impractical to use on

Table 19-3 Continued
Primary Alignment Measurement Methods

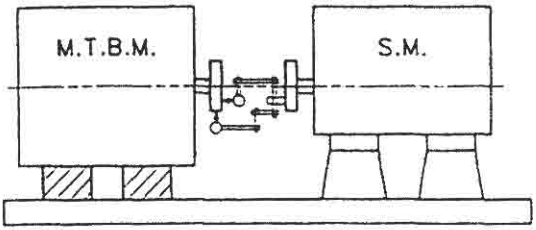
Method	Advantages	Disadvantages
	than face-and-rim arrangement, for spacer coupling applications. Face gap difference can be derived easily from rim measurements without the need for direct measurement. Reverse-indicator brackets are usually variable geometry type, clamped to hub rim or shaft, applicable to a range of machine sizes, and easy to calibrate for sag on a horizontal pipe. Both shafts turn together, so eccentricity and surface irregularities do not cause errors.	small, close-coupled machines.
Face-Face-Distance (See Reference 6)	Good for long spans. Applicable mainly to cooling tower drives, but sometimes used for pumps with long coupling spacers. Can also be used when spacers pass through a wall or bulkhead, as on some shipboard applications. Bracket sag is usually negligible.	Less accurate than other dial indicator methods. May require precautions to avoid axial float errors.
Partial Projection Reverse-Indicator (See Reference 14)	Good for long spans, mainly cooling tower drives with metal disk couplings. Avoids axial float error. More accurate than face-face-distance arrangement if used with metal disk couplings provided bracket sag is accounted for.	Must calibrate and account for bracket sag. Subject to diametral tooth clearance error if used with gear-type couplings.
Laser-Optic (OPTALIGN®)	Fast. Accurate.	Basic equipment. Will not handle two-element

(table continued on next page)

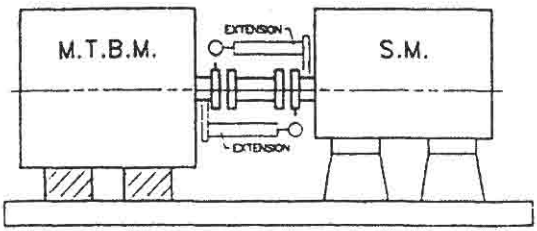
Table 19-3 Continued
Primary Alignment Measurement Methods

Method	Advantages	Disadvantages
	Laser beam has zero sag even over long spans. Measurement and calculating capability are combined in one system. Easy to use for basic alignment problems. Can also handle certain more complex problems, but procedures are more complex, or extra equipment is required. Requires about 7/8 in. exposed shaft for mounting, but this can be reduced to 9/16 in. by using non-original-equipment brackets. Metric and inch systems both present in same calculator. Can be used to measure soft foot originating from common causes. Can be used for progressive error reduction horizontal moves similar to the Barton method used with dial indicators. Can be used to accurately measure sag and its effect for long spacers and pipes.	or multi-element optimum move problems without supplemental calculations. Supplemental equipment available at extra cost can handle such problems. Both shafts must be turned together. This requirement can sometimes be met by using a roller bracket with inclinometer on a shaft that remains stationary. Subject to error if coupling backlash cannot be eliminated during rotation. Requires careful handling to avoid damage. Electrical. Requires periodic battery replacement, and gas test for permit to use in hazardous areas. Shielding may be required in the presence of bright sunlight, steam, or heat waves. Expensive. Recalibration recommended every two years. Severe misalignment can sometimes exceed range of system, requiring preliminary alignment by other means. For basic equipment, computer gives numbers and their orientation on a display representing the machine elements but no graphical shaft centerline relationship display. The latter is available in supplemental equipment at extra cost.

The OPTALIGN® laser-optic alignment system is manufactured by Prueftechnik Dieter Busch + Partner GmbH & Co., Ismaning, Germany. Represented in the U.S. by Ludeca, Inc., Miami, Florida.



FACE AND RIM



REVERSE INDICATOR

Figure 19-3 A & B. Two most common methods of indicator alignment.

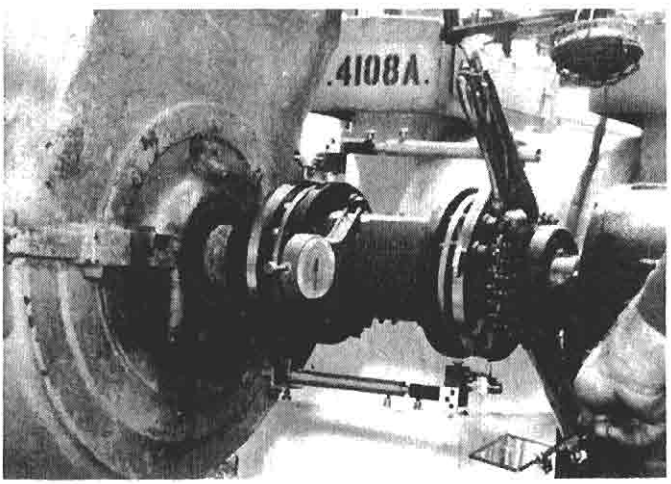


Figure 19-4. Reverse indicator alignment measurement set-up. Note the locking chain wrenches for ease of turning, and the inclinometers for accurate quadrantal rotation.

Methods of Calculating Alignment Movements

Several methods are available for calculating the required alignment movements using data recorded from the alignment measurements. Table 19-4 describes these, and Figure 19-5 shows the alignment plotting board, a popular graphical calculating device for determining alignment movements.

Jig Posts

For dial indicator measurements, it is sometimes useful to use auxiliary surfaces, sometimes called jig posts. These are most often used for rim measurements, but if necessary, can be used for face measurements. They accomplish the task of moving the measurement surface to a more convenient location rather than measuring directly on the shaft or coupling surface. Also, if the jig post has a flat surface, it eliminates curvature error that can sometimes reduce measurement accuracy on small diameters.

Table 19-4
Methods of Calculating Alignment Movements

Method	Advantages	Disadvantages
Nelson's Method	Simple pencil and paper math, using face-and-rim data. Easy to visualize and figure. Accurate.	Applicable only to face-and-rim measurement. A four-move method, less efficient than two-move methods. The second and fourth moves may reverse some of the corrections applied in the first and third moves.
Two-Step Mathematical Methods	Pencil and paper mathematics, using any type of measurement data. More efficient than Nelson's method. Accurate.	Harder to visualize than Nelson's method. More likelihood of calculation error due to incorrect choice of sign.
Graph Paper Plot	Pencil, straightedge, and graph paper, using any type of measurement data. Easier to visualize than straight mathematical methods. Can use on multi-element trains for optimum move calculations.	Slightly less accurate than straight mathematical methods, but not usually enough so to matter. Awkward to use in field.

Table 19-4 Continued
Methods of Calculating Alignment Movements

Method	Advantages	Disadvantages
Plotting Board with Overlay	Graphical method for any type of measurement data using plastic laminated graph and marked overlay (See Figure 19-35 and References 6 and 8.) More durable than graph paper and easier to use under field conditions. Easy visualization of shaft centerline relationships. Can use on two-element optimum-move problems.	More expensive than paper. Not practical to use for simultaneous optimum moves of three or more elements. Not as easy to learn as electronic calculator methods.
Barton Method	Progressive error reduction method. (See Reference 6.) Calculations can be done in one's head. No formal data recording required. Fast and simple, if machine can be adjusted easily and precisely with jackscrews.	Practical only for horizontal corrections. Requires a series of moves so good jackscrews must be present. Also requires reverse-indicator bracket arrangement with shafts that turn easily.
Electronic Calculators	Fast, easy to use, and accurate, for basic and some advanced calculations.	Most will not handle two or multiple element optimum move problems without supplemental calculations. Expensive. Electrical, usually battery operated. May require gas test for use in hazardous areas. In some models, programming can be lost while changing batteries, or if batteries go dead. Can be damaged if not handled carefully.
Combination Measurement/ Calculation Methods— OPTALIGN®	Fast. Easy to use. Accurate. (See Table 19-3 for more details.)	Expensive. Careful handling required. Electrical. (See Table 19-3 for more details.)

Figure 19-5. Machinery alignment plotting board used to determine corrective alignment movements.

Jig posts, although useful, can also cause problems if not handled properly. Their flat measurement surfaces can introduce an error in the case of a gear or gridmember coupling turned in such a way that backlash is present. The same problem would be present in a jig post with a curved measurement surface, if the surface is not concentric with the shaft center.

Another common source of jig post error is transverse inclined plane effect. To avoid this, level the rim measurement surfaces of both posts in coordination and rotate precise 90° quadrants using an inclinometer. With sleeve bearing machines, axial inclined plane effect may also be present. To eliminate this, the post surfaces must be axially parallel to the machine shafts. For face measurements on jig posts, the face surfaces must be parallel to the coupling faces, that is, perpendicular to the shafts in two 90° planes.

There are two ways to eliminate these last two inclined-plane errors. One is by using precisely machined jig posts that give the desired result automatically when mounted on shafts. The other way is to use tri-axially adjustable jig posts, with a leveling procedure that achieves the desired

parallelism/perpendicularity. The T + B vs. S + S test, which is described in Murray [6], will detect inclined-plane error if this is present.

Numerical Examples

Numerical examples will not be given here due to lack of space to give proper coverage. Those wishing to examine a variety of such examples are urged to see Murray [6].

Thermal Growth

When machines are operating, their temperatures usually change from that at which they were aligned. In most cases they get hotter, although the opposite is occasionaly true. In the case of pumps, turbines, and compressors, the temperature of the connected piping also changes. These temperature changes cause expansion or contraction in the metal. In most cases, this expansion or contraction is not uniform and differs significantly for driver and driven machines. Their relative positions therefore change during operation, causing changes in the alignment relationship. In addition to temperature effects, torque, hydraulic effects, and oil film thickness may contribute to alignment changes. Because of these factors, a coupled machine train with good alignment in its "cold" or non-operating state may have less precise alignment under operating conditions.

For a pump that operates at conditions subject to little temperature variation and aligned at an ambient temperature that also varies little, the relationship between cold and hot alignment will be constant and repeatable. If the amounts and directions of growth can be determined, they can be used in the form of thermal offsets to include a deliberate misalignment when the machines are aligned. Then, under operating conditions, the growth will act equal and opposite to this misalignment, causing the machines to become aligned during operation.

So much for the theory. We live in an imperfect world, and things do not usually behave exactly as we might wish. The best laid plans of mice and men oft go astray, and thermal growth offsets are a good example of this.

There are two common approaches. One, favored by certain authorities, is to use formulas or rules of thumb involving fluid or metal temperatures, coefficients of thermal expansion, and machine geometric measurements to calculate predicted growths. These are then converted to offsets and incorporated into the initial machine alignment. If the machine runs well, with low vibration and without premature failure, the offsets are assumed to be correct, and are used thereafter. If the machine does not run satisfactorily, and the problem is attributed to misalignment caused by thermal growth other than that used to derive the offsets, the

growth is then measured by some means, and the offsets are revised accordingly.

The other approach, favored by this author for most pumps (but not necessarily for large turbomachinery), is simpler. This is to say that formulas and rules of thumb are unreliable because they fail to account for piping growth effects that may influence machine movement more than the growth of the machines themselves. The formulas also fail to account for uneven temperature rises in different parts of the machine frames, which is quite common. In effect, formulas are unlikely to give good figures for amounts of relative growth and are equally unlikely to get the *directions* of such relative growths correct. This being the case, our most conservative course of action, as well as the easiest, is to align everything to zero offset, cold. Then, at least, the machines are starting from a good alignment condition, and relative growth in one direction is no more harmful than in another. If alignment-related vibration and reliability prove satisfactory, as they do about 90% of the time, we have simplified our alignment task by eliminating offsets. In the other 10% of the cases, the growths should be *measured,* not calculated, and offsets applied. Several methods of growth measurement are described in Table 19-5.

Table 19-5
Methods for Determining Thermal Growth Offsets for Machinery Alignment

Method	Advantages	Disadvantages
Manufacturers' Recommendations	When available, these require little effort to obtain. Sometimes they give good results.	Often not available, and often unreliable when they *are* available.
Calculations based on temperatures, thermal expansion coefficients, and machine geometry, using formulas and rules of thumb.	Easy to obtain, and very scientific-looking.	Unreliable. Likely to do more harm than good by getting the relative growth direction wrong and thus increasing the misalignment. Formulas fail to account for nonuniform temperature rise in machines and effects of temperature rise in connected piping.
Shut down and "quickly" take alignment measurements.	Makes the measurer feel he is doing something worthwhile.	Utterly useless and self-deceiving. The majority of the thermal movement occurs during the first

Table 19-5 continued
Methods for Determining Thermal Growth Offsets for Machinery Alignment

Method	Advantages	Disadvantages
		minute after shutdown—too quickly to obtain good measurements.
Use extension gauges to take triangulation measurements on tooling balls at several planes—Essinger/ Acculign (See Reference 3.)	Consistent results obtained if lower tooling balls are mounted on foundation or sole plates rather than on baseplate. Easy to apply, with straightforward calculations based on measurements taken at hot running and cold shutdown conditions. Electronic or non-electronic as desired, and non-optical; fairly rugged tooling, not overly expensive.	Piping congestion can sometimes make access difficult. Some loss of accuracy due to inability to measure to center of shaft—usually must stop short of this by about 15%.
Eddy current proximity probes mounted in water stands. (See Reference 5.)	Good for ground mounted machines. Accurate results if applied properly. Can also be applied to elevated foundation machines. The mount bases should be on foundations or soleplates, not on baseplates or structural steel.	Fairly bulky. Electronic.
Similar probes mounted in Dodd/Dynalign® bars. (See Reference 2.)	Good for both ground mounted and elevated machines, since measurements are relative, not ground based. Accurate results if applied properly.	Electronic. May run into span limitations with long coupling spacers.
Similar probes used in Indikon system. (See Reference 4.)	Effectively measures to center of rotating shafts, giving highly accurate results. No coupling span limitations. Good for both ground-mounted and	Electronic. Expensive. Built into coupling. Not easy to retrofit to existing machines.

(table continued on next page)

Table 19-5 Continued
Methods for Determining Thermal Growth Offsets for Machinery

Method	Advantages	Disadvantages
	elevated machines. Similar system can measure machine torque using strain gauges mounted in coupling.	
Optical Measurement	Non-electronic. Accurate if done properly. Can be used for both ground-mounted and elevated machines. No temperature limits.	Delicate instruments, fairly expensive. Considerable skill required.
Laser Measurement (See Reference 13.)	Accurate.	Expensive. Special mounts improve economics for multiple installations and accuracy.
Vernier-Strobe (See Reference 13.)	Special Vernier scales attached to coupling are read or photographed with aid of strobe light. Highly accurate and fairly inexpensive.	Limited to use with certain types of couplings, mainly metal disk and grease lubricated gear types.
Alignment adjustment while running in response to vibration.	Growth measurement bypassed, going instead to effects on running vibration.	Risk of machine damage if moves are not well controlled.

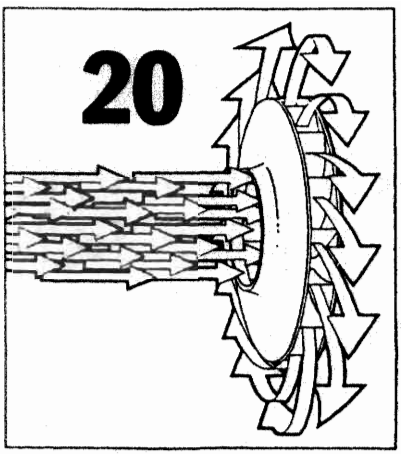
® *Dymac, a unit of SKF USA, Inc., San Diego, California.*

Considerably more could be said about machinery alignment than we have room for here. Those wishing to learn more are urged to investigate the references listed.

References

1. Bloch, H. and Geitner, F., *Practical Machinery Management for Process Plants, Vol. 3, Machinery Component Maintenance and Repair,* Houston: Gulf Publishing Company, 1985, pp. 159–224 and 106–114.
2. Dodd, V. R., *Total Alignment,* Tulsa: PennWell Publishing Company, 1975, pp. III-1–39.

3. Essinger, J. N., *Benchmark Gauges for Effective Hot Alignment of Turbomachinery,* Houston: Acculign, Inc., undated.
4. *Indikon Measuring Systems—Misalignment,* Cambridge, Massachusetts: Indikon Company, Inc., 1977.
5. Jackson, C., "Shaft Alignment Using Proximity Transducers—ASME Paper 68-PET-25," New York: The American Society of Mechanical Engineers, 1968.
6. Murray, M. G., *Alignment Manual for Horizontal, Flexibly-Coupled Rotating Machines,* 3rd Edition, Baytown, Texas: Murray & Garig Tool Works, 1983.
7. _____, "Flange Fitup Problems? Try a Dutchman," *Hydrocarbon Processing,* Vol. 59, No. 1, January 1980, pp. 105–106.
8. _____, *Machinery Alignment Plotting Board,* Baytown, Texas: Murray & Garig Tool Works, 1980
9. _____, *Machinery Alignment Postioners,* Baytown, Texas: Murray & Garig Tool Works, 1978.
10. _____, "Machinery Shaft/Coupling Alignment Strategy Cuts Manhours and Downtime," *Oil & Gas Journal,* Vol. 81, No. 28, July 11, 1983, pp. 55–61.
11. _____, "Out of Room? Use Minimum Movement Machinery Alignment," *Hydrocarbon Processing,* Vol. 58, No. 1, January 1979, pp. 112–114.
12. _____, "Practical Considerations for Pump Baseplates, Grouting and Foundations," *3rd International Pump Symposium Proceedings,* 1986, pp. 131–136.
13. _____, "Two New Systems for Measuring Alignment Thermal Growth," *Vibrations,* Vol. 6, No. 4, Willowbrook, Illinois: The Vibration Institute, December 1990, pp. 9–15.
14. Piotrowski, J., *Shaft Alignment Handbook,* New York: Marcel Dekker, Inc., 1986.



Rolling Element Bearings and Lubrication*

by **Heinz P. Bloch, P.E.**
Consultanting Engineer

Next to mechanical seal failures, bearing failures are most often responsible for pump outages and repair events. Both rolling element (antifriction) bearings and sliding element (plain) bearings are used in centrifugal pumps. Each of these two bearing categories is found in small as well as large and in single as well as multistage pumps. However, very large or very high speed pumps tend to favor sliding element bearings over rolling element bearings.

Because the preponderance of centrifugal pumps is small, say less than 200 hp (~ 150 kW), most pumps are equipped with rolling element bearings. Avoiding or reducing rolling element bearing failures should thus be a prime topic whenever pump life extensions are being discussed.

A rolling element bearing is a precision device and a marvel of engineering. It is unlikely that any other mass-produced item is machined to such close tolerances. While boundary dimensions are usually held to tenths of a thousandth of an inch, rolling contact surfaces and geometries are maintained to millionths of an inch. It is for this obvious reason that very little surface degeneration can be tolerated [12].

The life of a rolling element bearing running under good operating conditions is usually limited by fatigue failure rather than by wear. Under optimum operating conditions, the fatigue life of a bearing is determined by the number of stress reversals and by the cube of the load causing these stresses. As example, if the load on the bearing is doubled, the the-

* Material on lubrication of rolling element bearings adapted by permission from TRW Bearings Division, Jamestown, New York.

oretical fatigue life is reduced to one-eighth. Also, if speed is doubled, the theoretical fatigue life is reduced to one-half.

Friction Torque

The friction torque in a rolling element bearing consists essentially of two components. One of these is a function of the bearing design and the load imposed on the bearing. The other is a function of the lubricant type, lubricant quantity, and the speed of the bearing.

It has been found that the friction torque in a bearing is lowest with a quantity of the correct viscosity oil just sufficient to form a film between the contacting surfaces. This is just one of the reasons why oil mist (to be explained later) is a superior lubrication method. The friction will increase with greater quantity and/or higher viscosity of the oil. With more oil than just enough to separate the rolling elements, the friction torque will also increase with the speed.

Function of the Lubricant

A bearing lubricant serves to accomplish the following:

- To lubricate sliding contact between the cage and other parts of the bearing.
- To lubricate any contact between races and rolling elements in areas of slippage.
- To lubricate the sliding contact between the rollers and guiding elements in roller bearings.
- In some cases, to carry away the heat developed in the bearing.
- To protect the highly finished surfaces from corrosion.
- To provide a sealing barrier against foreign matter.

Oil Versus Grease

The ideal lubricant for rolling element bearings is oil. Grease, formed by combining oil with soap or non-soap thickeners, is simply a means of effecting greater utilization of the oil. In a grease, the thickener acts fundamentally as a carrier and not as a lubricant.

Although realtively few centrifugal pump bearings are grease lubricated, greases are in fact used for lubricating by far the largest overall number of rolling bearings. The extensive use of grease has been influenced by the possibilities of simpler housing designs, less maintenance, less difficulty with leakage, and better sealing against dirt. On the other hand, there are limitations that do not permit the use of grease. Where a

lubricant must dissipate heat rapidly, grease should not be used. In many cases, associated machine elements that are oil lubricated dictate the use of oil for rolling element bearings. Listed below are some of the advantages and disadvantages of grease lubrication.

Advantages

- Simpler housing designs are possible; piping is greatly reduced or eliminated.
- Maintenance is greatly reduced since oil levels do not have to be maintained.
- Being a solid when not under shear, grease forms an effective collar at bearing edges to help seal against dirt and water.
- With grease lubrication, leakage is minimized where contamination of products must be avoided.
- During start-up periods, the bearing is instantly lubricated whereas with pressure or splash oil systems, there can be a time interval during which the bearing may operate before oil flow reaches the bearing.

Disadvantages

- Extreme loads at low speed or moderate loads at high speed may create sufficient heat in the bearing to make grease lubrication unsatisfactory.
- Oil may flush debris out of the bearing. Grease will not.
- The correct amount of lubricant is not as easily controlled as with oil.

Oil Characteristics

The ability of any oil to meet the requirements of specific operating conditions depends upon certain physical and chemical properties. In cold ambients, the lube oil must have a low enough pour point to ensure that the oil remains in the liquid condition, whereas in hot ambients, the lube oil must have a high enough viscosity to ensure that the rolling elements are coated with an oil film of sufficient thickness to prevent metal-to-metal contact.

Viscosity

The single most important property of oil is viscosity. It is the relative resistance to flow. A high viscosity oil will flow less readily than a thinner, low viscosity oil.

Viscosity can be measured by any of a number of different instruments that are collectively called viscosimeters. In the United States, the viscos-

ity of oils is usually determined with a Saybolt Universal Viscosimeter. It simply measures the time in seconds required for 60 cc of oil to drain through a standard hole at some fixed temperature. The common temperatures for reporting viscosity are 100°F and 210°F (38°C and 99°C). Viscosities are quoted in terms of Saybolt Universal Seconds (SUS).

Generally, for ball bearings and cylindrical roller bearings, it is a good rule to select an oil that will have a viscosity of at least 13.1 cSt or 70 SUS at operating temperature. It is especially important to observe this guideline in hot climates where it would be prudent to opt for ISO Grade 100 lube oils whenever possible.

Viscosity Index

All oils are more viscous when cold and become thinner when heated. However, some oils resist this change of viscosity more than others. Such oils are said to have a high viscosity index (V.I.). Viscosity index is most important in an oil that must be used in a wide range of temperatures. Such an oil should resist excessive changes in viscosity. A high V.I. is usually associated with good oxidation stability and can be used as a rough indication of such quality.

Pour Point

Any oil, when cooled, eventually reaches a temperature below which it will no longer flow. This temperature is said to be the pour point of the oil. At temperatures below its pour point, an oil will not feed into the bearing, and lubricant starvation may result. In selecting an oil for rolling element bearings, the pour point must be considered in relation to the operating temperature. In other words, equipment operating at 10°F should be lubricated with oils that have a pour point of zero or perhaps even −10°F. It should be noted that many diester or polyalpha-olefin synthetic lubricants have pour points in the vicinity of −60°F and are thus well suited for low temperature operation.

Flash and Fire Point

As an oil is heated, the lighter fractions tend to become volatile and will flash off. With any oil, there is some temperature at which enough vapor is liberated to flash into momentary flame when ignition is applied. This temperature is called the flash point of the oil. At a somewhat higher temperature, enough vapors are liberated to support continuous combustion. This is called the fire point of the oil. The flash and fire points are significant indications of the tendency of an oil to volatilize at

high operating temperatures. High V.I. oils generally have higher flash and fire points than lower V.I. oils of the same viscosity.

Oxidation Resistance

All petroleum oils are subject to oxidation by chemical reaction with the oxygen contained in air. This reaction results in the formation of acids, gum, sludge, and varnish residues that can reduce bearing clearances, plug oil lines, and cause corrosion.

Some lubricating fluids are more resistant to this action than others. Oxidation resistance depends upon the fluid type, the methods and degree of refining used, and whether oxidation inhibitors are used.

There are many factors that contribute to the oxidation of oils and practically all of these are present in lubricating systems. These include temperature, agitation, and the effects of metals and various contaminants that increase the rate of oxidation.

Temperature is a primary accelerator of oxidation. It is well known that rates of chemical reaction double for every 18°F (10°C) increase in temperature.

Below 140°F (60°C), the rate of oxidation of oil is rather slow. Above this temperature, however, the rate of oxidation increases to such an extent that it becomes an important factor in the life of the oil. Consequently, if oil systems operate at temperatures above 140°F (60°C), more frequent oil changes would be appropriate. Figure 20-1 shows recommended oil replacement frequencies as a function of oil sump or reservoir capacity.

The oxidation rate of oil is accelerated by metals such as copper and copper-containing alloys and to a much lesser extent by steel. Contaminants such as water and dust also act as catalysts to promote oxidation of the oil.

Emulsification

Generally, water and straight oils do not mix. However, when an oil becomes dirty, the contaminating particles act as agents to promote emulsification. In rolling element bearing lubricating systems, emulsification is undesirable. Therefore, the oil should separate readily from any water present, which is to say, it should have good demulsibility characteristics.

Rust Prevention

Although straight petroleum oils have some rust protective properties, they cannot be depended upon to do an unfailing job of protecting rust-

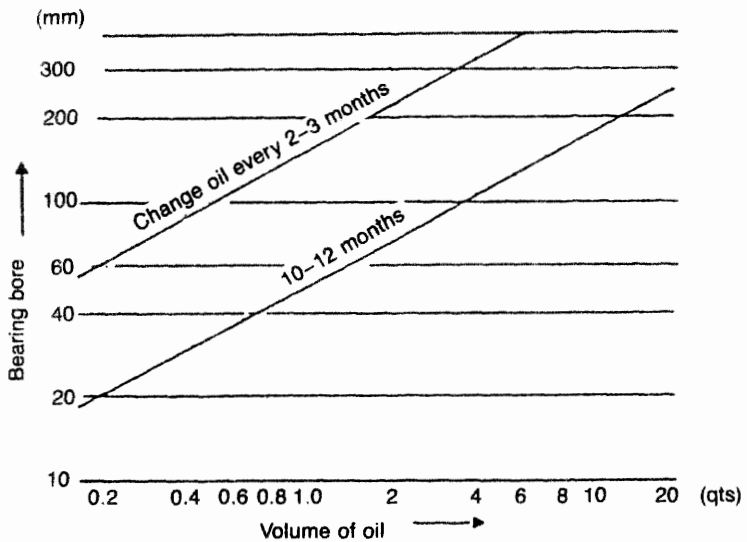


Figure 20-1. Recommended lube oil change frequency as a function of bearing bore diameter and lube oil sump capacity (courtesy of FAG Bearing Corporation, Stamford, Connecticut).

susceptible metallic surfaces. In many instances, water can displace the oil from the surfaces and cause rusting. Rust is particularly undesirable in a rolling element bearing because it can seriously abrade the bearing elements and areas pitted by rust will cause rough operation or failure of the bearing.

Additives

High grade lubricating fluids are formulated to contain small amounts of special chemical materials called additives. Additives are used to increase the viscosity index, fortify oxidation resistance, improve rust protection, provide detergent properties, increase film strength, provide extreme pressure properties, and sometimes to lower the pour point.

General Lubricant Considerations

The proper selection of a grease or oil is extremely important in high-speed bearing operation. Proper lubrication requires that an elastohydrodynamic oil film be established and maintained between the bearing rotating members. To ensure the build-up of an oil film of adequate load-carrying capacity, the lubricating oil must possess a viscosity adequate to withstand the given speed, load, and temperature conditions.

These requirements would probably be best satisfied by a different lubricant for each specific application. For the relatively unsophisticated rolling element bearing service found in the typical centrifugal pump, it is often possible to economize by stocking only a few lubricant grades. Maintaining a minimum base oil viscosity of 70 Saybolt Universal Seconds (SUS) or 13.1 centistokes (cSt) has long been the standard recommendation of many bearing manufacturers. It was applied to most types of ball and some roller bearings in electric motors with the understanding that bearings would operate near their published maximum rated speed, that naphthenic oils would be used, and that the viscosity be no lower than this value even at the maximum anticipated operating temperature of the bearings [3].

Figure 20-2 shows how higher viscosity grade lubricants will permit higher bearing operating temperatures. ISO viscosity grade 32 (147 SUS @ 100°F or 28.8–35.2 cSt @ 40°C) and grade 100 (557 SUS @ 100°F or 90–110 cSt @ 40°C) are shown on this chart. It shows a safe allowable temperature of 146°F (63°C) for rolling element bearings with grade 32 lubrication. Switching to grade 100 lubricant and requiring identical bearing life, the safe allowable temperature would be extended to 199°F (93°C). If a change from grade 32 to grade 100 lube oil should cause the bearing operating temperature to reach some intermediate level, a higher oil viscosity would result and the bearing life would actually be extended. Most ball and roller bearings can be operated satisfactorily at temperatures as high as 250°F (121°C) from the metallurgy point of view. The only concern would be the decreased oxidation resistance of common lubricants, which might require more frequent oil changes. However, the "once-through" application of oil mist solves this problem. Therefore, oil mist lubricated anti-friction bearings are often served by ISO grade 100 naphthenic oils. Where these oils are unavailable or in extremely low ambients, dibasic ester-based synthetic lubes have been very successfully applied. These synthetics eliminate the risk of wax plugging that has sometimes been experienced with mineral oils at low temperatures.

When applying greases to lubricate pump bearings, certain precautions are in order. Soft, long-fibered type greases, or excessively heavy oils, will result in increased churning friction at higher speeds, causing bearing overheating due to the high shear rate of these lubricants. Excessive amounts of lubricant will also create high temperatures.

Using oils of adequate film strength, but light viscosity, or using channeling or semi-channeling greases has the benefit of substantially reducing the heat-generating effects of lubricants. The advantages of these greases rest in their ability to "channel" or be pushed aside by the rotating ball or roller elements of a bearing and to lie essentially dormant in the side cavities of the bearing or housing reservoir. Channeling greases normally are "short-fibered" and have a buttery consistency that imparts

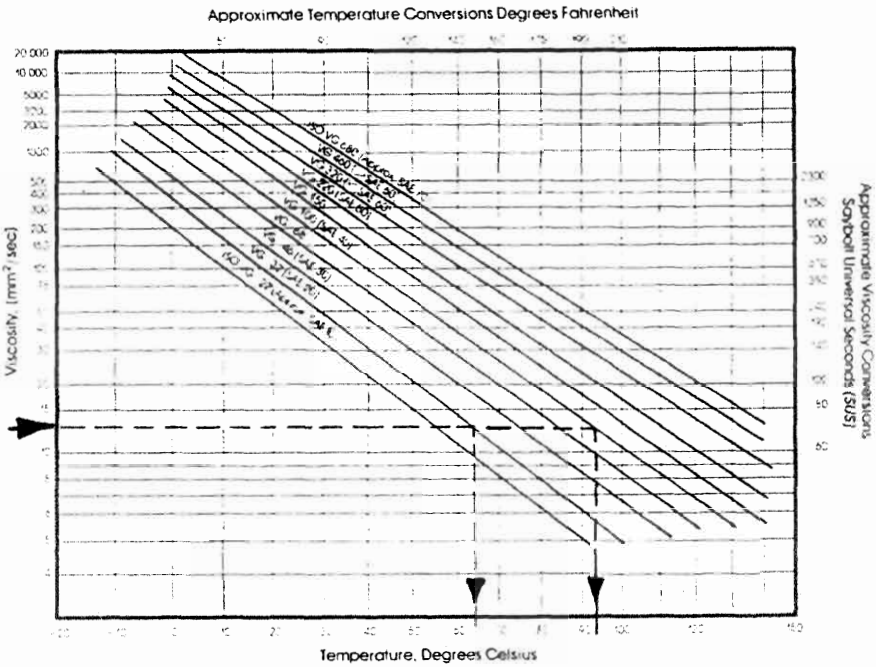


Figure 20-2. Temperature vs. viscosity relationships for various lube oil grades. Note that centrifugal pumps are typically lubricated with ISO grades 32 through 100.

a low shear rate to the lubricant. This low temperature aids an operating bearing to establish a temperature equilibrium, even if a lubricant is applied having a slightly higher viscosity than the application demands. Higher fluid friction increases the temperature of the lubricant until the viscosity is reduced to the proper level. It should be noted, however, that short-fibered greases may lead to "false brinelling" damage in applications subject to vibration without equipment rotation.

Greases generally are applied where oils cannot be used, for example, in places where sealing does not exist or is inadequate, dirty environments, in inaccessible locations, in places where oil dripping or splashing cannot be tolerated, and often, where "sealed-for-life" lubrication is desired.

Greases are fine dispersions of oil-insoluble thickening agents—usually soap—in a fluid lubricant such as a mineral oil. When a bearing lubricated with grease starts to move, the grease structure (created by the thickening agent) is affected by the shearing action, and lubrication is provided by the oil component of the grease. As the bearing slows to a

stop, the grease regains its semi-solid structure. In non-moving parts of the bearing, this structure does not change.

The type and amount of the thickener, additives used, the oil, and the way in which the grease is made can all influence grease properties. The chosen base-oil viscosity generally matches that for a fluid lubricant used for the same service—low-viscosity oil for light loads, fast speeds and low temperatures; high-viscosity oils for differing conditions. The thickener will determine grease properties such as water resistance, high-temperature limit, resistance to permanent structural breakdown, "stay-put" properties, and cost.

Greases are classified on the basis of soap (or thickener) type, soap content, dropping point, base oil viscosity, and consistency. Consistency mainly is a measure of grease-sealing properties, dispensability, and texture. Once the grease is in a bearing, consistency has little effect on performance. But despite this, greases are widely described primarily on the basis of consistency [5].

Sodium-soap greases are occasionally used on small pump bearings because of their low torque resistance, excellent high-temperature performance, and ability to absorb moisture in damp locations. Since all sodium soaps are easily washed out by water sprays, they should not be employed where splashes of water are expected.

Lithium-soap greases generally are water resistant and corrosion inhibiting, and have good mechanical and oxidation stability. Many automobile manufacturers specify such grease—often with additives to give wide protection against problems caused by shipment, motorist neglect, and the extended lubrication intervals now popular. Widely used in centralized lubrication systems, these versatile greases are also favored in both sliding and rolling element bearings.

Simple calcium-soap greases resist water-washout, are noncorrosive to most metals, work well in both grease cups and centralized lubrication systems, and are low-cost lubricants. They are, dependent on manufacture and ingredients, limited to services $< 160°F$ to $200°F$ ($71°C$ to $93°C$).

Complex calcium-soap greases, wisely applied, can provide multi-purpose lubrication at a fraction of the cost of a lithium-soap grease; however, misapplication of these greases will more likely cause more difficulty than the same error committed with lithium greases.

Special-purpose greases are available for food processing (both the thickener and oil are nontoxic), fine textile manufacture (light colors for nonstaining, or adhesive grades to avoid slingoff), rust prevention, and other special services.

The user is well advised to observe the grease compatibility constraints indicated in Table 20-1. Experience shows that many pumps and electric motor drivers are initially factory-lubricated with polyurea greases to

Table 20-1
Results of Grease Incompatibility Study

	Aluminum Complex	Barium	Calcium	Calcium 12-hydroxy	Calcium Complex	Clay	Lithium	Lithium 12-hydroxy	Lithium Complex	Polyurea
Aluminum Complex	X	I	I	C	I	I	I	I	C	I
Barium	I	X	I	C	I	I	I	I	I	I
Calcium	I	I	X	C	I	C	C	B	C	I
Calcium 12-hydroxy	C	C	C	X	B	C	C	C	C	I
Calcium Complex	I	I	I	B	X	I	I	I	C	C
Clay	I	I	C	C	I	X	I	I	I	I
Lithium	I	I	C	C	I	I	X	C	C	I
Lithium 12-hydroxy	I	I	B	C	I	I	C	X	C	I
Lithium Complex	C	I	C	C	C	I	C	C	X	I
Polyurea	I	I	I	I	C	I	I	I	I	X

B = Borderline Compatibility
C = Compatible
I = Incompatible

Derived from a paper given by E. H. Meyers entitled "Incompatibility of Greases" at the NLGI's 49th Annual Meeting.

prevent oxidation damage during storage. Obviously, this grease should not be mixed with many of the typical greases used by modern process plants.

Application Limits for Greases

Bearings and bearing lubricants are subject to four prime operating influences: speed, load, temperature, and environmental factors. The optimal operating speeds for ball and roller type bearings—as related to lu-

brication—are functions of what is termed the DN factor. To establish the DN factor for a particular bearing, the bore of the bearing (in millimeters) is multiplied by the revolutions per minute, that is:

$$75\text{mm} \times 1000 \text{ rpm} = 75{,}000 \text{ DN value}$$

Speed limits for conventional greases have been established to range from 100,000 to 150,000 DN for most spherical roller type bearings and 200,000 to 300,000 DN values for most conventional ball bearings. Higher DN limits can sometimes be achieved for both ball and roller type bearings, but require close consultation with the bearing manufacturer. When operating at DN values higher than those indicated above, use either special greases incorporating good channeling characteristics or circulating oil. Typical relubrication intervals are given in Figure 20-3.

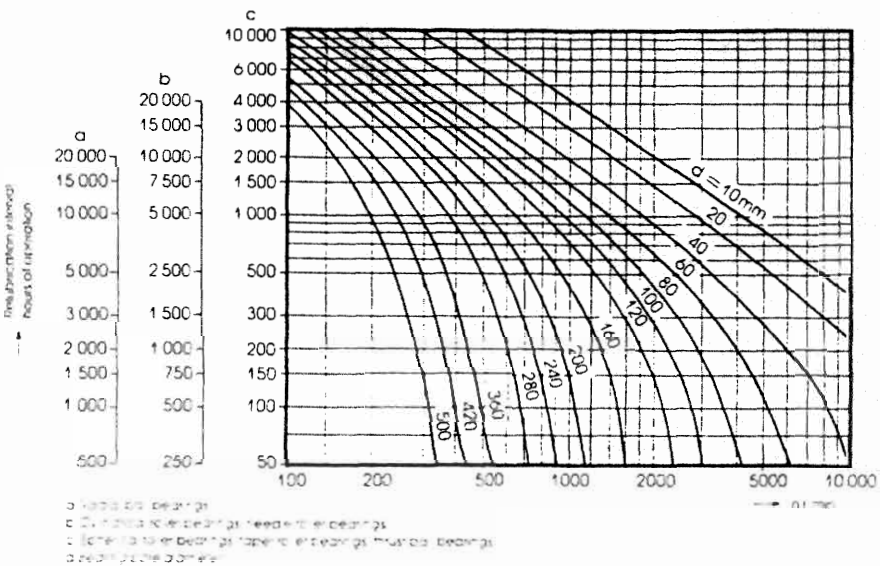


Figure 20-3. Relubrication intervals recommended by SKF Bearing Company.

Life-Time Lubricated, "Sealed" Bearings

As shown in Table 20-2, "lubrication for life" ranks last in order of preference [8]. Lubed-for-life bearings incorporate close-fitting seals in place of, or in addition to shields. These bearings are customarily found on low horsepower motors or on appliances that operate intermittently.

Table 20-2
Influence of Lubrication On Service Life
(Source: FAG Bearing Corporation)

	Oil	Oil	Grease	Dry lubricant
	Rolling bearing alone	Rolling bearing with gearwheels and other wearing parts	Rolling bearing alone	Rolling bearing alone
Decreasing service life**	Circulation with filter, automatic oiler Oil-air Oil-mist Circulation without filter*	Circulation with filter Oil-air Oil-mist Circulation without filter*	Automatic feed	
	Sump, regular renewal	Sump, regular renewal*	Regular regreasing of cleaned bearing	
		Rolling bearing (a) in oil vapour (b) in sump (c) oil circulation	Regular grease replenishment	
	Sump, occasional renewal	Sump, occasional renewal		Regular renewal
		Rolling bearing (a) in oil vapour	Occasional renewal	
		(b) in sump	Occasional replenishment	
		(c) oil circulation	Lubrication for-life	
				Lubrication for-life

* By feed cones, bevel wheels, asymmetric rolling bearings.
** Condition: Lubricant service life ≪ Fatigue life.

However, at least one large petrochemical company in the United States has expressed satisfaction with sealed ball bearings in small centrifugal pumps as long as bearing operating temperatures remained below 150°C (302°F) and speed factors DN (mm bearing bore times revolutions per

minute) did not exceed 300,000 [15]. Studies showed, on the other hand, that close-fitting seals can cause high frictional heat and that loose fitting seals cannot effectively exclude atmospheric air and moisture that will cause grease deterioration. It should be assumed that these facts preclude the use of lubed-for-life bearings in installations that expect "life" to last more than three years in the typical petrochemical plant environment. Moreover, we believe this to be the reason why some bearing manufacturers advise against the use of sealed bearings at DN values in excess of 108,000. At least one bearing manufacturer considers the DN range between 80,000 and 108,000 the "gray" area, where either sealed or open (shielded) bearing would be acceptable. The DN range below 80,000 is generally considered safe for sealed bearings. This would generally exclude sealed bearings from all but the smallest centrifugal pumps.

Oil Viscosity Selection

The Oil Viscosity Selection Chart (Figure 20-2) may be used as a guide for the selection of the proper viscosity oil. The chart may be used for applications where loads are light to moderate with moderately high conditions of speed and temperature.

Applications of Liquid Lubricants in Pumps

The amount of oil needed to maintain a satisfactory lubricant film in a rolling element bearing is extremely small. The minimum quantity required is a film averaging only a few micro-inches in thickness. Once this small amount has been supplied, make-up is required only to replace the losses due to vaporization, atomization, and seepage from the bearing surfaces [12].

How small a quantity of oil is required can be realized when we consider that 1/1000 of a drop of oil, having a viscosity of 300 SUS at 100°F (38°C) can lubricate a 50 mm bore bearing running at 3,600 RPM for one hour. Although this small amount of oil can adequately lubricate a bearing, much more oil is needed to dissipate heat generated in high speed, heavily loaded bearings.

Oil may be supplied to rolling element bearings in a number of ways. These include bath oiling, oil mist from an external supply, wick feed, drip feed, circulating system, oil jet, and splash or spray from a slinger or nearby machine parts.

One of the simplest methods of oil lubrication is to provide a bath of oil through which the rolling element will pass during a portion of each revolution. It can be shown that only a few drops of oil per hour would satisfy the lubrication requirements of a typical rolling element bearing in

a centrifugal pump. However, unless frictional heat generated by the bearing is readily dissipated, the lubricating oil must serve also as a coolant.

The MRC Bearing Division of TRW calculates the theoretical oil flow required for cooling from the expression.

$$Q = \frac{.000673 \times Fr \times P \times PD \times RPM}{H_s \times (T_o - T_i)} \text{ lbs/minute}$$

where Fr = Coefficient of friction referred to PD
- .00076 = cylindrical roller bearings
- .00089 = pure thrust ball bearings
- .00103 = radial ball bearings
- .00152 = angular contact ball bearings
- .002–.005 = tapered roller bearings

P = imposed equivalent load in pounds
PD = pitch diameter in inches
RPM = operating speed in revolutions per minutes
T_o = outlet oil temperature in °F
T_i = inlet oil temperature in °F ($T_o - T_i$, generally about 50°F)
H_s = specific heat of oil in BTU/LB/°F (usually .46 – .48)
= .195 + .000478 (460 + T_i)

Conversion: (pounds of oil per minute) × (0.135) = gal/min

Hence, where cooling is required in high speed and heavily loaded bearings, oil jets and circulating systems should be considered. If necessary, the oil can be passed through a heat exchanger before returning to the bearing.

Oil Bath Lubrication

A simple oil bath method is satisfactory for low and moderate speeds. The static oil level should not exceed the center line of the lowermost ball or roller. A greater amount of oil can cause churning that results in abnormally high operating temperatures. Systems of this type generally employ sight gauges to facilitate inspection of the oil level.

Figure 20-4 shows a constant level arrangement for maintaining the correct oil level. Figure 20-5 shows a similar arrangement that includes a pressure-balanced constant level oiler.

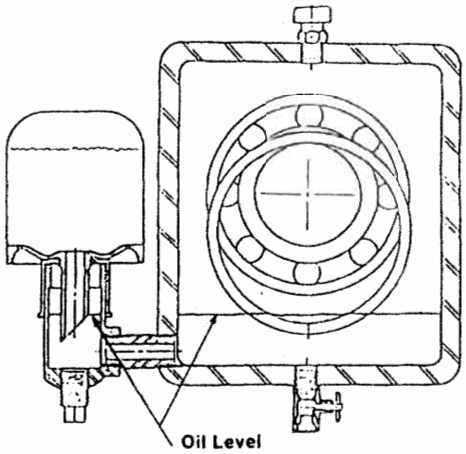


Figure 20-4. Constant level oilers are typically used for conventional lubrication of centrifugal pumps.

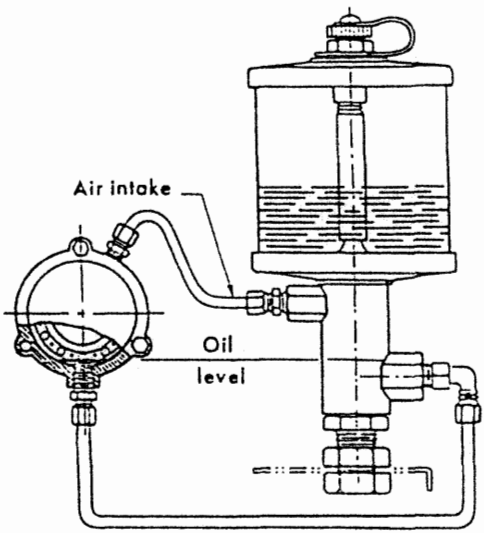


Figure 20-5. Pressure-balanced constant level oilers are required for bearing housings with excessive back pressure or vacuum. They are ideally suited for purge oil mist applications (courtesy of Oil-Rite Corporation, Manitowoc, Wisconsin).

Drip Feed Lubrication

Although not usually found on centrifugal pumps, this system is widely used for small and medium ball and roller bearings operating at moderate to high speeds where extensive cooling is not required. The oil, introduced through a filter-type, sight feed oiler, has a controllable flow rate that is determined by the operating temperature of the bearings.

Figure 20-6 illustrates a typical design and shows the preferred location of the oiler with respect to the bearings.

Forced Feed Circulation

This type of system uses a circulating pump and is particularly suitable for low to moderate speed, heavily loaded applications where the oil has a major function as a coolant in addition to lubrication. If necessary, the oil can be passed through a heat exchanger before returning to the bearing. Entry and exit of the oil should be on opposite sides of the bearing. An adequate drainage system must be provided to prevent an excess accumulation of oil. Oil filters and magnetic drain plugs should be used to minimize contamination.

In applications of large, heavily loaded, high speed bearings operating at high temperatures, it may be necessary to use high velocity oil jets. In such cases the use of several jets on both sides of the bearing provides more uniform cooling and minimizes the danger of complete lubrication loss from plugging. The jet stream should be directed at the opening between the cage bore and inner ring O.D., see Figure 20-7. Adequate scavenging drains must be provided to prevent churning of excess oil after the oil has passed through the bearing. In special cases, scavenging may be required on both sides of the bearing.

At extremely high speeds, the bearing tends to reject the entry of sufficient oil to provide adequate cooling and lubrication with conventional oil jet and flood systems. Figure 20-8 shows an under-race lubrication system with a bearing having a split inner ring with oil slots. This method ensures positive entrance of oil into the bearing to provide lubrication and cooling of the inner ring and may well become a standard on the high-performance pump of the future.

Oil Mist Lubrication

Oil mist lubrication, as shown schematically in Figure 20-9, is a centralized system that utilizes the energy of compressed air to supply a continuous feed of atomized lubricating oil to multiple points through a low pressure distribution system operating at $\approx$ 20 in ($\approx$ 500 mm) H_2O. Oil

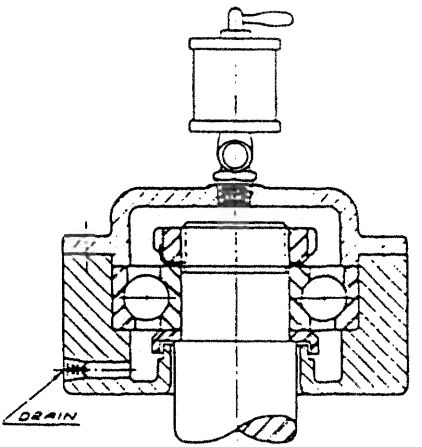


Figure 20-6. Drip feed lubrication occasionally used on small vertical pumps.

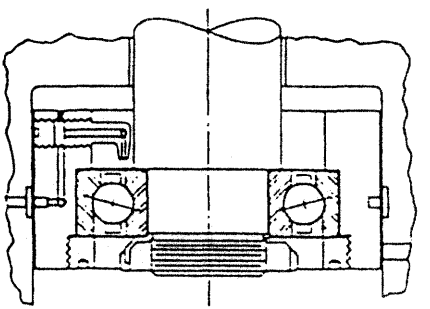


Figure 20-7. Oil jet lubrication for heavily loaded high speed bearings.

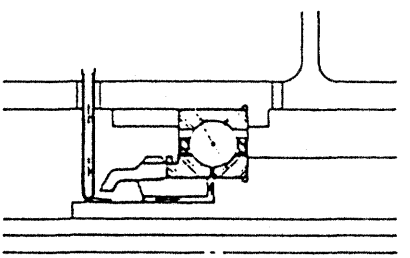


Figure 20-8. Under-race oil jet lubrication applied through split inner ring with oil slots ensures positive entrance of oil into bearing.

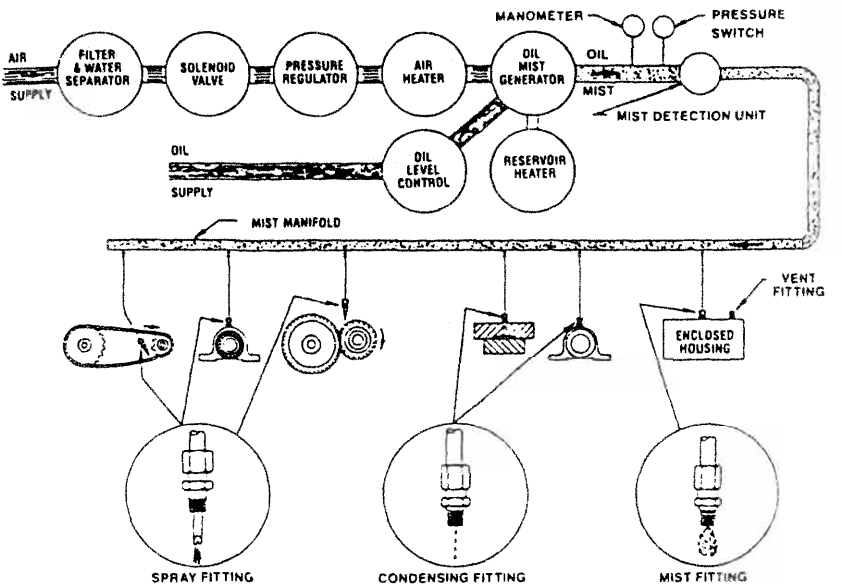


Figure 20-9. Oil mist lubrication schematic (courtesy of Alemite Division of Stewart-Warner Corporation, Chicago, Illinois).

mist then passes through a reclassifier nozzle before entering the point to be lubricated. This reclassifier nozzle establishes the oil mist stream as either a mist, spray, or condensate, depending on the application of the system.

Since the mid-1950s, the oil mist lubrication concept has been accepted as a proven and cost-effective means of providing lubrication for centrifugal pumps. Typical petrochemical plant pump applications are illustrated in Figures 20-10 through 20-13. Centralized oil mist systems have also been highly successful on electric motors, gears, chains, and horizontal shaft bearings such as on steam turbines and steel plant rolling mill equipment.

The actual method of applying oil mist to a given piece of equipment is governed to a large extent by the type of bearing used. For sliding bearings, oil mist alone is not considered an effective means of lubrication because relatively large quantities of oil are required. In this case, oil mist is used effectively as a purge of the oil reservoir and, to a limited extent, as fresh oil make-up to the reservoir. Purge mist lubricated pumps generally use a constant level oiler with a balance line between the bearing housing and the oiler lower bowl. This type of installation, which was shown earlier in Figure 20-5, is required for bearing housings having an excessive back pressure or vacuum and is ideally suited for purge oil

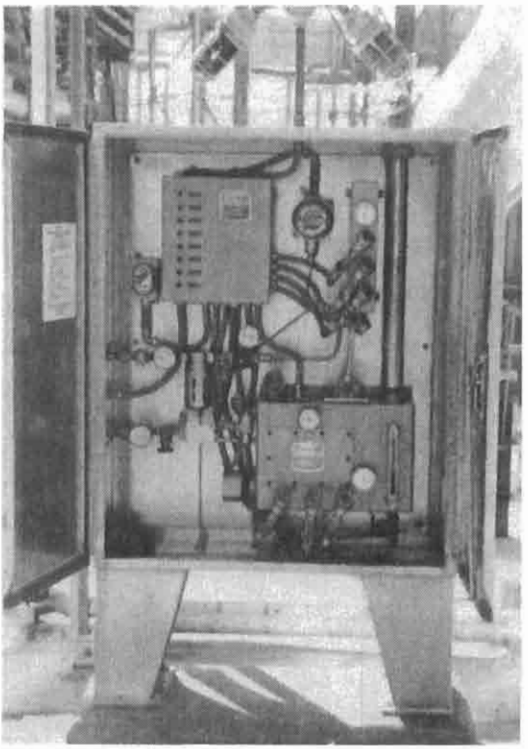


Figure 20-10. Oil mist console suitable for lubricating 30–60 pumps in a refinery (courtesy of Lubrication Systems Company, Houston, Texas).

mist applications. A constant level can be maintained in spite of pressure or vacuum in the housing as the equalizing tube provides static balance of pressure between the bearing housing and oiler bowl.

Rolling element bearings, on the other hand, are ideally suited for dry-sump lubrication. With dry sump oil mist, the need for a lubricating oil sump is eliminated. If the equipment shaft is arranged horizontally, the lower portion of the bearing outer race serves as a mini-oil sump. The bearing is lubricated directly by a continuous supply of fresh oil condensation. Turbulence generated by bearing rotation causes oil particles suspended in the air stream to condense on the rolling elements as the mist passes through the bearings and exits to atmosphere. This technique offers four principal advantages:

- Bearing wear particles are not recycled back through the bearing, but are washed off instead.

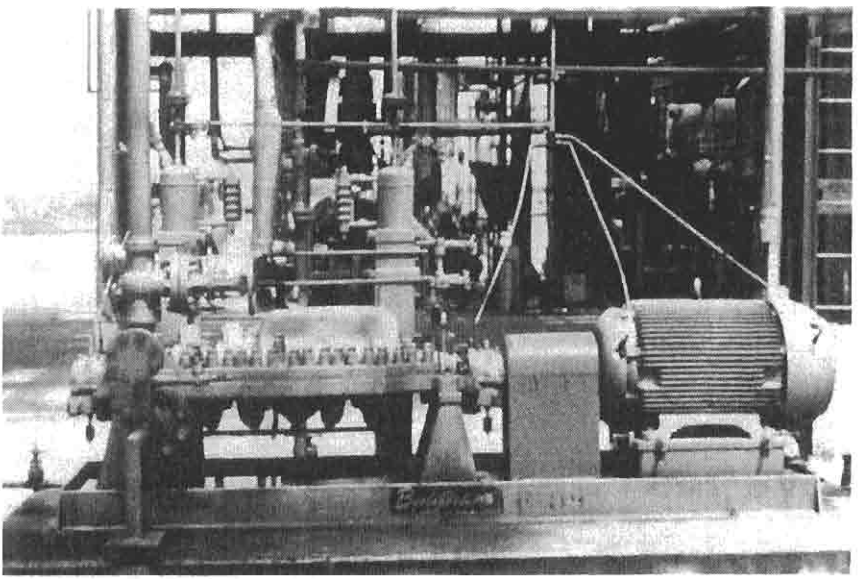


Figure 20-11. Multi-stage pump and electric motor connected to oil mist tubing *(courtesy of Lubrication Systems Company, Houston, Texas).*

- The need for periodic oil changes is eliminated.
- Long-term oil breakdown, oil sludge formation, and oil contamination are no longer factors affecting bearing life.
- The ingress of atmospheric moisture into the motor bearing is no longer possible and even the bearings of standby equipment are properly preserved. Without oil mist application, daily solar heating and nightly cooling cause air in the bearing housing to expand and contract. This allows humid, often dusty air to be drawn into the housing with each thermal cycle. The effect of moisture condensation on rolling element bearings is extremely detrimental and is chiefly responsible for few bearings ever seeing their design life expectancy in a conventionally lubricated environment.

It has been established that loss of mist to a pump or motor is not likely to cause an immediate and catastrophic bearing failure. Tests by various oil mist users have proven that bearings operating within their load and temperature limits can continue to operate without problems for periods in excess of eight hours. Furthermore, experience with properly maintained oil mist systems has demonstrated outstanding service factors. Because there are no moving parts in the basic oil mist components, and

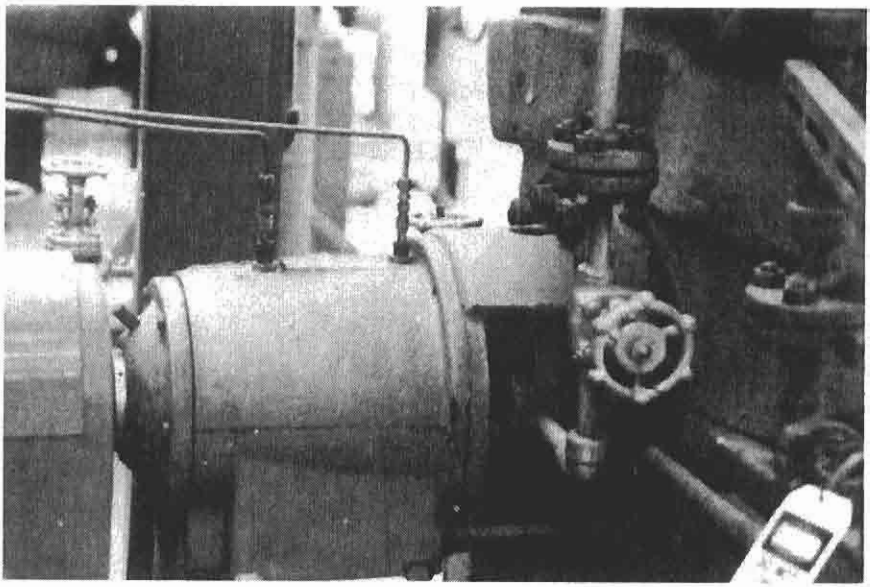


Figure 20-12. Single-stage overhung pump using oil mist lubrication (courtesy of Lubrication Systems Company, Houston, Texas).

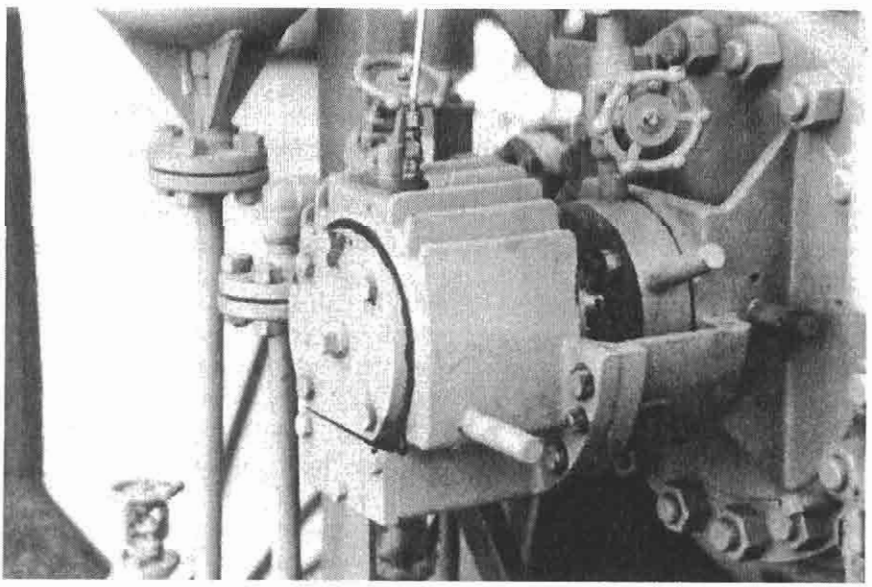


Figure 20-13. Between-bearing pump bracket receiving continuous oil mist lubrication (courtesy of Lubrication Systems Company, Houston, Texas).

because the system pressure is very low, oil mist is a reliable lubrication method. Proper lubrication system operation can be interlocked with pump operation or an alarm system, assuring adequate lubrication.

The savings due to lower preventive maintenance labor requirements, equipment repair cost avoidance, and reductions in unscheduled production outage events have been very significant and cannot possibly be overlooked by a responsible manager or cost-conscious manufacturing facility. Oil mist systems have become extremely reliable and can be used not only to lubricate operating equipment, but to preserve stand-by, or totally deactivated ("mothballed") equipment as well.

The majority of centrifugal pump installations using oil mist lubrication have, as of 1991, applied open, once-through mist flow. However, closed systems with virtually no losses to the environment do exist and should be the preferred configuration in the future [4, 2].

Selecting Rolling Element Bearings for Reduced Failure Risk

Less than 10% of all ball bearings run long enough to succumb to normal fatigue failure, according to the Barden Corporation [13]. Most bearings fail at an early age because of static overload, wear, corrosion, lubricant failure, contamination, or overheating. Skidding (the inability of a rolling element to stay in rolling contact at all times), another frequent cause of bearing problems, can be eliminated by ensuring the bearing will always be loaded. With pairs of angular contact bearings, preloading may be necessary.

Actual operations have shown that better bearing specification practices will avert the majority of static overload problems. Other problems caused by wear, corrosion, and lubricant failure, contamination and overheating can be prevented by the proper selection, application, and preservation of lubricants. Oil viscosity and moisture contamination are primary concerns, and higher-viscosity lubricants are generally preferred [3]. The detrimental effects of moisture contamination are described in Table 20-3.

Unlike API pump bearings, which petrochemical companies often specify for a B-10 life of 40,000 hours, ANSI pump bearings are selected on the basis of an expected 24,000-hour life. Nominally, this means that 90% of the ANSI pump bearings should still be serviceable after approximately three years of continuous operation. However, the failure statistics quoted by MacKenzie [13] indicate that conventionally lubricated ANSI pump bearings do not even approach this longevity. Lack of lubrication, wrong lubricants, water and dirt in the oil, and oil-ring debris in the oil sump all cause bearing life expectancies to be substantially less. It must be assumed that similar findings by other major users of ANSI

Table 20-3
Fatigue Life Reduction of Rolling Element Bearings Due to Water Contamination of Lubricant (Mineral Oil)

Base Oil Description	Water Content of Wet Oil, %	Fatigue-Life Reduction, %	Test Equipment and Hertzian Stress
Mineral oil, dried over sodium	0.002	48	Rolling 4-Ball test, 8.60 GPa (1.25×10^4 psi)
	0.014	54	
	3.0	78	
	6.0	83	

pumps prompted the search for "life-time lubricated" rolling element bearings that we had alluded to earlier, but which nevertheless have their own particular vulnerabilities.

Problem incidents caused by dirt and water have been substantially reduced by oil mist lubrication. However, serious failure risk can also be introduced by certain specification practices, including some contained in API 610. Without going into the many possible factors that could influence bearing life, several items must be considered in the mechanical design of reliable centrifugal pumps. First among these would be that deep-groove Conrad-type radial bearings with loose internal clearance* are more tolerant of off-design pump operation than bearings with precision tolerances. Also, it should be recognized that centrifugal pumps which undergo signficant bearing temperature excursions are prone to cage failures. Phenolic cages are best suited for operation of radial bearings below 225°F (107°C). New cage materials, such as Polyamide 66, provide a much higher temperature limit and excellent lubricity at these temperatures. Metallic cages are, of course, equally suitable and are sometimes preferred in thrust bearings.

The pump designer and pump user must also come to grips with questions relating to preload values and contact angles. The API requirement to utilize duplex 40° contact angle thrust bearings was prompted by the desire to obtain maximum load capacity from a given size bearing [11]. Similarly, the requirement of installing these bearings back-to-back with a light axial preload was aimed at reducing axial shuttling of rotors to prevent brinelling of contact surfaces (raceways) and to prevent ball skidding. Occasionally, an effective way to prevent this destructive skidding is to install matched, preloaded sets of angular contact bearings with *dif-*

* The AFBMA (Anti-Friction Bearing Manufacturers Association) calls this recommended clearance a "C3" clearance as opposed to looser or tighter clearances (C1 or C5, for instance).

ferent contact angles. The MRC Division of SKF markets these bearings under the trade designation "Pumpac." Experience shows these bearings are capable of extending the mean time between pump repairs, especially in single-stage, overhung pumps with unidirectional thrust load, in sizes above 20 hp and at speeds in excess of 1,800 rpm [7].

As regards quantification of preload, Figure 20-14 will prove very enlightening. It shows that for a given bearing (FAG 7314 B.UO, 70 mm bore diameter) a shaft interference fit of 0.0003 in. will produce an al-

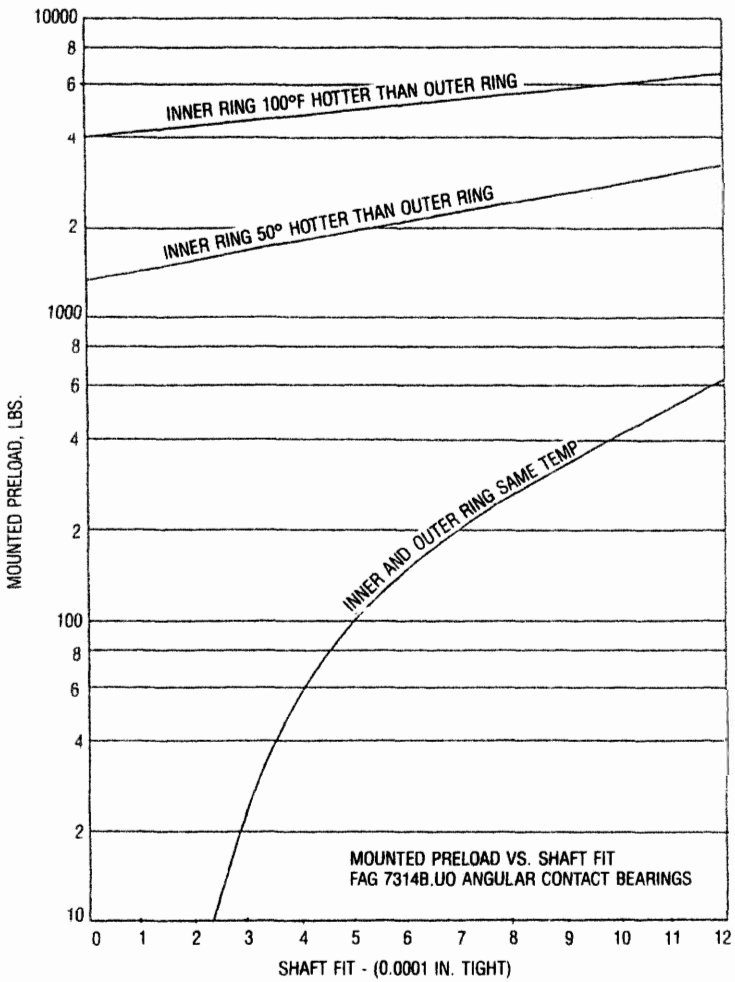


Figure 20-14. Mounted preload vs. shaft fit for a specific 70mm bore diameter bearing (courtesy of FAG Bearing Corporation).

most insignificant preload of approximately 22 lbs, whereas an interference fit of 0.0007 in. would result in a mounted preload of 200 lbs. A much more significant preload would result from temperature differences between inner and outer bearing rings. Such differences could exist in pumps if heat migrated from high temperature pumpage along the shaft or if the pump design incorporated cooling provisions that might artificially cool the outer ring and would thus prevent it from expanding. By far the worst scenario would be for a pump operator to apply a cooling water stream from a firehose. It is sad to see this done today, in the age of high tech, space travel, and information explosion.

Figure 20-15A is very important because it allows us to visualize how the cage inclination of back-to-back mounted angular contact bearings with steeper angles promotes a centrifugal outward-oriented flinging action from side "A" to side "B" [8]. If conventionally lubricated angular contact ball bearings are back-to-back mounted as shown in Figure 20-15B, lubricant flow may become marginal or insufficient. Subject to proper selection and utilization of proper installation procedures, face-to-face mounting (Figure 20-15C) may be advantageous because it promotes through-flow of lube oil. This presupposes that the temperature difference between inner and outer race is minimal. If the temperature difference were substantial, growth of the inner ring would force the bearing into a condition of high axial preload.

The least vulnerable thrust bearing execution is illustrated in Figure 20-16. Here, the vendor has opted to guide the lube oil into spacer ring "A." This ensures that lubricant flows through both bearings before exiting at each end. A second spacer ring "B" facilitates making preload adjustments. Flinger disc "C" tosses lube oil onto the surrounding surfaces and from there it flows into trough "D" and on towards both inboard and outboard bearing locations. The periphery of flinger "C" dips into the lube oil level; however, the lube oil level is generally maintained well below the center of the lowermost ball. This reduces oil churning and friction-induced heat-up of lube oil and bearings. Needless to say, unless lubricant application methods take into account all of the above, bearing life and reliability may be severely impaired.

The pump designer and user should also realize that double row "filler notch" bearings are considerably more vulnerable in pump thrust applications than other bearing types and should not be used. Similarly, ball bearings are sensitive to misalignment and must be properly mounted to eliminate this cause of failure. Misalignment must be no greater than 0.001 inch per inch of shaft length. Bearings operating in a misaligned condition are subject to failure regardless of cage type, although riveted cages seem particularly prone to rivet head fatigue in misaligned condition.

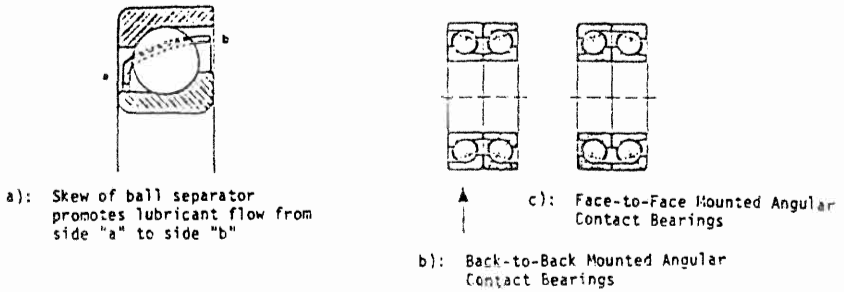


Figure 20-15. Angular contact bearings commonly used for thrust take-up in centrifugal pumps.
a) Skew of ball separator promotes lubricant flow from side "a" to side "b."
b) Back-to-back mounted angular contact bearings.
c) Face-to-face mounted angular contact bearings.

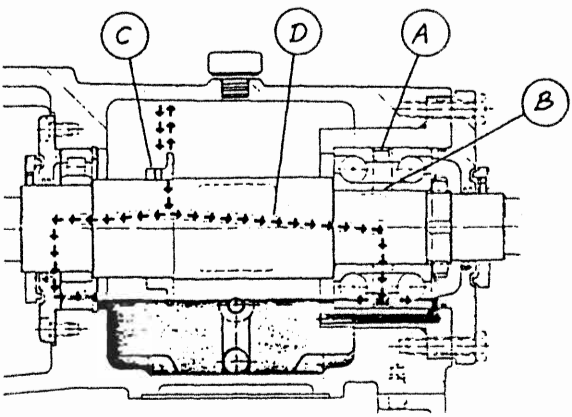


Figure 20-16. Advantageous lubrication arrangement ensures optimized delivery of lube oil to all bearings (courtesy of KSB Pump Company, Kaiserslautern, Germany).

Magnetic Shaft Seals in the Lubrication Environment

Most pump shaft seals are generally inadequate. ANSI pumps are usually furnished with elastomeric lip seals. When these seals are in good condition, they contact the shaft and contribute to friction drag and temperature rise in the bearing area. After 2,000 to 3,000 operating hours, they are generally worn to the point at which they no longer present an effective barrier against contaminant intrusion. Percent failure vs. hours to leakage for two types of lip seals is shown in Figure 20-17 [10].

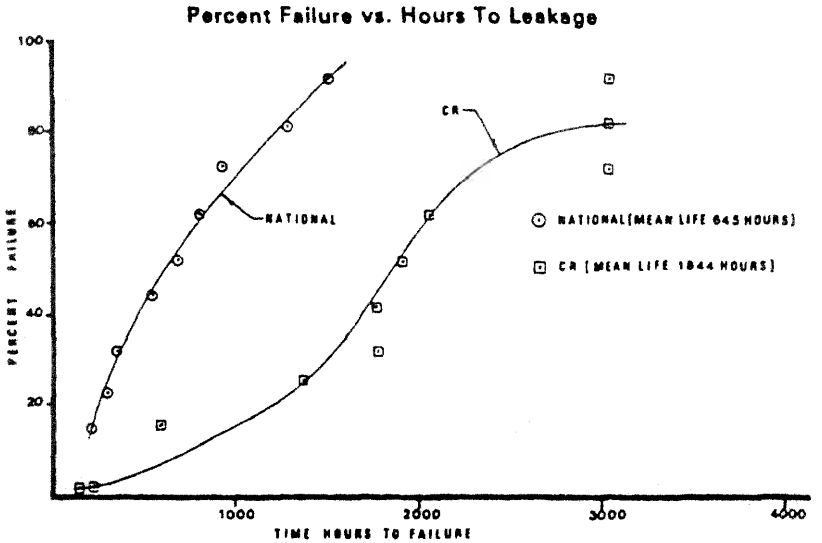


Figure 20-17. Percent failure vs. hours to leakage for two types of lip seals (courtesy of L. A. Horve, "CR Waveseal—A Detailed Synopsis of Five Comparative Life Tests").

API-type pumps are generally furnished with non-contacting labyrinth seals or bearing isolators similar to the one depicted in Figure 20-18. However, neither of these two executions can claim to represent a totally effective barrier against the intrusion of atmospheric dust or moisture. Moisture, in particular, can cause unexpectedly high decreases in bearing fatigue life. Armstrong and Murphy [1] summarize the deleterious effects of water in lube oil. They state that in the 1960s, researchers Grunberg and Scott had found the fatigue life at a water content of 0.002% reduced 48% and, at 6.0% water, found it to be reduced 83%. Others found a fatigue life reduction of 32% to 43% for squalene containing 0.01% water, and a third group of researchers detected about an 80% drop with a moist air environment contacting dried mineral oil (see Table 20-3). While the detailed mechanism for the reduction of fatigue life by water in a lubricant is not completely understood, it is thought to relate to aqueous corrosion. There is much evidence that the water breaks down and liberates atomic hydrogen. This causes hydrogen embrittlement and increases the rate of cracking of the bearing material by a significant margin.

A solution to the problem of sealing pump-bearing housings can be found in aircraft and aerospace hydraulic pumps, which make extensive use of magnetic face seals. This simple seal consists of two basic compo-

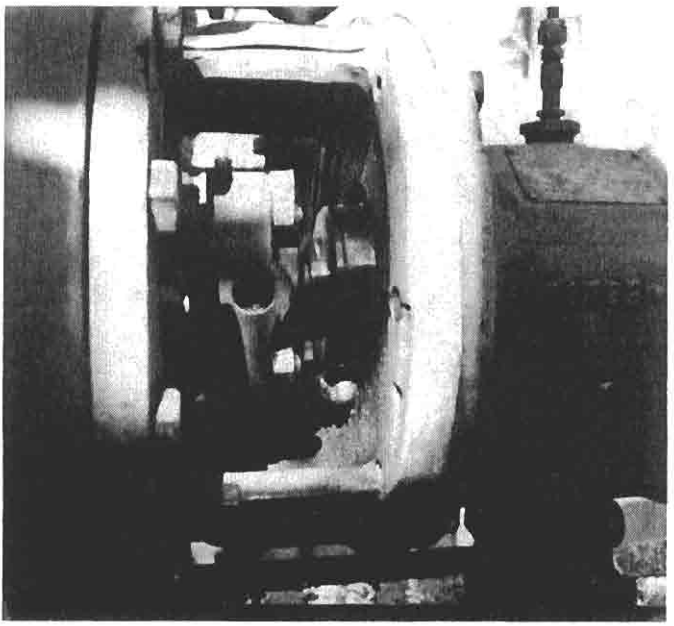


Figure 20-18. "Bearing isolators" are non-contacting bearing housing seals which are sometimes used in centrifugal pumps (courtesy of INPRO Seal Company, Rock Island, Illinois).

nents as shown in Figure 20-19. These components are: (1) a magnetized ring, having an optically flat sealing surface that is fixed in a stationary manner to the housing and sealed to the housing by means of a secondary O-ring, and (2) a rotating ring having a sealing surface that is coupled to the shaft for rotation and sealed to the shaft with an O-ring.

The rotating ring, which is fabricated from a ferromagnetic stainless steel, can be moved along the shaft. When no fluid pressure exists, the sealing surfaces are held together by the magnetic force, which is reliable and uniform, creating a positive seal with minimum friction between the sealing faces and ensuring the proper alignment of the surfaces through the equal distribution of pressure.

Magnetic seals are very reliable. Many have operated continuously for 40,000 hours without repair or adjustment under conditions considerably more severe than those to which chemical process pumps are typically exposed.

Finally, if sealed-bearing housings must accommodate changes in internal pressure and vapor volume, an expansion chamber similar to that shown in Figure 20-20 can be used. This small device, which incorpo-

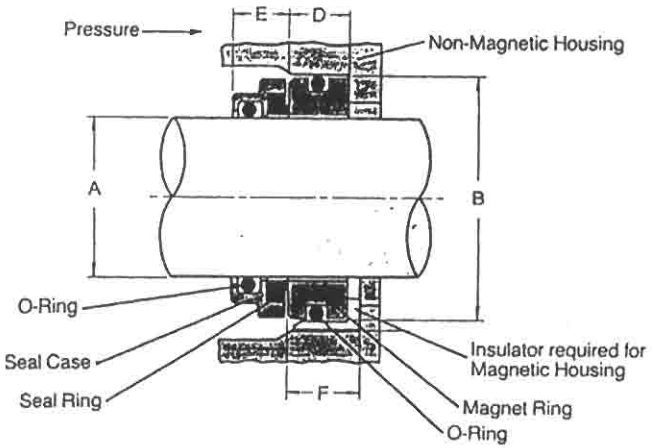


Figure 20-19. Magnetic face seals find extensive application in aerospace pumping services (courtesy of Magnetic Seal Corporation, West Barrington, R.I.).

rates an elastomeric diaphragm and constitutes a completely enclosed system, is screwed into the housing vent opening. It accommodates the expansion and contraction of vapors in the bearing housing without permitting moisture and other contaminants to enter. Carefully selected from a variety of plain and fabric-supported elastomers, the diaphragm will not fail prematurely in harsh chemical environments [14].

The upgraded medium-duty bearing housing shown in Figure 20-21 incorporates the various bearing life improvement features that have been discussed: (1) a deep-groove Conrad-type bearing with loose internal clearance (C3); (2) a duplex angular contact (40°), lightly preloaded back-to-back-mounted thrust bearing; (3) a vent port that remains plugged for dry-sump oil-mist lubricated bearings and that can be fitted with an expansion chamber if conventionally lubricated; (4) a magnetic seal; and (5) a bearing housing end cover made to serve as a directed oil-mist fitting [6].

If oil mist is not available at a given location, the conventional lubrication method shown earlier in Figure 20-16 would be the second choice for engineers experienced with pump operation and maintenance. Third choice would be as indicated in Figure 20-22, with the oil level at "A" or reaching to the center of the lowermost ball. The oil ring, Item 1, serves only to keep the oil in motion. If the oil level is allowed to drop to point "B," the oil ring is expected to either feed oil into the bearings or generate enough spray to somehow get adequate amounts of lubricant to the various bearings. Unfortunately, there is no real assurance of this happening. An oil ring with a given geometry will be fully effective only if it

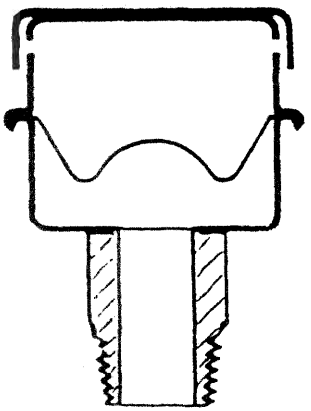


Figure 20-20. A small expansion chamber accommodates vapor expansion and contraction in hermetically sealed bearing housings (courtesy of Gits Manufacturing Company, Bedford Park, Illinois).

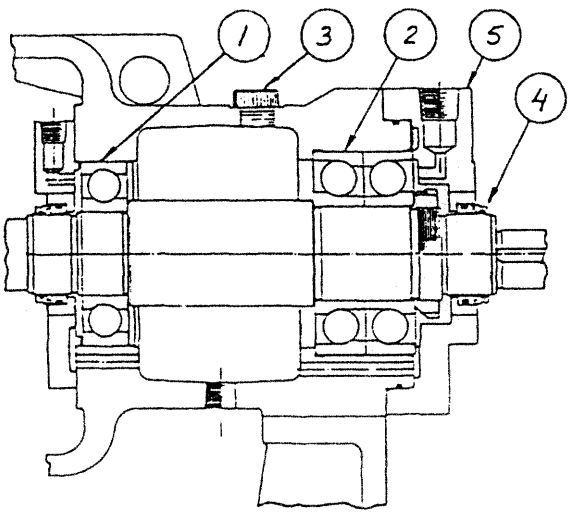


Figure 20-21. Bearing housing for so-called upgraded medium duty (UMD) pump marketed in the U.S. since 1985 (courtesy of Carver Pump Company, Muscatine, Iowa).

is immersed to the proper working depth, turns at the proper speed, and is surrounded by lube oil of the proper viscosity. This is graphically illustrated in Figure 20-23, which is derived from Heshmat and Pinkus [9]. In this context, *fully effective* would mean that the oil ring should deliver a sufficient amount of oil even as the shaft speed increases. This is achieved by the ring provided with grooves on the inside diameter, whereas the ring with the conventionally executed inside diameter appears to deliver sufficient amount only at low shaft speeds.

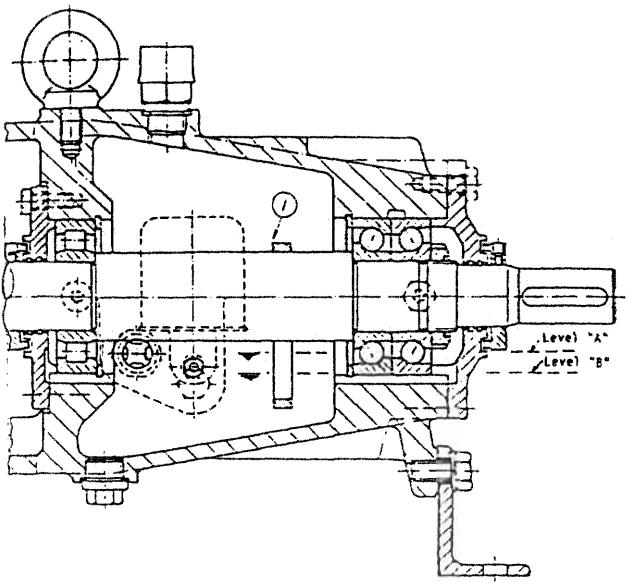


Figure 20-22. Conventionally lubricated centrifugal pumps using oil rings must pay close attention to oil levels (courtesy of Sulzer-Weise, Bruchsal, Germany).

In summary, proper lubrication of rolling element bearings in centrifugal pumps depends on such factors as bearing preload, cage inclination, point of introduction of lube oil, and oil ring design, immersion, oil vis-

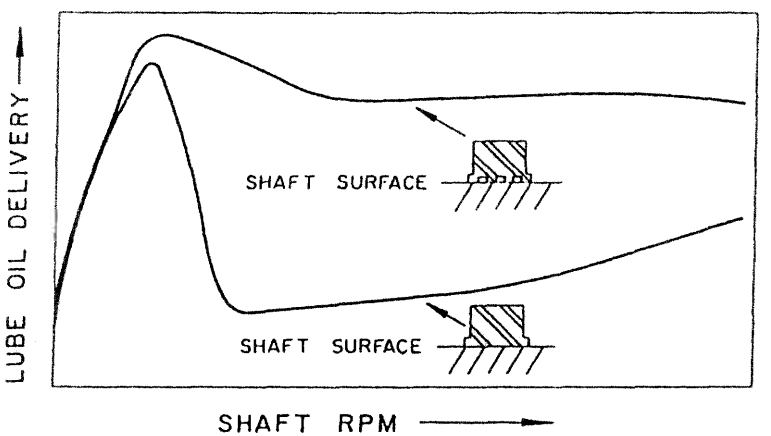
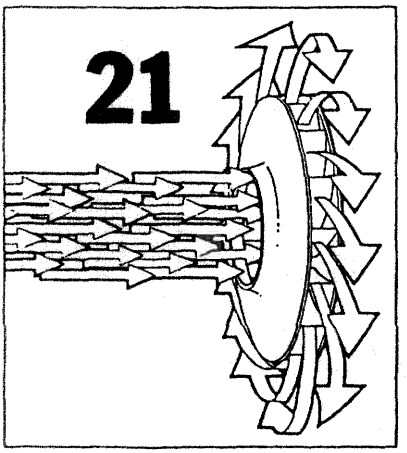


Figure 20-23. Amount of oil delivered by grooved and non-grooved rings (courtesy of H. Heshmat and O. Pinkus, "Experimental Study of Stable High-Speed Oil Rings").

cosity, and oil cleanliness. Unless these factors are understood and unless both user and designer take them into full account, bearing life may be erratic or perhaps even consistently too low.

References

1. Armstrong, E. L., Murphy, W. R., and Wooding, P. S., "Evaluation of Water-Accelerated Bearing Fatigue in Oil-Lubricated Ball Bearings," *Lubrication Engineering.* Vol. 34, No. 1, pp. 15–21.
2. Bloch, H. P., "Executing Oil Mist Lubrication in the 90s," *Hydrocarbon Processing,* October 1990, p. 25.
3. _____, *Improving Machinery Reliability,* Second Edition, Gulf Publishing Company, Houston 1988.
4. _____, *Oil Mist Lubrication Handbook,* Gulf Publishing Company, Houston, 1987.
5. Bloch, H. P. and Geitner, F. K., *Machinery Component Maintenance and Repair,* Second Edition, Gulf Publishing Company, Houston, 1990.
6. Bloch, H. P. and Johnson, D. A., "Downtime Prompts Upgrading of Centrifugal Pumps," *Chemical Engineering,* November 25, 1985.
7. Elliott, H. G. and Rice, D. W.; "Development and Experience with a New Thrust Bearing System for Centrifugal Pumps," Proceedings of 4th International Pump Users Symposium, Texas A&M University, 1987.
8. Eschmann, Hasbargen, Weigand, *Ball and Roller Bearings,* John Wiley & Sons, New York, 1983.
9. Heshmat, H. and Pinkus, O., "Experimental Study of Stable High-Speed Oil Rings," Transactions Of The ASME, Journal of Tribology, January 1985.
10. Horve, L. A., "CR Waveseal—A Detailed Synopsis of Five Comparative Life Tests," CR Industries, Elgin, Illinois, 1977.
11. James, R., Jr., "Pump Maintenance," *Chemical Engineering Progress,* February 1976, pp. 35–40.
12. "Lubrication of Anti-Friction Bearings," Form 446-1, TRW Bearings Division, Jamestown, New York 14701, January 1983.
13. MacKenzie, K. D., "Why Ball Bearings Fail," Product Data Bulletin No. 5, The Barden Corporation, Danbury, Connecticut.
14. Nagler, B., "Breathing: Dangerous to Gear Case Health," *Power Transmission,* January 1981.
15. Shelton, W. A., "Lubed-For-Life Bearings on Centrifugal Pumps," *Lubricating Engineering,* June 1977.



Mechanical Seal Reliability

by **Gordon S. Buck**
John Crane Inc.

In spite of recent advancements in mechanical sealing technology, excessive leakage is the most common cause of "pump repairs." In fact, seal-related repairs represent approximately 60% to 70% of all centrifugal pump maintenance work. Therefore, seal-related repairs are an excellent area to concentrate pump reliability improvement programs. Unfortunately, because of the many facets of seal reliability, this chapter can only serve as an overview and introduction to mechanical seal reliability.

Although a mechanical seal may be small enough to hold in your hand and simple in concept (see Chapter 17, Figure 17-1), it is actually a complex device. The successful operation of mechanical seals in centrifugal pumps calls for careful attention to detail in several areas. These areas are:

- Seal hardware (including sleeve, gland, and gaskets).
- Seal installation.
- Pump hardware (including piping).
- Pump repair and installation.
- Pump operation (including flush to seal).

Of course, these areas are not independent, but often seal reliability can be improved by concentrating on individual areas to solve particular problems. Naturally, the pertinent area must be selected or the effort may be entirely wasted. Therefore, the first step in reducing seal failures is to establish a seal failure analysis program.

Failure Analysis

The purpose of a mechanical seal is to prevent, or at least limit, leakage. Normally, the leakage from a mechanical seal is so slight that it is difficult to measure. For example, a 2-inch seal containing water at 100 psig should leak less than 2 ml/hr. Current federal regulations require that the concentration of certain volatile organics be less than 10,000 ppm near the seal leakage point. In some areas, this limit is 1000 ppm. It is likely that none of these examples would show any visible leakage. Because most seals are wearing devices, they will fail, that is, leak *excessively,* at some point in time. The machinery engineer must therefore be prepared to learn from each seal failure and use that information to improve future designs and applications so that seal life is improved. In order to do this, detailed records of each failure should be maintained.

Failure analysis is simply allowing the seal to tell why it did not perform as expected. This means acting the part of a detective. The following simple guidelines will make failure analysis easier and more consistent:

- Know what a new seal looks like.
- Know what a successfully applied seal looks like.
- Examine the failed seal carefully.
- Write down the differences between the new, successful, and failed seals.
- Formulate a consistent explanation about the differences.

Although manufacturers and mechanics are familiar with new seals, sometimes a machinery engineer may not see the unused item. He only sees the failures. Similarly, he may not see many successfully applied seals because they are not called to his attention. He is too busy looking at failures. On the other hand, some failure modes are so common that a mechanic may come to accept them as "normal wear." For failure analysis to be useful, all differences in the new and failed seal should be noted.

These differences must be written down in a consistent format. The form shown as Table 21-1 is a good starting point and can be modified to suit particular needs. There is a great temptation to take mental notes and write them later. This temptation must be overcome, and written notes must be made while examining the seal. If these notes become soiled, they can be rewritten later.

For many refineries and chemical plants, the statistics in Table 21-2 have been developed from analysis of many pump seal failures.

Statistics show that most seals fail prematurely, that is, prior to wearing out. They also show that these early failures can be prevented by sim-

Table 21-1
Mechanical Seal Failure Analysis Checklist

Machine Identification _____
Reason for Service Request _____
Last Known Failure _____

Summary of Analysis _____

_____

Seal Identification

Manufacturer _____ Type _____
Face Materials _____
Gasket Materials _____
Mounting _____ Balance Ratio _____

Operating Conditions

Liquid _____ Specific Gravity _____
Temperature, F _____ Sealed Pressure, psig _____
Seal RPM _____

Inspection of Seal

Stationary Face	(OK)	Rotating Face	(OK)
Work out	( )	Worn out	( )
Grooved	( )	Grooved	( )
Heat checked	( )	Heat checked	( )
Warped	( )	Warped	( )
Corroded	( )	Corroded	( )
Gasket, Stationary Face	(OK)	Gasket, Rotating Face	(OK)
Hardened	( )	Hardened	( )
Chemical attack	( )	Chemical attack	( )
Broken/cut	( )	Broken/cut	( )
Gasket, Gland	(OK)	Gasket, Sleeve	(OK)
Hardened	( )	Hardened	( )
Chemical attack	( )	Chemical attack	( )
Broken/cut	( )	Broken/cut	( )
Gland	(OK)	Drive Collar	(OK)
Corroded	( )	Corroded	( )
Bushing rubbed	( )	Turned on shaft	( )
Number of bolts	____	Setscrews	____
Antirotation Pins	(OK)	Sleeve/Shaft	(OK)
Fretted	( )	Fretted	( )
Corroded	( )	Corroded	( )
Flush Plan _____		Springs/Bellows	(OK)
Location _____		Broken	( )
Hole Size _____		Corroded	( )

Table 21-2
Causes of Seal Failures

Operating problems	30% to 40%
Improper installation	20% to 30%
Pump design/repair	10% to 20%
Misapplied seal	10% to 20%
Worn out	1% to 5%
Miscellaneous	15% to 20%
Total	100%

ply selecting the proper seal, installing it correctly, and operating the pump carefully.

Seal Hardware Failures

Another way to look at seal reliability is to insist that the seal must survive in spite of any problems with the pump or its operation. In this case, failures are examined from a hardware or design point of view. In other words, if a pump requires repair because the seal is leaking, the corrective measures are directed towards the seal hardware. This may not always be correct; however, the statistics presented in Table 21-3 seem to apply to hardware failures in refinery and chemical plant pump seals. Naturally, seals or pumps that were not properly installed or repaired are omitted from Table 21-3.

Table 21-3
Mechanical Seal Hardware Failures

Rotating or stationary face	40% to 70%
Loss of axial movement (hangup)	10% to 20%
Fretting	10% to 20%
Corrosion	10% to 20%
Miscellaneous	5% to 20%
Total	100%

Seal Faces. Table 21-3 shows that, in pump repairs caused by hardware failures, the seal faces are the apparent problem 40% to 70% of the time. These faces were severely worn or damaged. In some instances, an immediate improvement was made by simply changing face materials. For example, a stellite or ceramic face might be replaced with a solid tungsten carbide face. In other instances, the face materials could not be changed or improved and efforts were focused on other areas.

Because mechanical seal failures can be expensive, there is no excuse for using second rate materials. In many refinery and chemical plant services, premium grade carbon and tungsten carbide or silicon carbide faces have proven reliable. Springs are frequently Alloy 20. Glands, sleeve, shells, adapters, etc. are made from 316SS. In order to reduce inventories and avoid costly mix-ups, these same materials are also used in easy services—even cold water.

Hangup. Loss of axial movement, or seal hangup, is caused when leakage accumulates and hardens beneath the flexible element (see Figure 21-1). In most seals, this is the rotating element. Hangup is a particularly troublesome problem in high temperature seals because any leakage tends to form "coke." There are four approaches to solving hangup problems:

- Change the liquid (external flush that will not solidify).
- Reduce leakage (higher balance ratio and/or narrow face seal).
- Remove leakage (steam quench).
- Remove opportunity to hangup (change to a stationary seal).

Some stationary seals are designed so that there is a relative motion between the "floating" element and the rotating shaft. This relative motion makes it difficult for solids to accumulate and cause hangup. Because stationary metal bellows seals include this feature as well as narrow faces and are rated for high temperatures, they are often used in services that tend to cause hangup of conventional seals.

Fretting. Fretting is the damage caused by the constant rubbing of the flexible element against its drive lugs or the dynamic gasket against the sleeve. The movement may be less than a thousandth of an inch, but eventually the sleeve or drive lug is damaged. The damage may cause leakage directly or may restrict axial movement so that leakage occurs between the faces.

The solution to fretting problems usually involves one of the following approaches:

- Improve seal alignment during installation.
- Reduce shaft axial float (adjust bearings).
- Use O-rings for gaskets.
- Change to a bellows seal.

Corrosion. Most mechanical seals are made of relatively corrosion-resistant materials. For example, a typical off-the-shelf seal might be made of 316 stainless steel, carbon graphite and tungsten carbide with fluoroelastomeric gaskets. Metal parts are also available in Alloy 20, Hastelloy, and other corrosion resistant materials. Carbon graphites are available

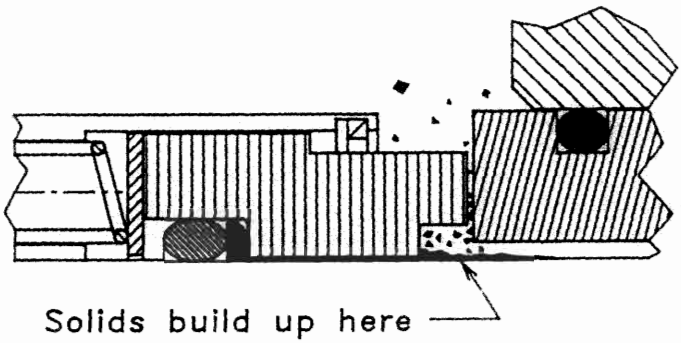


Figure 21-1. Seal hangup from accumulated solids (courtesy John Crane Inc.).

with many different types of resin and metallic binders. Hardfaces include both nickel and cobalt bound tungsten carbide, various silicon carbides and other ceramics. Gaskets may be made from many materials including graphite foil, Teflon, nitrile, Kalrez, etc.

It is most likely that a suitable set of materials may be found for even the most corrosive seal environment. Still, while searching for those materials, corrosion problems do occur. Even after the proper materials have been found, the wrong material may be inadvertently substituted at the next repair.

Other Problems. There are other hardware problems that occur in addition to face damage, hangup, fretting, and corrosion. The most perplexing problem occurs when there is no obvious damage. This often happens when sealing light hydrocarbons. For these products, the liquid may flash to a vapor between the seal faces. Since most seals are designed for liquids, the resulting force imbalance from flashing causes the faces to "pop open." The solution may involve changing the balance ratio or seal type or installing an external flush that will not vaporize so readily.

Seal Failures from Installation Problems

Proper installation is so important that all seal manufacturers conduct extensive training courses on how to install their seals correctly. Most of the larger seal users have their own internal training courses as well. Consultants are available to teach proper seal installation. Still, the statistics show that 20% to 30% of all seal failures are caused by a poor or incorrect installation. Many installation problems seem to stem from improper assembly of seal components and internal misalignment.

Improper Assembly. A large refinery may use several different seal types from four or five manufacturers. Some plants do not have techni-

cians who specialize in rotating equipment repair. Many plants use independent repair shops for pump repairs. All of these factors make it increasingly difficult to assure that the separate seal components are assembled properly. For these reasons, more and more users are turning to cartridge seals.

A cartridge seal is a complete seal assembly, including the gland, that is mounted on a sleeve (see Chapter 17, Figure 17-26). In addition to being assembled, the seal is also adjusted to the correct operating position. The user simply slides the cartridge assembly over the shaft and fastens it in place. This approach is much less prone to assembly and installation errors.

In the larger plants, used cartridge seals may be reconditioned by a specialist and stocked for future installation. Smaller plants may rely on the seal manufacturer to both recondition and stock the cartridge units.

Internal Misalignment. Most mechanical seal designs incorporate one flexible element and one fixed element. The most common arrangement has the flexible element rotating and the fixed element stationary. Seal life is greatly improved by assuring that the fixed element is mounted perpendicular to the rotating shaft. Because the fixed element is contained in the gland and the gland is bolted to the stuffing box, several surfaces must be checked for perpendicularity. Furthermore, the shaft is mounted in the bearing housing that is bolted to the pump case. These mounting surfaces must also be checked.

For overhung process pumps, a precision dial indicator can be used to assure that internal misalignment is minimized. The bearing bracket should be assembled as a complete unit so that the following checks can be made:

- Shaft run-out < 2 mil TIR.
- Shaft lift < 3 mil.
- Shaft float < 5 mil (this author recommends < 1 mil).
- Bracket face square to shaft within 3 mil.
- Sleeve (mounted on shaft) run-out < 2 mil.

In addition, the stuffing box face must be checked to ensure that it is perpendicular to the shaft within 3 mils TIR. The most reassuring method of checking this is to measure it directly. On some pumps, this may be done by mounting the assembled bearing bracket to the pump without including the gland. A dial indicator that reads the stuffing box face is placed on the shaft. If there is not enough room to do this, an indirect method must be used. On these pumps, the cover should be checked to see that the face of the stuffing box and the mounting face for the bearing bracket are parallel within 3 mils TIR.

Seal Failures Related to Pump Hardware

Sometimes seal failures are so strongly related to the pump and its performance that reliability can be improved only by modifying the pump. The pump may have a true design problem or may simply have been misapplied. Some of the more common pump hardware related seal failures are:

- Excessive shaft deflection.
- Narrow annulus of stuffing box.
- Shaft axial movement.
- Excessive piping loads.

Excessive Shaft Deflection. If the shaft deflects, the seal must move axially each revolution to compensate. API Standard 610 for centrifugal pumps specifies a maximum of 0.002-inch shaft deflection at the location of the seal faces. Some older pumps and non-API pumps may not meet this specification. In particular, older pumps designed for packing may have excessive shaft deflection. Shaft deflection is reduced by increasing the shaft diameter and/or reducing the bearing span or shaft overhang.

Narrow Annulus. In spite of the fact that the vast majority of centrifugal pumps are equipped with mechanical seals, most older pumps are actually designed for packing. This is particularly true for heavy duty API type pumps manufactured prior to the seventh edition of API Standard 610.

A pump designed for packing has a cylindrical "stuffing box" through which the shaft passes. Packing is compressed within the annulus between the OD of the shaft and the ID of the stuffing box. This annulus may be from 3/8 inch to 1 inch in cross section. If the pump is actually equipped with a seal, the seal must fit inside this annulus. The result is that very little liquid surrounds the seal.

Seal manufacturers know that their products are more reliable when operated within a "seal chamber" that contains more liquid than a stuffing box. However, many pump users insist on the ability to convert from seals to packing. Also, pump manufacturers prefer to use existing designs and patterns that are interchangeable with older models. The result is that most pumps are designed for packing but use seals. It should be no wonder that the seal is the most common cause of a pump repair.

In actual laboratory tests, seal face temperatures have been observed to decrease when the stuffing box bore is increased. For example, a stan-

dard arrangement had about 1/16-inch radial clearance between the seal and the stuffing box bore. When this clearance was increased to 1/2 inch, the seal face temperature decreased by 38°F. In response to this sort of information and experience, many pump manufacturers are now supplying "oversized" stuffing boxes for their pumps.

Shaft Axial Movement. Whenever the shaft moves axially, one of the sealing elements must move in the opposite direction to maintain the existing gap between the faces. If this movement occurs repeatedly, fretting may occur between the seal and shaft/sleeve. Table 21-3 shows that fretting is a significant cause of seal failures.

For reasons that are not completely clear, many manufacturers assemble their pumps so that the shaft can "float" axially for 3 mils to 8 mils. This is a built-in opportunity for the seal to fret. To reduce this potential, many pump users go against the manufacturer's recommendation and modify the pumps to restrict "float" to less than 1 mil.

Excessive Piping Loads. A centrifugal pump is not a good pipe anchor. API 610 specifies maximum loads that must be used by both pump manufacturer and piping designer. Normally this is not a problem except for end suction pumps used in high temperature services.

Beginning with the sixth edition, API 610 specifies piping loads that are considerably higher than previous editions. While this makes the job of the piping designer easier, it has imposed a severe constraint on the pump manufacturer. Most pumps that were designed to meet the early editions of API 610 will not meet the more stringent requirements of the sixth and seventh editions. Because they are not as rigid, older pumps may require even more attention to installation and hot alignment than the newer models.

Seal Failures Caused by Pump Repair and Installation

The mechanical seal is sensitive to the environment in which it operates. In particular, the mechanical condition of the pump has a significant effect on seal reliability. Some of the major considerations include:

- Rotating imbalance.
- Alignment to driver.
- Flushing the seal.

Rotating Imbalance. Vibration reduces seal life and any rotating imbalance will create vibration. The frequency of this vibration will be the

running speed of the pump. The best way to prevent this is to dynamically balance the rotating assembly. Although many technicians will work with an assembly until no further improvement can be detected, a good commercial standard is 0.0045 ounce-inch per pound of assembly.

Alignment to Driver. Misalignment between the pump and driver is another cause of vibration. This vibration occurs at two times the running speed of the pump. Both radial and axial components may be present. In order to reduce this vibration, many pump users take great pains to align pumps to drivers.

Flushing the Seal. Because a mechanical seal is a rubbing device that generates heat, it must be lubricated and cooled. Cooling and lubrication are normally accomplished by circulating a suitable liquid around the seal. This is called flushing the seal. The most common arrangement is the discharge bypass flush, sometimes called API Plan 11 (see Chapter 17, Figure 17-9), but there are many others (see Chapter 17, Figure 17-26).

Perhaps the most important feature of a good flushing system is that the liquid circulates around the rubbing interface of the seal. This is accomplished by injecting or withdrawing the liquid in close proximity to the faces.

In general, seal flush systems should be as simple as possible to avoid installation and operating errors. This means designing the system to minimize valves, orifices, filters, etc. A common problem with flushing systems is that liquid does not actually flow because a valve is closed, a filter is blocked, etc.

Seal Failures Caused by Pump Operation

Although mechanical seals should be designed to survive operational upsets, seal life can certainly be enhanced by observing a few simple precautions. Some of the more common operating problems include:

- Operation at low flow.
- Operation at no flow.
- Cavitation.

Low Flow. In spite of the performance curve, which shows operation from no flow to maximum, centrifugal pumps may not operate smoothly

at less than about 50% of Best Efficiency Point (BEP) flow. Vibration and noise may increase markedly at less than this minimum stable flow. The result is a decrease in seal life. Resolution of this problem may require hydraulic modification by the pump manufacturer or a by-pass line to artifically increase flowrate.

At still lower flows (less than 20% of BEP), the liquid passing through the pump may be significantly raised in temperature. According to API 610, the minimum continuous thermal flow is "the lowest flow at which the pump can operate and still maintain the pumped liquid temperature below that at which net positive suction head available equals net positive suction head required."

No Flow. In spite of assurances to the contrary by the operating crew, sometimes pumps have obviously been running "dry." There are many ways this can happen—a broken level controller, a plugged suction strainer, a broken check valve or a discharge valve that was never opened. All of these and more can prevent liquid from entering or leaving the pump.

Some batch processes actually are designed so that the pump runs dry as it pumps out a reactor or supply vessel. Dissimilar pumps operating in parallel are a classic example of how one pump may force another to run at no flow.

In addition to the noise and vibration problems described for low flow, operation with no flow may generate enough heat to vaporize any liquid trapped in the pump. In particular, seals that are flushed from the pump discharge or suction may quickly become "gassed up." The obvious solution is to *avoid* the no flow situation. If the pump must run under these conditions, a separate flushing system may be required for the seal.

Cavitation. Cavitation has been rightly and wrongly blamed for many ills in both pumps and seals. Certainly cavitation increases pump vibration and vibration reduces seal life. Recent studies have shown that the simple 3% head loss rule that is used to define NPSHR may not adequately define the onset of cavitation problems. Also, operation at low flow sometimes produces symptoms similar to cavitation.

Cavitation problems are sometimes difficult to solve. Frequently, increasing the available NPSH is prohibitively expensive. Reducing the required NPSH is sometimes possible with special impellers or inducers. For these reasons, cavitation problems must be carefully addressed during the initial specification of the pumps.

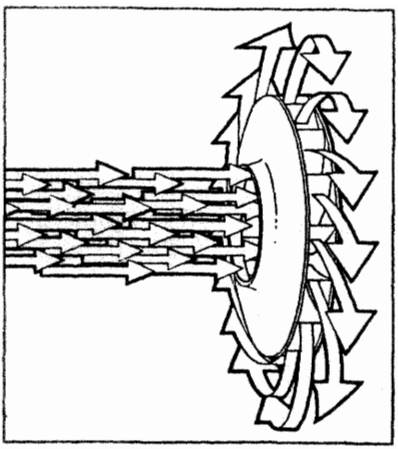
Reliability

Once the potential problems of selection, installation, and operation are addressed, mechanical seals can provide reliable sealing for many years. Many large refineries and chemical plants have obtained a gross average mean time between pump failures that is greater than two years. (This statistic includes all causes of pump repairs and not just mechanical seals.)

Some services will always be more difficult than others and have a correspondingly lower reliability. Higher pressures, higher temperatures, higher solids content, and increased corrosion rates will generally result in a lower pump and seal reliability. Similarly, large seals and high shaft speeds may have lower reliability. Rather than accept lower reliability, many users elect to use special purpose seals and sealing systems that are engineered for these specific applications.

Considering the wide variety of designs, arrangements, and materials available, it is no wonder that the mechanical seal is the most popular device for sealing modern centrifugal pumps.

This page intentionally left blank



Index

A

Affinity law, 12–14
Alignment, 497–523
  baseplate, 501–502
  bracket sag, 510
  calculating movements, 516–517
  causes of misalignment, 498–502
  conditions of misalignment, 497–498
  couplings, 506–507
  foundation, 500–501
  grouting, 502, 505–506
  horizontal movement methods, 504–505
  jig posts, 516, 518–519
  measurement methods, 511–515
  pre-alignment steps, 507–511
  soft foot, 508–509
  thermal growth, 519–522
  vertical movement methods, 503
Axial thrust, 343–352

B

Balance disk, 208
Balance drum, 208
Balance dynamic, 223–224
Barkse-type pump, 175–179
Baseplates, 218, 304, 501–502
Bearings. *See also* Bearings and lubrication.
  chemical pumps, 299–303
  double case pumps, 216, 218
  high speed pumps, 202–203
  slurry pumps, 237
  stiffness and damping, 445–448

Bearings and lubrication, 524–555. *See also* Bearings.
additives, 529
bearing failure, 545–548
bearing life, 545–548
bearing temperature, 530–531
drip feed, 539–540
emulsification, 528
fire point, 527–528
flash point, 527–528
forced feed, 539–540
friction torque, 525
function of the lubricant, 525
grease limitations, 533–534
liquid lubricant applications, 536–537
lubricant selection, 529–533, 536
magnetic shaft seals, 549–550
oil bath, 537–538
oil characteristics, 526–529
oil mist, 539, 541–545
oil replacement frequency, 528–529, 534
oil ring, 552–555
oil versus grease, 525–526
oxidation, 528
pour point, 527
rust prevention, 528–529
sealed bearings, 534–536
shaft seal types, 549–551
viscosity, 526–527, 536
Bi-rotors, 148
Blades. *See also* Vanes.
Brumfield criterion, 187–191

C

Cavitation
definition, 87, 88
NPSHR cavitation free, 92–103, 107
SIGMA cavitation free, 107–108
suction specific speed cavitation free, 107–108
Thoma, 186
vibration, 430–432
Chemical pumps, 283–329
ANSI specifications, 283–290
bearing housing, 298
bearing life, 285
bearings, 299–303
bedplates, 304
canned motor, 305–306
casing, 293–297
cover, 297
frame, 297
hydraulic coverage, 288
impeller, 290, 293
mag drive, 305–306
material, 323–329
mechanical seals, 387–388
non-metallic, 309–323
nozzle loading, 318
NPSHR, 317
processes, 323–328
sealless pumps, 305–306
sump pumps, 307–309
support head, 298
CO_2 pumps, 166–172
BHP (brake horsepower), 168–172
dry running applications, 167
enhanced recovery, 166
GHP (gas horsepower), 168–172
lubricating properties, 166
mechanical seals, 167
pump selection, 169–172
Concentric bowl pump, 179, 184–185, 191–195

Critical speed analysis. *See also* Vibration and noise.
Curve shape, 4–6, 175, 194–196

D

Deflection, 340–342, 563
Destaging, 144, 151–152
Differential head, 3
Diffuser area ratio, 180
Diffuser casing, 223–224
Double case pumps, 206–225
  abrasives, 219
  applications, 208–209, 219
  auxiliary take-off nozzles, 212
  balance disk, 208
  balance drum, 208
  baseplates, 218
  bearings, 216, 218
  boiler feed, 209
  casing diffuser, 222
  casing inner, 210–212
  casing volute, 223–224
  charge, 209
  dynamic balance, 223–224
  floating ring seals, 216
  foundations, 218
  impeller, 212–213
  impeller deformation, 220–222
  mechanical seals, 213
  pipeline, 210
  rotor dynamics, 219–220
  sag bore, 224
  throttle bushings, 213–215
  waterflood, 210
Double-suction pumps, 77–84
  area progression, 83
  impeller, 79
  nozzle layout, 79–84
Dynamic balance, 223–224
Dynamic head, 179–180

E

Efficiency
  crossovers, 69–70
  diffuser, 181
  estimating, 15–18, 43
  high speed pumps, 183–185, 194, 222
  hydraulic turbines, 247, 254, 256
  hydrodynamic size, 141–143
  pump losses, 140–141, 183
  specific speed, 141
  volute types, 57
Erosion wear rate, 226–227

F

Factoring law, 21–25

G

Gears, 200–202

H

Head
  actual, 180–181
  coefficient, 177, 181
  differential, 3
  dynamic, 179–180
  static, 179–180
  theoretical, 179–180
  velocity, 179–180
  vortex, 179–180
High speed pumps, 173–205
  Barske-type pump, 175–179
  bearings, 202–203
  Brumfield criterion, 187–191

High speed pumps (continued)
concentric bowl pump, 179, 184–185, 191–195
diffuser efficiency, 181
diffusion recovery, 181
efficiency, 183–185, 194, 222
flow coefficient, 177, 180, 182
full emission design, 178
gears, 200–202
head actual, 180–181
head coefficient, 177, 181
head dynamic, 179–180
head static, 179–180
head theoretical, 179–180
head velocity, 179–180
head vortex, 179–180
high solidity impellers, 194–196
impeller diameter, 184
impeller eye diameter, 194–195
inducers, 187–191
lubrication, 203
noise, 196
NPSHR, 187–190
partial emission design, 178–181
pump losses, 183
radial load, 192–193
shaft dynamics, 203–204
specific speed, 181–183
suction specific speed, 186, 189
Thoma cavitation parameter, 186
throat area, 184
volute collector, 192–194
Hydraulic gradient, 139
Hydraulic turbines, 246–282
application range, 248
applications, 272–279
BHP (developed horsepower), 249
costs, 247–248
efficiency, 247, 254, 256
gas release, 271
guide vanes, 261–271
impeller design, 257, 261–262
impulse type, 246, 250
losses, 252–254
NPDHA (net positive discharge head available), 249
NPDHR (net positive discharge head required), 249
operation, 279–281
performance prediction, 251–252, 254–255
radial load, 258, 259
reaction type, 246, 250
selection process, 248, 251
specific speed, 249–250
TAEH (total available exhaust head), 249
testing, 271–272
TREH (total required exhaust head), 249
two-phase flow, 271
Hydrodynamic size, 141–142

I

Impeller
area between vanes, 39, 41
chemical pumps, 290, 293
closed, 133
collet mounting, 131–132
deformation at high speed, 220–222
design, 28–44
diameter, 29, 184
discharge angle, 29–30, 41, 42
double-suction, 79
eye diameter, 32, 194–195
high solidity, 194–196
hydraulic turbine, 257, 261–262

inlet angle, 37
layout, 37–39
open, 175–176, 292–293
semi-open, 133
trim correction, 16–19
underfiling, 143–144, 146–147
vane number, 29–30, 42, 236
vane thickness, 40, 236
width b^2, 31, 43
Inducers, 187–191

L

Losses in a pump, 183, 252–254
Lubrication. *See also* Bearings and lubrication.
high speed pumps, 203

M

Material
chemical pumps, 323–329
mechanical seals, 379–382
non-metallic pumps, 309–329
slurry pumps, 228, 230–234
Mechanical seal reliability, 556–567. *See also* Mechanical seals.
annulus, 563–564
failure analysis, 557–559
failure causes, 559–566
low flow operation, 565–566
misalignment, 562–563
shaft axial movement, 564
shaft deflection, 563
Mechanical seals, 354–421. *See also* Mechanical seal reliability.
applications, 390–401
balance, 360–361
chemical service, 387–388
classifications, 373–379
CO_2, 167
double-case pumps, 213
face pressure, 361–362
floating ring, 216
gland, 383–386
installation, 401–418
leakage, 370–371
materials, 379–382
mating ring types, 381, 383
power consumption, 363–364
PV (pressure-velocity) 363, 371–372
refinery service, 389–390
temperature control, 364–370
theory, 355–359
throttle bushings, 213–215
troubleshooting, 401–418, 420
upstream pumping, 386–387
wear, 371–372
Misalignment. *See also* Alignment.
Model law, 19–20
Multi-stage pumps
anti-rotation splitters, 72
casing design, 65–76
core assembly, 73–74
crossover design, 67–72
crossover efficiency, 69–70
patterns, 72–73

N

Noise. *See also* Vibration and noise.
high speed pumps, 196
Non-metallic pumps, 309–329
NPDHA (net positive discharge head available), 249
NPDHR (net positive discharge head required), 249

NPSH (net positive suction head), 85–109
NPSHA (net positive suction head available), 4, 85–86, 104–106, 116–117, 120–121, 129, 131, 186–187
NPSHR (net positive suction head required)
  40,000 hours life, 92
  cavitation, 87–88, 189–191, 430–432
  cavitation free, 92–103
  cavitation free compared with Hydraulic Institute, 107–108
  chemical pumps, 317
  estimating, 33–34
  gas-to-liquid ratio, 100–103
  high speed pumps, 90, 92–93
  incidence angle, 94, 101
  inducers, 187–190
  leakage—influence of, 94–95
  liquid—influence of, 99–100
  suction nozzle—influence of, 96–99
  suction specific speed—influence of, 90
  predicting, 87–88
  stable operating window, 90–92
  suction piping, 103
  suppression test, 88–90
  Thoma cavitation parameter, 186
  vertical pumps, 120, 129
  viscosity, 103

P

Performance
  curve shape, 4–6, 175, 194–196
  hydraulic turbines, 251–252, 254–255
  prediction, 25–27
  range charge, 47–50
  slurry corrections, 244–245
Pipeline Pumps, 139–164
  analysis of pump losses, 140–141
  bi-rotors, 148
  destaging, 144, 151–152
  double case, 210
  efficiency, 140–143
  hydraulic gradient, 139
  hydrodynamic size, 141–142
  impeller underfiling, 143–144, 146–147
  number of pumps, 140
  number of stations, 140
  pipeline horsepower, 139
  pump selection, 158–164
  series vs. parallel, 163–164
  slurry, 156
  specific speed, 141
  volute chipping, 143–145, 147
  volute inserts, 147–150
Pulsations, 150, 433–436, 481–483

R

Radial load, 52–56, 192–193, 258–259, 296
Rotor dynamic analysis. *See* Vibration and Noise.
Rotor dynamics, 219–220

S

Sag bore, 224
Sealless pumps, 305–306

Shaft design, 333–343
deflection, 340–342
dynamics, 203–204
failure, 483–484
fatigue analysis, 336–339
key stress, 343
torsional stress, 334
Slurry pumps, 226–245
abrasivity, 226, 232–234
bearings, 237
casing design, 234–236
drivers, 240, 243
erosion wear rate, 226–227
impeller design, 236
materials, 228, 230–234
performance-slurry corrections, 244–245
pump types, 232–233
sealing, 237–238, 240
selection factors, 228
specific gravity, 228–229
specific speed range, 232
sump design, 240
wear plate, 236–237
Specific gravity
horsepower—influence on, 8
pressure—influence on, 7
slurry mixture, 228–229
Specific speed
definition, 11
efficiency—influence on, 15–18
high speed pumps, 181–183
hydraulic turbines, 249–250
metric conversion, 12
nomograph, 14–16
slurry pumps, 232
vertical pumps, 113–114
Static head, 179–180
Submersible pumps, 115–116
Suction specific speed
definition, 11–12
high speed pumps, 186
inducers, 189
NPSHR—influence on, 90
stable operating window, 90–92
System analysis, 3–10
differential head, 3
liquid characteristics, 7
NPSHA, 4
pump construction, 8
pump speed, 6
curve shape, 4
specific gravity, 7
viscosity, 7

T

TAEH (total available exhaust head), 249
Theoretical head, 179–180
Torsional analysis, 462–469
Torsional stress, 334
TREH (total required exhaust head), 249
Two-phase flow, 271

U

Unbalance, 426, 457–461

V

Vanes
area between, 39, 41
discharge angle, 29–30, 41–42
hydraulic turbine, 257
inlet angle, 37
layout, 37–39
number, 29–30, 42, 236

Vanes (continued)
pump out, 236
thickness, 40, 236
Velocity head, 179–180
Vertical pumps, 113–138
barrel-mounted, 120–122, 128
boiler feed, 130
bowl assembly, 131–133
can-mounted, 120–122, 128
column assembly, 134–135
condensate, 129
condensor cooling, 126
cooling tower, 127
cryogenic, 130–131
flood control, 127
head assembly, 135–136
heater drain, 129
loading, 131
pipeline booster, 131
process, 129
specific speed range, 113
submersible motor drive, 115–116
transfer, 127–128
vibration, 136–138
well pumps, 114, 122–124
wet pit pumps, 116, 125
Vibration and noise
acoustic frequency, 150–155
acoustic resonance, 436–440
acoustic velocity of liquids, 486–491
bearing housing resonance, 474–477
bearing stiffness and damping, 445–448
bearings, 428–429
blade passing frequency, 150–155, 474–475
causes of vibration, 424–441
cavitation, 430–432
critical speed analysis, 442–443, 477–481
critical speed evaluation, 451
critical speed map, 443–445
flashing, 430–432
impact tests, 472–477
instabilities, 440–441, 484–486
liquid noise, 423
measurement techniques, 471–474
mechanical noise, 423
misalignment, 426–427
piping, 428
pulsations, 150, 433–436, 481–483
rotor stability analysis, 461–462
rotor dynamic analysis, 441–456
seal effects, 448–451
shaft failure, 483–484
torsional analysis, 462–469
transients, 429
troubleshooting, 474–491
turbulence, 432–433
unbalance, 426, 457–461
vertical pumps, 136–138
vibration levels, 457, 459–461
Viscosity
NPSHR—influence on, 103, 107
oil, 526–527, 536
performance—influence on, 7
Volute
area, 33, 184
chipping, 143–144
circular, 56
collector, 192–194
concentric, 235
concentric—semi, 235
cutwater diameter, 36
design, 50–64
diffusion angle, 50
diffusion area ratio, 180

double, 52–54
efficiency comparison, 57
inserts, 147–150
quad, 54
radial load, 52–56
single, 52
spiral, 235
triple, 54
universal layouts, 58–62
velocity constant, 35
velocity ratio, 50–51
width, 36
Vortex head, 179–180

W

Waterflood pumps
double-case pumps, 210
enhanced recovery, 166
materials, 166
secondary recovery, 164