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ROTARY SHAFT SEAL SELECTION HANDBOOK FOR PRESSURE-EQUALIZED, DEEP OCEAN EQUIPMENT

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DEEP OCEAN TECHNOLOGY PROGRAM
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ROTARY SHAFT SEAL SELECTION HANDBOOK
FOR PRESSURE-EQUALIZED, DEEP-OCEAN EQUIPMENT

By
Kenneth R. Sasdelli and John D. Spargo

October 1971 7-573

ABSTRACT

The various considerations involved in the selection and application of off-the-shelf seal configurations to the rotating shafts on pressure-compensated deep ocean equipment are discussed. Seal performance data from deep-ocean electric propulsion motor tests and seal screening tests are included to give the seal user an understanding of the level of performance obtainable from an off-the-shelf seal. The handbook will be revised to include additional seal performance data to be available from a laboratory investigation of representative rotating shaft seals now under way.

PREFACE

The text of this handbook is organized around the suggested rotating shaft seal selection procedure, table 1. The procedure shows in diagrammatic form the several variables and alternatives which must be considered in the selection process and suggests the sequence in which they be considered.

Chapter I - Establishes seal system requirements; brings to the readers attention those seal and machinery system variables which are important to the sealing problem, those variables unique to deep-submergence applications, and stresses the importance of specifying these variables in detail to those responsible for manufacturing the seal.

Chapter II - Considers sealing alternatives; discusses various seal types, inherent features and limitations, which are inherently best suited to deep submergence applications in light of seal system requirements, seal and seal system variables, and design alternatives.

Chapter III - Selection factors; discusses relative advantages and disadvantages of various design features and ways to apply the seal.

Chapter IV - Validates seal system selection; gives typical performance data on off-the-shelf seals, operational experience, and what kind of performance to expect to aid the user in selection judgment.

Table 1 – Seal Selection Considerations

Consider Machinery Alternatives

- Hard Shell
  - High-pressure seal in liquid/air environment required
- Free Flooded
  - No seals required
- Fluid Filled, Prssure Compensated
  - Low-pressure seal in liquid/liquid environment required

Establish Seal System Requirements

- Normal Operating Conditions
  - Function of seal
  - Number of penetrations to be sealed
  - System ambient pressure
  - Magnitude and direction of pressure differential
  - Type shaft motion, speed and size of shaft, stop-start frequency
  - Fluid environment
  - Physical environment
- Operating Practices
  - Testing
  - Filling
  - Transporting and storage
- Performance Requirements
  - Acceptable leakage rates
  - Acceptable power loss
  - Acceptable life
  - Availability

Consider Sealing Alternatives

Seal Types

- Clearance Seals
  - Contact
  - Controlled Clearance
  - Fixed Clearance
- Fluid Force Seals
  - Viscous or screw
  - Centrifugal or slinger
  - Magnetic liquid
- Hermetic Seals

Variables and Limitation Common to Contact Seals

- Forces on Seal
  - Hydrostatic forces, concept of a balanced seal
  - Spring forces
  - Elastic forces
- Leakage of the Seal
  - Hydrostatic leakage
  - Hydrodynamic leakage
  - Combined leakage
  - Design variables
- Seal Design Alternatives
  - Seal operates hydro-dynamically
  - Seal operates under mixed film or contact conditions
- Seal System Alternatives
  - Slinger and bushings
  - Multiple seals
  - Flushing arrangements
  - Filter arrangements

Selection Factors

Basic Differences

- Mounting Convenience, Space Requirements
- Directional Sealing Ability
- Availability and Cost
- Performance Limits

Table 1 (Cont)

Design Features and Application Considerations for Various Configurations

Axial Configurations

Mechanical End Face Seal

- Hydraulic balance ratio
- Directional sealing ability
- Secondary seal
- Springs
- Mounting, type construction
- Face materials

O-Ring Slipper Seal

- Conformity of slipper
- Materials
- Mounting

Radial Configurations

General Considerations

- Mounting
- Mating surface finish
- Alignment, tolerances on shaft diameter
- Compatibility of mating surface and sealing element material

Lip Seal

- Materials
- Mounting
- Lip configuration

O-Ring

Slipper Seals

- Conformability of slipper
- Materials
- Mounting

U-Cups

- Mounting
- Springs

Validate Seal System Selection

Performance Data

Annapolis Seal Screening Test

Annapolis Drive System Tests

Other

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INTRODUCTION

The limited commercial market for deep-submergence equipment and dynamic seals for this equipment has been too small to generate sufficient interest or to justify costly development of special seals by the equipment or seal manufacturer. Thus, economic, logistic, replacement, and maintenance considerations have resulted in the use of commercially available seals.

The process of selecting a commercially available seal requires communication between the seal user and seal manufacturer. The seal user, and only the seal user, knows what operating conditions are imposed on the seal by the machinery system and what performance is required from the seal. His role in the selection process is to make the seal manufacturer aware of these application requirements. The seal manufacturer, aware of the user's requirements, can then select or design the desired features into the seal to meet particular requirements. The user, to be effective in this process, must be aware of the many seal and machinery system variables which must be considered in the selection process, the type of information required by the seal manufacturer, the general types of seals available, their limitations, and application considerations. This handbook was assembled to provide this information to supplement the open literature on seals and the information of seal manufacturers. It is limited to consideration of dynamic seals suitable for application to the fully rotating shafts on pressure-compensated, deep-submergence equipment.

BACKGROUND

MACHINERY ALTERNATIVES

Deep-submergence machinery systems are placed external to the pressure hull, in the sea-water environment. Several configurations for protecting the rotating machinery from the sea-water environment are possible. Each configuration, as discussed below, presents a different dynamic sealing problem.

Hard Shell

The rotating machinery can be enclosed in a sealed case or hard shell so that the machinery operates in air at atmospheric pressure. The disadvantages of this approach are the size and weight of the shell needed to withstand deep-submergence pressures. A dynamic seal capable of limiting the leakage of seawater into the shell under very high-pressure differentials, up to 13,500 pounds per square inch differential (psid), would be required at the shaft penetrations. A dynamic seal for such pressures is beyond the state of the art and beyond the scope of this handbook.

Free-Flooded Machinery

The machinery can be open to the sea and specifically designed to operate in a sea-water environment. The advantages of this approach are that a light machinery structure can be used; a supply of machinery fluid is readily available; and no dynamic seals are required. This approach, however, is limited to machinery systems where the poor lubricating, electrical, and corrosive properties of seawater can be tolerated.

Fluid-Filled, Pressure-Compensated System

The machinery system can be filled with a fluid which provides corrosion and electrical protection, lubrication, and cooling. The fluid within the machinery can be maintained at sea pressure by a pressure-equilization system. This approach has the advantage of light weight and increased reliability. Dynamic seals at the machinery-shaft-sea interfaces are required. The sealing problem, however, is less demanding than the hard shell approach because zero pressure, at the most, a small differential exists across the seal.

This guide concerns itself with selecting off-the-shelf seals for this fluid-filled machinery configuration.

SEALS AND SEAL SYSTEMS IN USE

Description

Off-the-shelf seals, of the axial mechanical face, and elastomer lip configurations have been used.

The simplest system uses one seal at the machinery-shaft-sea interface and one pressure compensator, to maintain the machinery compartment at sea pressure, or slightly higher.

The majority of the seal systems on the electric drives use two axial face seals at the machinery-shaft-sea interface and two pressure compensators. The two seals separate two compartments, as shown in figures 1 and 2. The inboard seal separates the machinery compartment from an intermediate seal cavity. The outboard seal separates this cavity from the sea. The compensating fluids in the machinery and seal cavity are usually the same. The inboard seal has the same compensating fluid on both sides. The outboard seal has compensating fluid on one side and seawater on the other side.

In a redundant arrangement, there is no pressure drop across either seal and both the machinery and seal cavity are maintained at sea pressure. The compensator on the machinery cavity is usually independent of the compensator on the seal cavity, figure 1.

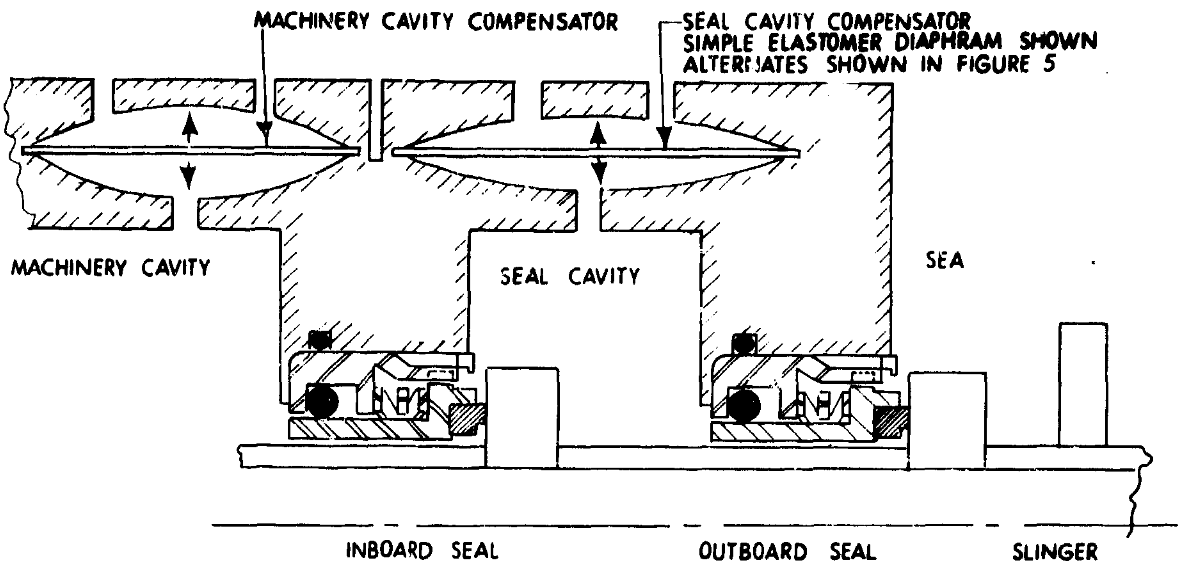

Figure 1 - Double Seal, Redundant Arrangement (ΔP Nominally Zero)

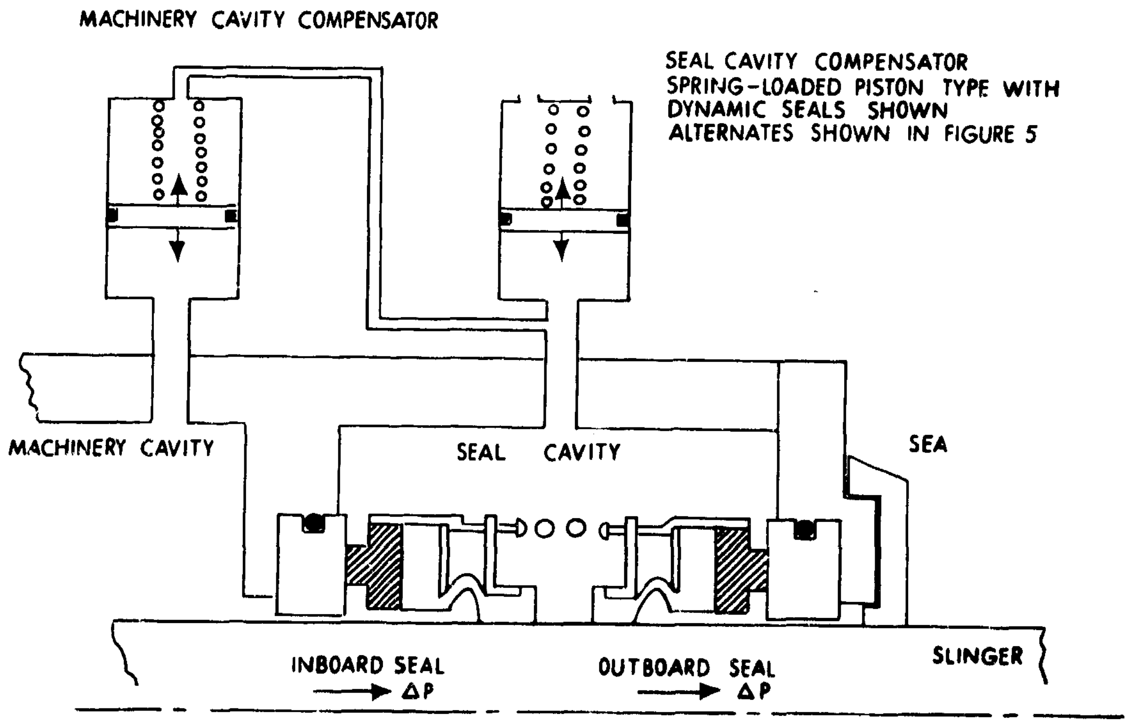

Figure 2 - Double Seal, Pressure Cascaded Arrangement (ΔP Nominally 2 to 5 PSID)

In a pressure-cascaded arrangement, the seal cavity is maintained at slightly higher pressure than the sea (typically 2 to 5 psid) by the seal-cavity compensator. The high-pressure side of the seal-cavity compensator is then connected to the low-pressure side of the machinery compartment compensator, which in turn maintains the machinery compartment at a slightly higher pressure (typically 2 to 5 psid) than the seal cavity, figure 2. Thus, a pressure drop exists across both seals and from the machinery cavity to the sea.

The rationale of these systems is that the seal will not leak when there is no pressure differential or will leak in the direction of the pressure differential.

Problem Areas

● Machinery and Operations - Related Problems - Application Problems. Problems arise because the equipment designer, manufacturer, or operator is unaware or fails to appreciate how features of the machinery system or certain operational procedures can interact with and affect the seal. Examples of this type of problem are:

• Machinery Failure/Seal Failure Interaction. Problems can arise when off-the-shelf machinery, originally intended to operate in an air-ambient-environment, is adapted to a deep ocean, high-pressure environment. Because of particular features of the machinery, an unbalanced hydrostatic thrust load may develop on the shaft of the machinery when the system is pressurized. In one application, the machinery had no thrust bearings to absorb this load since, in its intended application, there was no thrust. The shaft moved axially until the thrust load was absorbed by a face seal in the system. This resulted in destruction of the seal and subsequently a machinery failure.

• Pressure Compensator/Seal Interaction. Instances of malfunction have been reported in which the pressure compensator allowed the magnitude and or direction of the pressure differential to go beyond design limits. In some instances, this has resulted in catastrophic mechanical failure of the seal or blowing open of the seal with high leakage.

• Operational Procedure/Seal Interaction. Pressure or vacuum filling of the machinery system, to rid it of air, has in some instances resulted in seal failure. Excessive filling pressure can force the seal open and/or mechanically overstress it, causing permanent damage. Vacuum filling will reverse the normal direction and magnitude of the pressure differential across the seal. This can pull the seal open and also cause permanent damage.

- Inherent Features of the Seal. Hydrodynamic pumping effects in face and lip seals, when operated under low-pressure differentials, have been reported in the literature.[1-7] Hydrodynamic pumping can result in leakage where there is no hydrostatic pressure differential across the seal, or in leakage against the pressure differential. Pumping is usually not a problem in the more conventional applications, where the seal is in a liquid/air environment. The ingestion of air or mixing and contamination of the fluids usually poses no serious problem. At high-pressure differentials, the pumping leakage usually goes unnoticed because it is masked by the hydrostatic leakage. However, pumping could pose a problem in deep-submersible applications because of the existing low-pressure differentials and the liquid/liquid environment of the seal, particularly for a seal at the machinery/sea interface. At the present time, there is no way to predict from theory or experience whether a given seal design will pump, in which direction it will pump, or the magnitude of the leakage.

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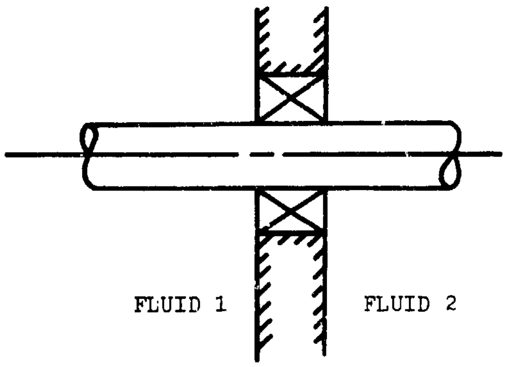
[1]Superscripts refer to similarly numbered entries in the Technical References in Chapter V of this handbook.

CHAPTER I

ESTABLISH SEAL SYSTEM REQUIREMENTS

The first and most important step in selecting a seal is to define the application requirements. Discussions with seal manufacturers indicate their experience has been that the majority of seal problems, in all types of applications, can be traced to some seal-operating condition, operating procedure, or seemingly unrelated operating condition within the system which was not appreciated and thus not defined by the seal user. Consequently, no provisions were designed into the seal or seal system to cope with these conditions, and the seal fails in operation. This lack of appreciation has resulted in problems on deep-ocean applications that can be prevented by careful consideration of the seal and its system interaction while the machinery system is still in the design stage.

The seal user is urged to consider and specify the following parameters in the light of his particular application. The specifications should be passed on to those responsible for selecting or designing the seal. A specification form, figure 3, is provided for this purpose.



Show print or sketch of area where seal is located, features of the shaft over which seal must be assembled (keyways, splines, threads, etc), location of the seal in relation to bearings, type of bearings and features of housing should be shown. Fill in specifications for each shaft penetration.

<u>Fluid Level in Relation to Seal</u>

	Fluid 1	Fluid 2
Continuously submerged	☐ Yes ☐ No	☐ Yes ☐ No
Periodically exposed to air	☐ Yes ☐ No	☐ Yes ☐ No

<u>Type Fluid on Either Side of Seal</u> (if proprietary give properties)

- Normal Operating Fluids

Type____Applicable Spec____	Type____Applicable Spec____
Fluid Temperature, Min____° F	Fluid Temperature, Min____° F
Max____° F	Max____° F

Figure 3 - Seal Specification Form

Viscosity, Max____ at Max Pressure and Min Temperature

Min____ at Min Pressure and Max Temperature

Vapor Pressure, at Max Temperature____

at Min Temperature____

Surface Tension, at Max Temperature____

at Min Temperature____

Abrasives in Fluid ☐ Yes ☐ No

(If Yes, give composition and concentration)

Viscosity, Max____ at Max Pressure and Min Temperature

Min____ at Min Pressure and Max Temperature

Vapor Pressure, at Max Temperature____

at Min Temperature____

Surface Tension, at Max Temperature____

at Min Temperature____

Abrasives in Fluid ☐ Yes ☐ No

(If Yes, give composition and concentration)

Fluids Used for Storage, Flushing, or Test (If different than normal operating fluids, specify.)

Type____ Applicable Spec____

Fluid Temperature, Min____ ° F

Max____ ° F

Type____ Applicable Spec____

Fluid Temperature, Min____ ° F

Max____ ° F

System Pressure

During normal operation, Min
Max
During filling, Min
Max
During testing, Min
Max

Magnitude and Direction of Pressure Differential

During normal operation
Direction 1 → 2 ☐
2 → 1 ☐
Magnitude, Min
Max
During filling
Direction 1 → 2 ☐
2 → 1 ☐
Magnitude, Min
Max
During testing
Direction 1 → 2 ☐
2 → 1 ☐
Magnitude, Min
Max

Figure 3 (Cont)

Space and Mounting Requirements

Mounting bore size, Max OD_____

Min OD_____

Shaft size, Max Dia_____

Min Dia_____

Axial length, Max_____

Min_____

Is a split seal required? ☐ Yes ☐ No

Are modifications to shaft, such as sleeves, steeps, grooves, etc, or to housing acceptable? ☐ Yes ☐ No

From which direction is seal assembled? From Side 1 ☐ From Side 2 ☐

Materials

Shaft material_______________Hardness______________Surface finish_______________

Machinery mounting bore material__________________Bore finish___________________

Shaft Speed

☐ Rotating

☐ Unirotational

☐ Birotational

Speed range, Max_____rpm

Min_____rpm

Stop-start or reversing frequency, _____starts/hour

Approximate duty cycle, _____% time birotational

_____% time static

_____% time at low speed

_____% time at intermediate speed

_____% time at max speed

☐ Oscillating or discontinuous rotation

Range_____Degrees

Rate_____Degrees or cps

Shaft-to-Mounting Bore Variations

Possible axial shaft-to-bore assembly variations due to stack up tolerances, _____in.

Axial shaft-to-bore movement in operation_____in., rate_____c/sec

Radial shaft-to-bore misalignment (eccentricity plus radial dynamic runout at sea location), _____in. Angular misalignment at seal location, _____degrees

Figure 3 (Cont)

Performance Requirements

Acceptable leakage rates

Q1 → 2_____cc/hr

Q2 → 1_____cc/hr

Acceptable life, _____hours

Acceptable losses

Breakout or starting torque, _____oz-in.

Running torque, _____oz-in. at _____rpm

Other

Figure 3 (Cont)

NORMAL OPERATING CONDITIONS

Functions of the Seal

A seal at the machinery/sea interface must prevent the escape of compensating fluid from the system and must exclude seawater from the machinery system.

An internal seal, at the interface of different machinery elements, may be required for several purposes:

- To limit leakage between two different types of compensating fluids. Because of the different fluid requirements of each machinery element in the system, it may be necessary to use different types of compensating fluids in each machinery element (gearbox, motor).

- To serve as a contaminant barrier. Contaminants generated in one machinery element, such as brush wear products in d-c* motors, can spread within the machinery system and cause damage to other machine elements.

- To serve as a thermal barrier. One machinery element may operate at a higher temperature than other elements in the system. If the fluid in this element is allowed to circulate, it can damage or impair the efficiency of other elements. Machinery lubrication can be affected by viscosity changes due to heat.

*Abbreviations used in this text are from the GPO Style Manual, 1967, unless otherwise noted.

Leakage from a seal at the machinery-sea interface would be more serious than that from an internal seal.

Seals are sometimes used on sea-water-flooded machinery applications, figure 4. Seawater is usually admitted to the machinery cavity through a filter medium or through a settling chamber arrangement so that the machinery cavity and bearings operate in relatively clean seawater. The seal then functions to keep the clean water in and the dirty seawater out of the system.

Number of Penetrations to be Sealed

In sea-water and hydraulic pump applications, figure 4, it is usually possible to couple the prime mover, an electric or hydraulic motor, to the pump without a sea-water penetration. An internal seal at the motor/pump interface may, however, be required.

Electric propulsion systems, figure 4, have one sea-water penetration. In this application, an electric motor drives a propeller directly or the motor is directly coupled to a reduction gear which drives a propeller for main propulsion or thrusting. The motor and reduction gear usually share the same compensating fluid. The seal is on the output shaft of the reduction gear and seals both the gearbox and motor from the sea. An internal seal, at the motor/reduction gear interface, may be required for some coupled-drive-system designs.

Applications such as cyclodial propellers, pods, and steering gear (figure 4) can have a number of different sizes of sea penetrations, each with a different type of shaft motion, in the same machinery housing.

System Ambient Pressure at the Seal

The system pressure, i.e., the pressure inside and outside the machinery, is equalized by a pressure-compensating system. The system pressure is variable from air-ambient to the maximum submergence pressure of the equipment.

Considerations include the effects of pressure on the machinery, fluid viscosity, cavitation, and the mechanical integrity of the seal.

Magnitude and Direction of the Pressure Differential at the Seal

The magnitude and direction of the pressure differential across the seal is determined by the pressure compensator. Depending on the particular design of the compensator, the differential may be constant or it may vary with the volume of the compensator. For example, the pressure differential with a spring-loaded piston, or a simple elastomer diaphragm-type compensator, figure 5, will change as the fluid volume and extension or contraction of the spring or diaphragm change.

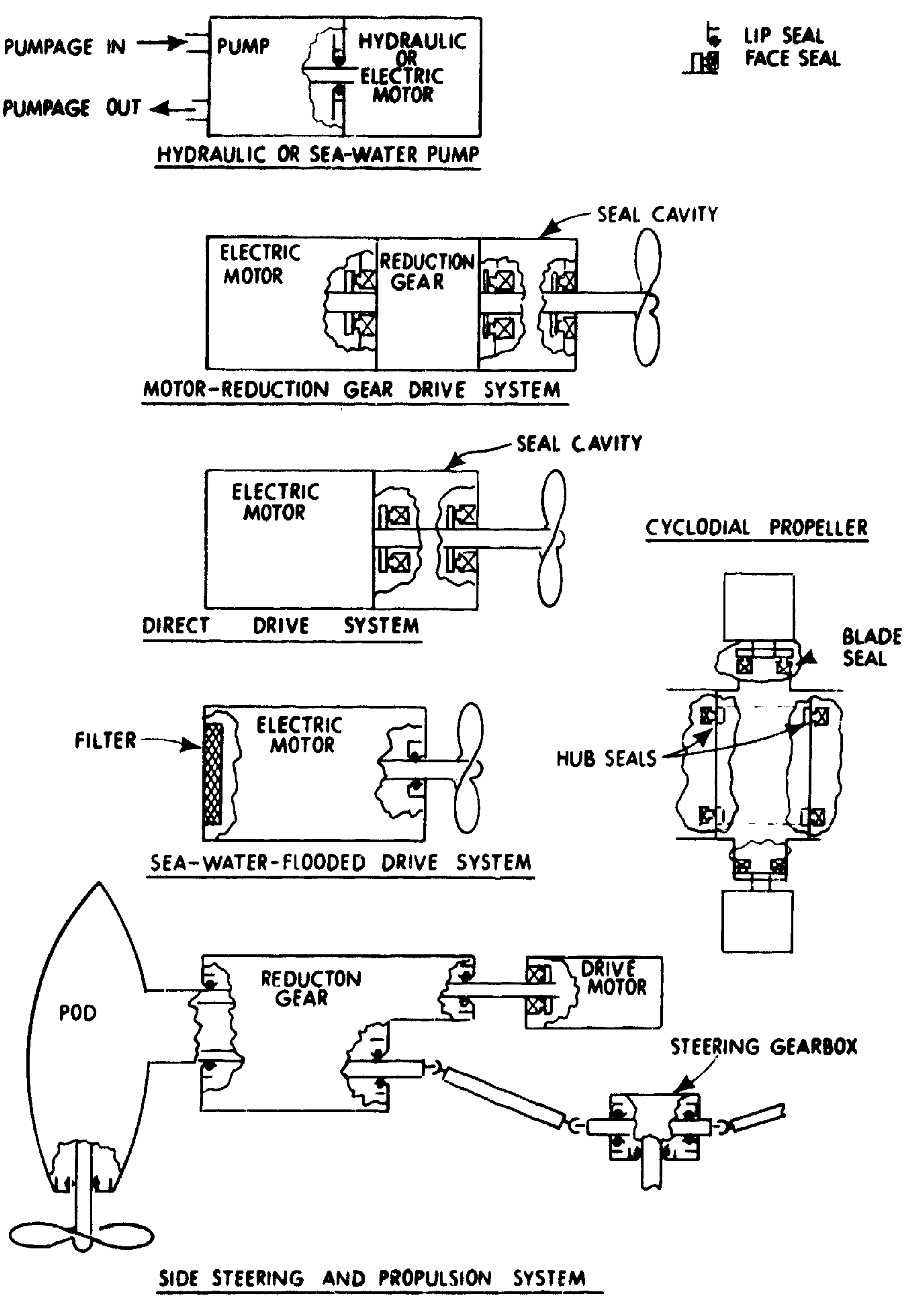
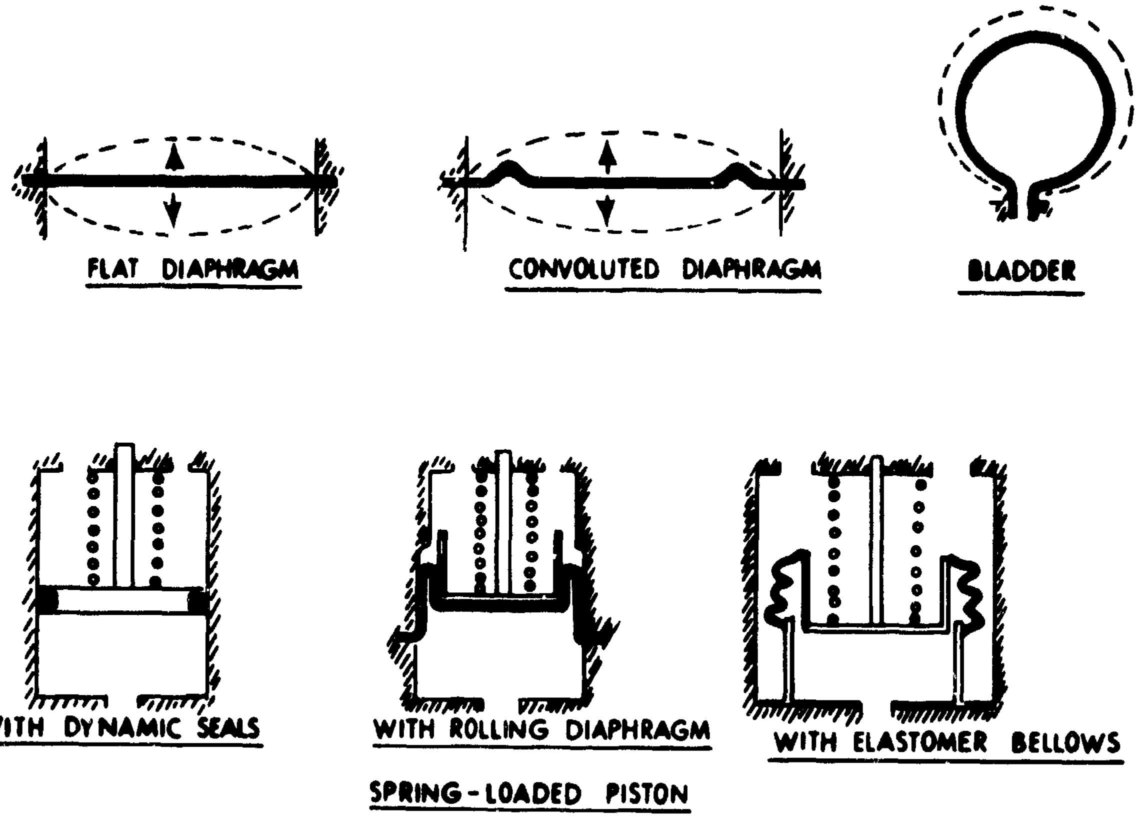
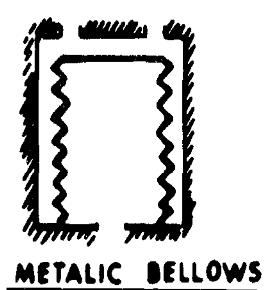


Figure 4 - Typical Deep-Ocean Dynamic Seal Applications





Check For:

1. Pressure differential - volume relationship
2. Possibilities for fouling or mechanical damage.
3. Possible interaction between different compensators in system.
4. Possible interaction between compensators and dynamic seals.

Figure 5 - Examples of Pressure Compensator Configurations

The user is urged to consider the interrelationship between the pressure compensator and dynamic seals on his particular application.[8]

- Does the compensator have a sufficient volume to accommodate changes due to fluid compressibility, temperature, and entrapped air?

● How does the pressure differential vary with the volume of the compensator? What are the pressure limits?

● Can the compensator become fouled with sea growth, sand, or silt or inhibited so that its useful working volume is reduced? Does the compensator itself contain seals which can leak or otherwise inhibit the motion of the compensator? Inhibiting the motion of the compensator can result in very high pressure differentials when the fluid temperature increases or a vacuum when the fluid temperature decreases.

● Is the direction of the pressure differential in the direction for which the seal was designed?

In existing systems, the machinery system is usually maintained at sea pressure or slightly above sea pressure, typically 2 to 5 psid.

Type of Shaft Motion, Speed and Size of Shaft, Stop-Start Frequency

● Rotating Shafts. The shafts on pump applications usually rotate in one direction, at constant speed, with an on-off duty cycle. The shafts are relatively small in diameter (0.5 to 1.5 inches) and rotate at relatively high speed, typically 1800 or 3600 rpm. The shafts on the electric drive systems rotate in either direction, at variable speed, in a variable-duty cycle. The output shafts are of relatively larger sizes (1.5 to 3 inches) and rotate at relatively lower speeds, typically 0 to ±100 rpm or 0 to ±600 rpm. Much larger shafts (8 to 24 inches) are used on pods and at the hub of cyclodial propellers. The shaft speed, however, is usually less than 100 rpm.

● Intermittent Rotating and Oscillating Shafts. Blade shafts on cyclodial propellers continuously oscillate at a fixed rate and through a limited angle. Other applications, such as pods and steering gear, have shafts which oscillate intermittently at a random rate and angle, while others may rotate intermittently but do not rotate through a full 360 degrees.

● Static Shafts. Most submersibles will spend considerable periods of time out of the water with the machinery inoperative, during a normal mission and during transit to an operational site. Present submersibles are capable of operating 8 to 24 hours submerged. The dynamic seals in the system will also be required to be effective as static seals for relatively long periods of time.

Fluid Environment of the Seal.

● Fluid Level and Type of Fluid on Either Side of the Seal. The seal is completely submerged in a liquid/liquid environment when the machinery is operative. The seal may have the same or different fluids on either side or it will have a compensating fluid on one side and seawater on the other side. The type of compensating fluid used is dictated by the fluid requirements of the machinery.[9] The fluid should be named. If proprietary, its properties should be specified.

A seal at the machinery/sea interface will be periodically exposed to air when the machinery is out of the water. The machinery may be operated in this condition for testing or checkout purposes.

• Effect of Pressure and Temperature on Viscosity. Viscosity is particularly significant in deep-submergence applications because of the very high system pressures. Most fluids considered for use as compensating fluids show a marked increase in viscosity at deep-submergence pressures.[9] Some are also non-Newtonian.[9]

• Abrasives in the Fluids; Tendency of the Fluids to Form Abrasives. A seal at the machinery-sea interface will be subjected to abrasives at the sea-water side. This may be particularly severe when the equipment is operating near the ocean floor and the propulsion system is churning up sand and silt. Abrasives can also be introduced into the compensating fluid from brush and component wear debris and from solids formed from the breakdown of the compensating fluid under electrical arcing which also produces gaseous products. The possibility of such gases expanding out of solution and resulting in changes in the pressure differential should be considered. Hard salt and silt deposits may also occur when the seal dries out with the equipment out of the water. The type, size and concentration of the abrasives should be specified.

• Chemical Activity of the Fluids. The effect of pressure and temperature on the corrosive properties of the fluids should be considered. The corrosive effects of seawater are well known. Sea-water-contaminated compensating fluids and the compatibility of various compensating fluids with commonly used materials is given in reference 9.

The possibility of the fluid breaking down and forming corrosive products, or of sludge formation and viscosity changes, should be considered.

• Vapor Pressure, Entrapped Gases. The vapor pressure of the liquids and the saturation pressure of gases dissolved in the liquids are important in relation to cavitation effects. Entrapped air also affects compressibility and viscosity.[9]

• Surface Tension. The significance of the surface tension will be considered later.

• Temperature of Environment. Deep-submersible equipment may be transported to its operational location by air, overland, or by sea. The seal must survive under the temperature extremes of these environments as well as under the operating temperature of the machinery:

- Air ambient: -40° to 140° F.
- Sea-water ambient: +28° to +80° F.
- Machinery fluid operating temperature: temperature of the environment to +190° F.

Physical Environment of the Seal.

- Space available for the seal.

- Provision for ease of assembly and replacement. Machinery configurations requiring a split seal assembly should be avoided if at all possible. A split seal will complicate the sealing problem. Effective sealing is difficult because of the discontinuity introduced at the seal interface by the split. In most existing applications, the seal can be assembled over the end of the shaft, thus not requiring split seals. The possibility and acceptability of using assembly aids such as shaft sleeves, shaft steps, collars, etc. should be considered.

- The Axial and Radial Runout, Shaft to Seal Eccentricity, Angular Misalignment, and Vibration of the Shaft Which the Seal Must Accommodate. Shaft-to-seal relative motion will generally cause increased leakage. Therefore, consideration should be given to the type of bearings used, the clearance in the bearings, the location of the seal in relation to the bearings, and the possibility of thrust or radial loads being placed on the seal. Placing the seal as close to the bearings as possible will minimize shaft to seal relative motion.

- Machinery Materials in Proximity of Seal. The possibility of setting up a galvanic couple between the machinery and seal materials and the possibility of stray currents resulting in impressed current corrosion should be considered.

OPERATING PROCEDURES

The seal user should analyze the operating practices and procedures to determine their effect on seal-operating conditions. These conditions should be specified. For example:

Testing Procedure

Tests of the machinery system, such as running the machinery without compensating fluid for purposes of determining churning losses, using higher pressures or higher pressure differentials during hydrostatic tests, or using a different compensator when testing, can subject the seal to more severe and or very different operating conditions than would normally be encountered.

Filling Procedure

Pressure or vacuum filling of the system, to rid it of air, can result in higher differentials or can result in reversed pressure differentials across the seal.

Transporting and Storage Procedures

Using a different type compensating fluid for flushing or as a preservative during storage can lead to material incompatibility problems if the fluids are not specified.

PERFORMANCE REQUIREMENTS

Acceptable Static and Dynamic Leakage Rates

Limits on the acceptable leakage rate of seawater into the system and of compensating fluid out of the system will vary for the different applications but will be determined by:

- The volume of the machinery and compensating system.
- The mission cycle.
- The susceptibility of the machinery system to sea-water damage.
- The ability of the compensating fluid to neutralize the harmful effects of seawater.
- The number of penetrations to the sea. A machinery system with a multitude of penetrations would place more stringent leakage requirements on each seal in the system than that for a machinery system with only one.
- The direction of the leakage. If it is absolutely necessary that the seal leak to function properly, it would be preferable to have the compensating fluid leak into the sea rather than the sea into the machinery.

Acceptable Power Loss and Breakout Torque

Machinery losses, appearing as heat, are dissipated through the machinery housing to the sea. There are usually no external, auxiliary provisions for cooling the seal or compensating fluid. Seal losses put an additional heat load on the system and reduce the overall efficiency of the machinery; the system's ability to dissipate this additional heat and the loss in efficiency which can be tolerated will determine the level of seal losses which are acceptable.

Heat generated by the seal can result in temperatures in excess of the system ambient locally at the seal. The temperature at the seal is important because of its effects on the seal materials and on the viscosity and vapor pressure of the fluid at the seal.

Breakout torque, i.e., the torque required to initiate rotation, is particularly important in positioning or control devices where low and predictable breakout torque is necessary to prevent overshooting.

Acceptable Life

The life, i.e., that period of time over which the seal must maintain the desired leakage rate, is determined by the mission cycle and time between overhauls for each specific application.

Availability

The use of an off-the-shelf seal as is, or one which can be readily modified to meet the particular application requirements, is desirable.

CHAPTER II

CONSIDER SEALING ALTERNATIVES

Consider now, in light of the application requirements, the general types of seals and their suitability for deep-submergence applications.

DESCRIPTION OF SEAL TYPES AND THEIR SUITABILITY

A variety of seals,[10,11] based on different sealing concepts, are in general use. They can be classified, by the way they restrict leakage, into three general categories: clearance seals, fluid-force seals, and hermetic seals.

Clearance Seals

Clearance seals restrict leakage by controlling the clearance and or geometry of the flow path between its dynamic and stationary elements. They can be further classified as:

- Contact Seals - seals with zero or uncontrolled clearance, figure 6. A contact seal consists essentially of two highly conforming surfaces, one of which is moving relative to the other, in bearing contact. Mechanical face seals, packing, and elastomer lip seals are common examples of this class of seal. They are referred to as contact seals because it is generally assumed that there is no clearance, i.e., true contact exists at the sealing interface. It has been shown, however, that these seal configurations can and do operate with a very minute film of fluid, or clearance, of the order of 100 μin or less, at the sealing interface. The clearance results from unplanned for or accidental effects rather than from some particular feature designed into the seal. Because the clearance is so small, they are capable of limiting leakage to low levels. The availability of this type of seal in a variety of configurations and materials makes them adaptable to a wide variety of applications. Availability, adaptability, simplicity, and low leakage capabilities make the contact type of seal the most attractive choice for deep-submergence applications. The configurations shown in figure 6 are possible candidates for deep-ocean applications. The contact-type seal will usually perform satisfactorily for the majority of rotating seal applications. Problems, in the form of high and erratic leakage, short life, and high mechanical losses, usually occur when they are used in applications with high differential pressure and/or high speed, with exotic fluids. In such applications, the seal designer will usually resort to a different type of seal, as described below.

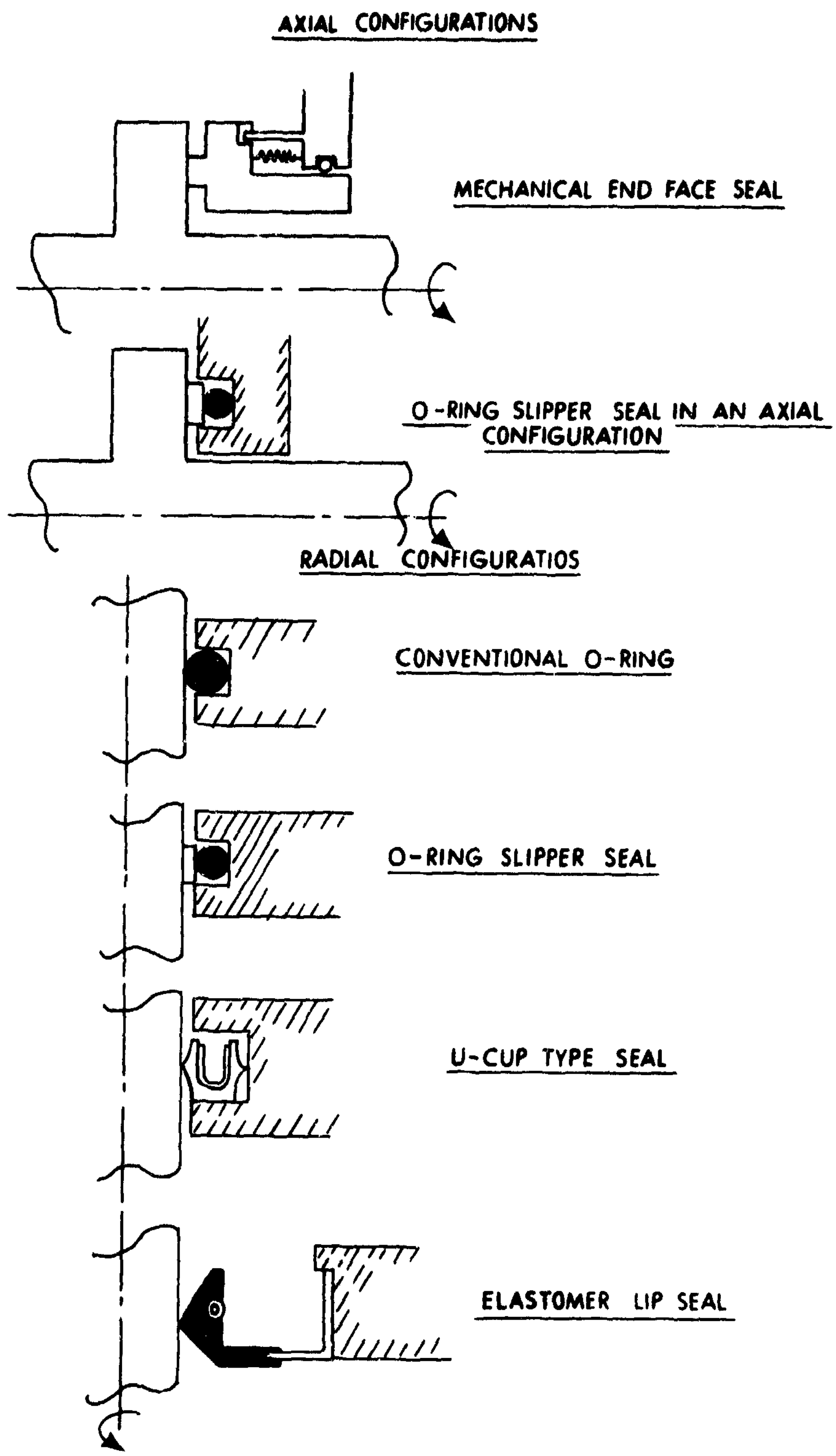


Figure 6 - Contact-Type Seal Configurations

• Controlled-Clearance Seal - clearance controlled by design features and operating conditions, figure 7. A controlled-clearance seal incorporates special, planned design features at the seal interface which utilize the pressure differential (seal behaves as a hydrostatic bearing) and/or the seal or shaft rotation (seal behaves as a hydrodynamic or hybrid bearing) to produce a controllable clearance at the seal interface. The clearance, and thus the leakage, vary with the pressure and/or speed of rotation in a predictable, controllable manner. They generally operate with greater clearance and thus have greater leakage than the contact type seal. Off-the-shelf hardware is not readily available because they are generally regarded as special purpose seals and must be designed for the specific application. They are used primarily in high-speed and/or very high differential pressure applications, where the simpler contact-type seal will not perform satisfactorily. The greater leakage and lack of availability make them unattractive alternatives to the contact-type seal.

• Fixed Clearance Seal - clearance fixed by design features, figure 7. The clearance in this type is built into the devices at the time of manufacture and remains essentially independent of operating conditions. Bushing and labyrinth seals are examples of this type. Off-the-shelf hardware is not available, since these seals are usually designed as an integral part of the machinery. The clearance, and thus leakage, will generally be greater than in either the controlled clearance or contact seal since enough clearance must be designed into the seal initially to accommodate manufacturing inaccuracies, shaft motion, and dimensional changes in operation.

Fluid-Force Seals

Fluid force seals achieve leakage restriction by controlling the fluid forces at the interface of their dynamic and stationary elements. Specific types are:

• Viscous or Screw Seal - utilizes viscous forces, figure 7. Sealing in a viscous device is accomplished by a pumping action which generates a pressure head in the device to oppose the applied pressure. This device is similar in construction to a screw pump. Its sealing ability depends upon rotation, it will not seal statically and will usually only seal in one direction of rotation (unirotational). Static sealing is usually accomplished by adding another type of seal, usually a contact seal, to the device. It is most effective at one given speed, pressure differential, and fluid viscosity. At its design point there is zero leakage; off the design point, it can pump.

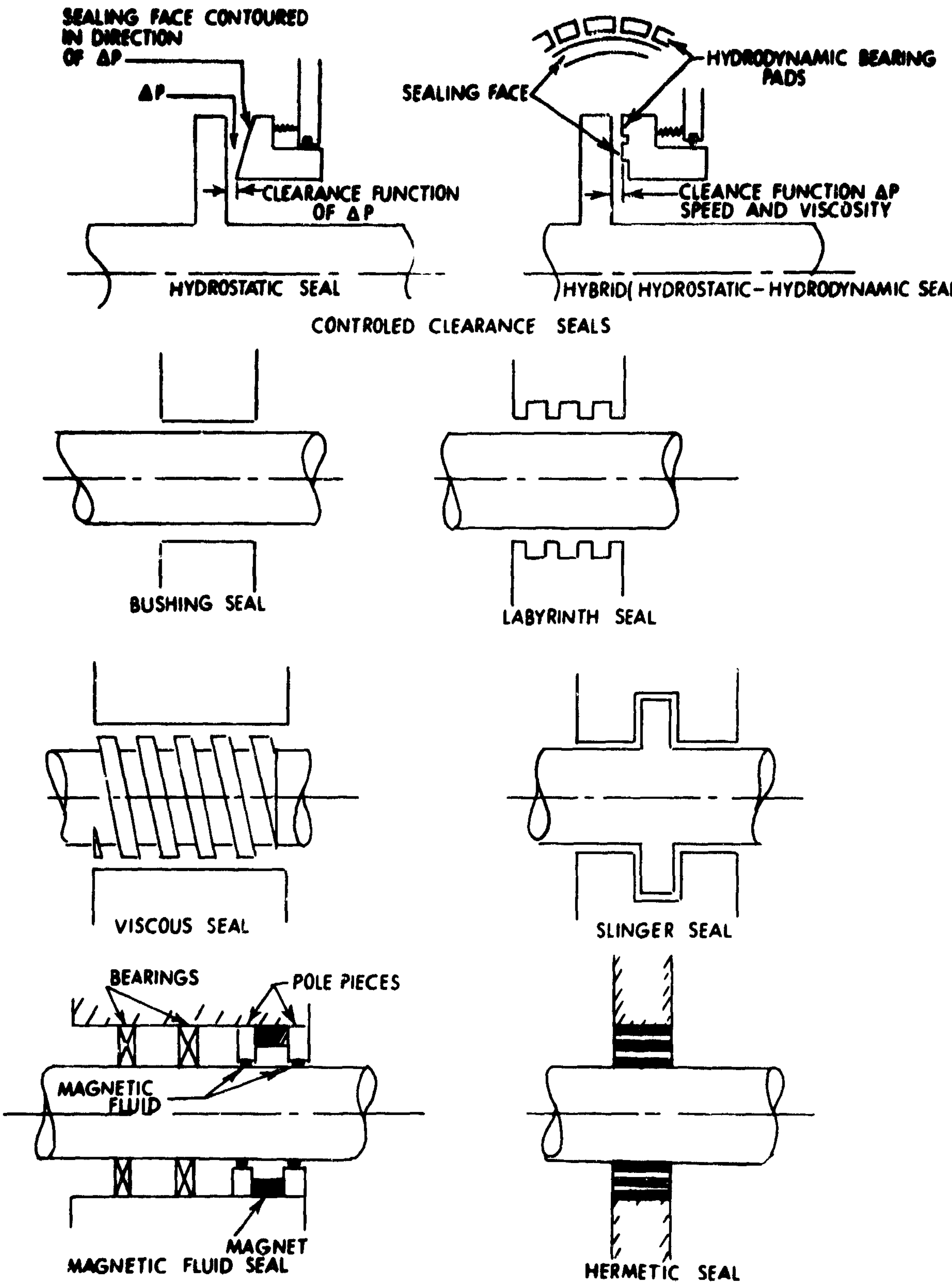


Figure 7 - Various Seal Configurations

• Centrifugal or Slinger Seals - utilize inertia forces, figure 7. In its simplest form, a centrifugal seal consists of a disk rotating in a close fitting annular space containing a fluid. The centrifugal force field generated by the rotating disk balances the pressure forces. It is similar to the viscous seal in that it will not seal statically and can pump when operated off its design point (i.e., a given density, speed and ΔP.). Since a seal in deep-submergence application is required to seal statically over a range of speeds and fluid properties, and may be required to be birotational, such seals are not applicable.

• Magnetic Fluid Seal - utilizes magnetic force, figure 7. This seal, in a bushing-type configuration, consists of a cylindrical permanent magnet which holds a magnetic liquid in the clearance between the magnet and shaft. Magnetic forces hold the liquid in place against the pressure forces. The magnetic liquid is insoluble in the fluids being sealed and acts as a barrier between the various fluids being sealed. This device is commercially available in a bushing-type configuration. Because its sealing ability is sensitive to the radial clearance between the magnet and shaft, the commercial version contains its own shaft and bearings. It has been successfully applied to vacuum applications with virtually no leakage. Testing this seal for marine applications has shown the device to be unsuccessful because of contamination of the magnetic liquid with ferromagnetic contaminants.[12] This device is in a new technology area where rapid development is taking place. A device applicable to marine use may be developed in the future.

• Surface Tension Seal - utilizes surface tension forces. It has been proposed[4] that the fluid forces acting at the interface of two immisible fluids can be used to restrict leakage across a clearance seal in a two-liquid system. However, no device specifically designed to exploit these forces is known.

Hermetic Devices

Hermetic seals provide a positive, leak-tight barrier between the dynamic and stationary elements of the machine.

The shaft motion could be passed through a deformable barrier at the machinery/sea interface. Examples of the deformable-barrier type of seal are the laminated elastomer bearing, figure 7, and harmonic drive.[13]

The laminated bearing consists of thin, alternate layers of metal and elastomer, bonded together. It is attractive for applications with oscillating shafts or shafts with limited rotary motion, since it can serve as both a bearing and seal. It will not permit full rotation, however.

A device like the harmonic drive, which transmits motion across a deformable can-like structure, does permit full rotation. This concept is being evaluated in the electric drive system task.[13]

VARIABLES AND LIMITATIONS COMMON TO CONTACT SEALS

The limitations and variables of seal operation are more readily appreciated if the similarities between a contact-type seal and bearing are recognized. A contact seal can behave as a hydrostatic, hydrodynamic, hybrid, or dry bearing. The essential difference between a contact seal and conventional bearing is one of function; the pressures and clearance generated by the seal are used to restrict leakage, whereas in a bearing they are used to support an external load.

Forces on Seal

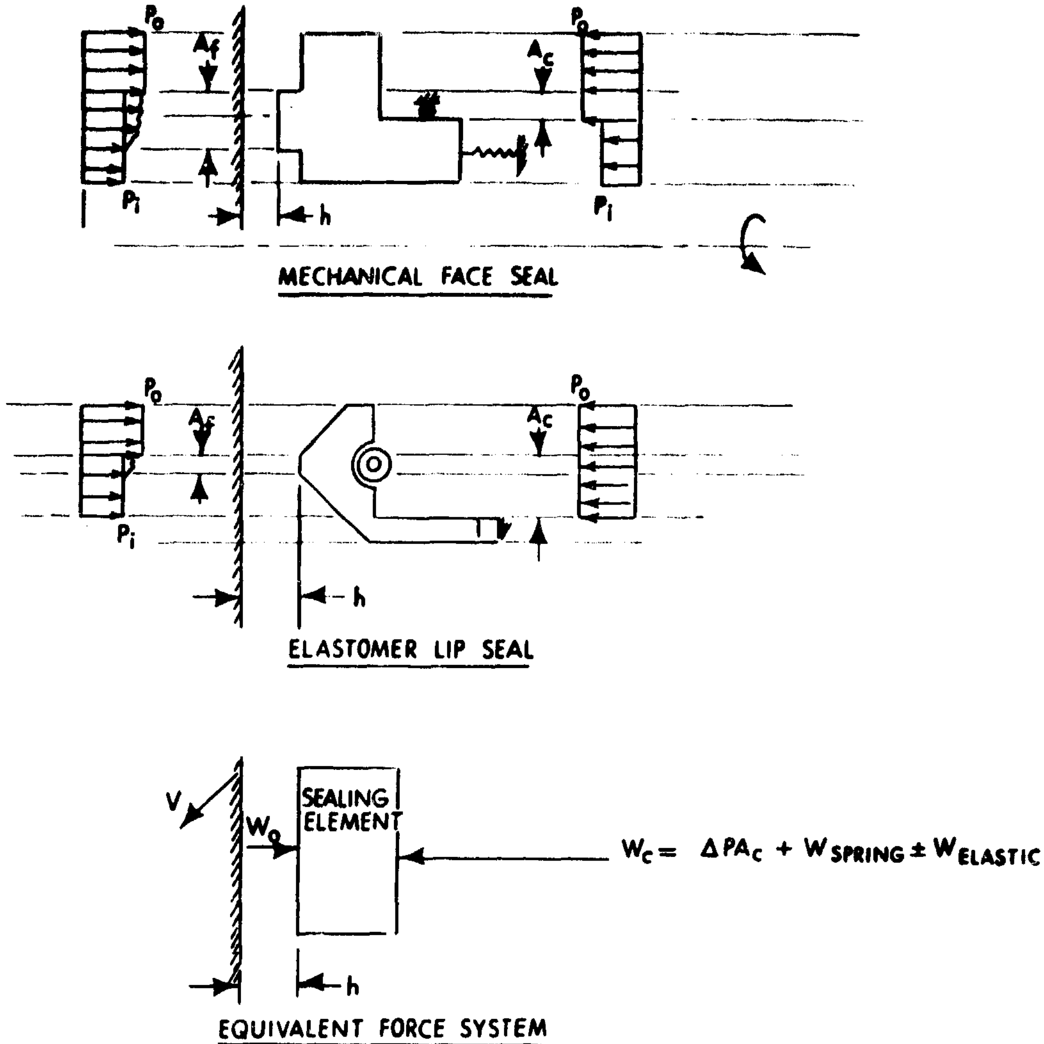
To demonstrate the mechanism of operation of a contact seal and its similarity to a bearing, consider the force system acting on the sealing element of these devices when subjected to a pressure differential as shown in figure 8. The face and lip seal configurations are used as examples. The other configurations shown in figure 6 behave similarly.

The sealing element is free to move relative to the dynamic surface and is loaded in a direction perpendicular to the dynamic surface by a combination of hydrostatic, spring, and elastic forces. The sum of these forces constitutes the seal closing force, W_c, since they are directed to close the seal interface. The closing force is analogous to the externally applied load on a bearing.

Hydrostatic and hydrodynamic forces are generated at the seal interface and tend to open it. The magnitude of these opening forces, W_o, is a function of the interface clearance. The opening force is analogous to the load capacity of a bearing. The force system on the sealing element will then adjust the interface clearance until the seal opening and closing forces are equal, i.e., until equilibrium or balance is obtained. The fluid pressure and clearance existing at the seal interface when equilibrium is established will then determine the leakage of the seal.

Leakage of Seal

The opening force of the seal depends upon the seal interface geometry just as the load capacity of a bearing depends upon the bearing geometry. The difficulty encountered when analyzing or designing a contact seal is defining this geometry. Ideally, the sealing surfaces are perfectly conforming; in reality, they are not. It has been shown that deviations from perfect conformity of the order of microinches have a significant effect on the pressure and clearance generating ability of the seal.



where:

A_c = Net closing area over which the pressure differential acts
A_f = Area of the seal interface
h = Seal interface clearance
ΔP = P_o-P_i; Hydrostatic pressure differential across the seal
W_c = Seal closing force, i.e., force tending to close seal interface
$W_{elastic}$ = Force on sealing element due to secondary sealing member (face seal) or t. Interference fit between lip and shaft (elastomer lip seal).
W_o = Seal opening force, i.e., force tending to open seal interface
W_{spring} = Force on sealing element due to springs
V = Relative sliding velocity at seal interface

Figure 8 - Force System on Sealing Element

Since these deviations are so small, it is difficult to measure and to control or avoid them in any real seal application. Consider now how these deviations can effect the leakage of the seal.

• Leakage Due to Hydrostatic Effects Only. The hydrostatic opening force is associated with deviations in the interface geometry in the direction of the pressure differential.[14] Such deviations can arise from manufacturing errors, mechanical, pressure, and/or thermally induced distortions, and from wear. The hydrostatic opening force, $W_{o(hs)}$, as a function of the interface clearance for a given pressure differential and various interface geometries is shown in item (a) of figure 9. The hydrostatic opening force is a function of the hydrostatic pressure drop across the seal interface. The maximum achievable hydrostatic opening force is, from item (a) of figure 9,

$$W_{o(hs)_{MAX}} = \Delta PA_f \qquad .....(1)$$

The ratio of the actual seal opening force to the maximum hydrostatic opening force when equilibrium is established is

$$\frac{W_o}{W_{o(hs)_{MAX}}} = \frac{W_c}{\Delta PA_f} = \frac{\Delta PA_c + W_{SPRING} \pm W_{ELASTIC}}{\Delta PA_f}$$

$$= \text{EFFECTIVE BALANCE RATIO, EBR} \qquad .....(2)$$

and will be called the effective balance ratio (EBR) of the seal. When the EBR is less than or equal to 0.5, the hydrostatic opening force, $W_{o(hs)}$, will exceed the closing force, opening the seal interface, or an unstable clearance, point (1) in item (a) of figure 9, will be formed. Such a condition is termed an underbalanced or unbalanced seal because stability is never achieved; the seal will either not seal at all or else the leakage will be erratic. For this reason, commercially available seals have EBR's greater than 0.5 when the pressure differential is applied in the direction for which the seal was designed. When the EBR is between 0.5 and 1.0, the seal is termed balanced because the closing force will exceed the hydrostatic opening force, closing the seal interface, point (2) in item (a) of figure 9, or the hydrostatic opening force will equal the closing force with a stable clearance at the seal interface, point (3) in item (a) of figure 9; the seal will then not leak at all or else the leakage will be stable.

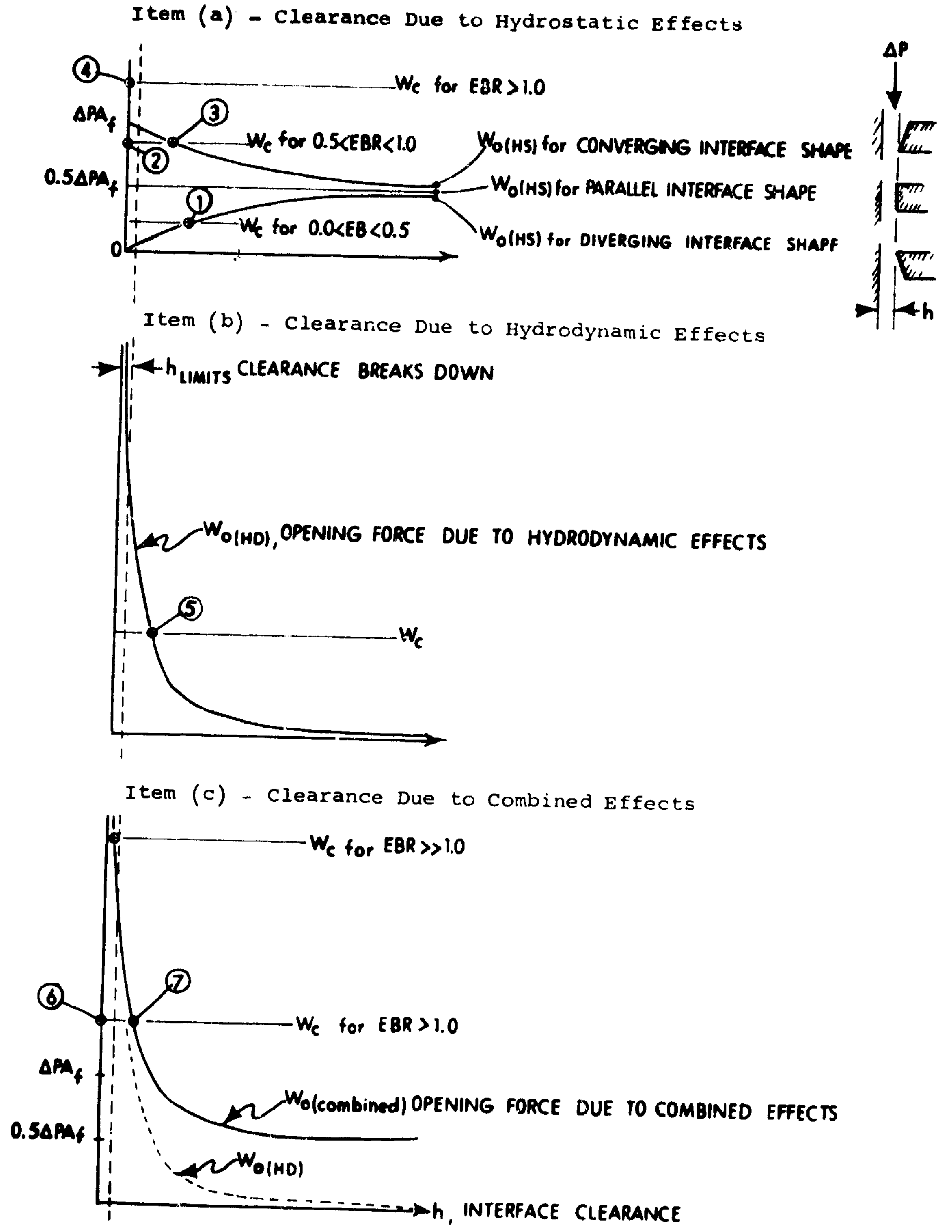


Figure 9 - Forces on Sealing Element
as a Function of the Interface Clearance

The hydrostatic leakage, $Q_{(hs)}$, can be expressed qualitatively as,

$$Q_{(hs)} = \frac{S\Delta Ph_{(3)}^{3}}{\mu} \qquad \text{.....(3)}$$

where S = a function of the seal geometry. The hydrostatic leakage is in the direction of the pressure differential. When the EBR is greater than or equal to 1.0, the seal is termed overbalanced because the seal closing force will always exceed the hydrostatic opening force. No clearance, point (4) in item (a) of figure 9, and thus no leakage is possible.

• Leakage Due to Hydrodynamic Effects Only. Hydrodynamic effects are associated with deviations in the interface geometry in the direction of relative motion. Deviations such as surface waviness, surface roughness, misalignment, eccentricity, and vibration, either singularly or in combination, have been related to hydrodynamic effects.[2, 7] The seal generates pressure at the seal interface when operating hydrodynamically. Pressures, higher than the ambient pressure surrounding the seal, can be generated in regions where unplanned-for deviations produce a converging clearance shape in the direction of relative motion. Subambient pressures can be generated in regions where a divergent clearance shape exists. Subambient pressures can lead to cavitation if the pressure at the seal interface falls to the vapor pressure of the liquid, or to the evolution of gases if the pressure falls to the saturation pressure. Integrating these pressures over the area of the seal interface gives the hydrodynamic opening force, $W_{o(hd)}$, of the seal. It has been found that the hydrodynamic opening force is of the form[1, 2]

$$W_{o(hd)} = B\left(\frac{\mu^{x}V^{y}}{h^{z}}\right) \qquad \text{.....(4)}$$

where

B = some function of the seal and seal interface geometry

μ = viscosity of the fluid at the seal interface

V = relative velocity at the seal interface

h = mean interface clearance

x,y,z = the values of the exponents vary, d[illegible]ding apparently on what is causing hydro[illegible]mic action.

For example, deviations such as surface waviness may give one set of exponents, while another type of deviation, such as vibration, etc, may produce a completely different set of exponents. It has been found that x and y are usually equal and the numerical value varies between 1 and 2. The numerical value of z usually varies between 2 and 6. If there is no pressure differential across the seal, there can be no hydrostatic opening or closing forces, or hydrostatic leakage. The closing force is then due to the spring and elastic forces; the interface clearance when equilibrium is established, point (5) in item (b) of figure 9, is from equation 4,

$$h_{(5)} = \left[\frac{B\mu^x V^y}{W_{o(hd)}}\right]^{1/z} = \left[\frac{B\mu^x V^y}{W_c}\right]^{1/z} = \left[\frac{B\mu^x V^y}{W_{SPRING} \pm W_{ELASTIC}}\right]^{1/z} \qquad \text{.....(5)}$$

The seal can leak even though there is no applied hydrostatic pressure differential because it is generating its own pressures. Fluid can be pumped out of the clearance to either side of the seal, in regions of high pressure, and into the clearance from either side of the seal, in regions of subambient pressure. This pumping phenomena is not yet completely understood, particularly when different liquids are present on both sides of the seal. It appears however, that pumping in a seal is similar to end-leakage effects in conventional hydrodynamic bearings[5, 15] and is of the form,

$$Q_{(hd)} = DVh_{(5)} \qquad \text{.....(6)}$$

where D is some function of the geometry and may be positive or negative, i.e., the direction of the leakage can vary. Substituting in equation (6) for the clearance, equation (5),

$$Q_{(hd)} = DV\left(\frac{B\mu^x V^y}{W_c}\right)^{1/z} \qquad \text{.....(7)}$$

shows that the pumping leakage is a function of the interface geometry, viscosity at the seal interface, relative velocity at the seal interface, and seal closing force.

• **Leakage Due to Combined Effects.** In the more general case, the seal will be subjected to both a pressure differential and relative motion; the opening force, clearance, and leakage will be developed by both hydrostatic and hydrodynamic effects. Notice in equation (2) that the EBR is a function of the pressure differential. Seals in deep-submergence applications are subjected to very low-pressure differentials, 5 psid or less. At these pressure differentials, the contact seal configurations shown in figure 6 will have EBR's greater than 1.0. As was shown earlier, the hydrostatic opening force is not sufficient in itself to balance the closing force when the EBR is greater than 1.0. Equilibrium under static conditions will then be established when there is no clearance, point (6) in item (c) of figure 9. There will then be no static leakage. Under dynamic conditions, however, hydrodynamic effects, combined with hydrostatic effects, can provide sufficient opening force to balance the closing force. The seal can then operate with a clearance, point (7) in item (c) of figure 9, and can leak. The dynamic leakage will be due to both hydrostatic and hydrodynamic effects[5, 15] and will be of the form,

$$Q_{(c)} = Q_{(hs)} + Q_{(hd)} \qquad \text{.....(8)}$$

$$Q_{(c)} = \frac{S\Delta P h_{(7)}^{3}}{\mu} + DVh_{(7)} \; . \qquad \text{.....(9)}$$

The torque absorbed by the seal, the coefficient of friction, and power loss of the seal are of the form

$$T = \frac{E\mu V}{h_{(7)}} \qquad \text{.....(10)}$$

$$f = \frac{E'\mu V}{W_c h_{(7)}} \qquad \text{.....(11)}$$

$$\text{power} = \frac{E''\mu V^2}{h_{(7)}} \qquad \text{.....(12)}$$

where E, E', and E" are some functions of the seal geometry.

• Design Variables. At low or zero pressure differentials, the operating clearance of the seal is determined primarily by the hydrodynamic opening force, i.e.,

$$h_{(7)} = h_{(5)} = \left(\frac{B\mu^x V^y}{W_c}\right)^{1/z} \quad .....(13)$$

Substituting for $h_{(7)}$ in equations (9) through (12) gives

$$Q_{(c)} = \frac{S\Delta P}{\mu}\left(\frac{B\mu^x V^y}{W_c}\right)^{3/z} + DV\cdot\left(\frac{B\mu^x V^y}{W_c}\right)^{1/z} \quad .....(14)$$

$$T = \frac{E\mu V}{\left(\frac{B\mu^x V^y}{W_c}\right)^{1/z}} \quad .....(15)$$

$$f = \frac{E'\mu V}{W_c\left(\frac{B\mu^x V^y}{W_c}\right)^{1/z}} \quad .....(16)$$

$$\text{power} = \frac{E''\mu V^2}{\left(\frac{B\mu^x V^y}{W_c}\right)^{1/z}} \quad .....(17$$

Consider now the various parameters in equations (14) through (17). The seal geometry parameters, S, B, D, and E, are functions of unplanned for or accidental features, such as waviness, misalignment, etc, and are largely beyond the control of the seal designer. The viscosity of the fluid at the seal interface, μ , will be dictated by the type fluid on either side of the seal, the system pressure, and system temperature. The relative velocity, V, will be proportional to the shaft speed and shaft diameter of the machinery. Thus, only W_c, the seal closing force, and ΔP, the pressure differential, remain as seal design variables.

• Seal Design Alternatives. The variables μ, V, and W_c are usually combined with b, the seal interface width, to form a dimensionless group called the duty parameter,[2] G, where

$$G = \frac{\mu V b}{W_c} . \qquad .....(18)$$

The duty parameter is analogous to the group, $zn/\bar{P}$, commonly used in conventional bearings. It has been shown[2] that when the coefficient of friction of a face seal is plotted against the duty parameter, figure 10, a curve similar to the friction versus $zn/\bar{P}$ curve of a conventional hydrodynamic bearing is obtained. This illustrates that a seal, like a bearing, can operate under different regimes of lubrication. When operating values of G are to the right of the minimum point, G_{min}, the seal is operating under full-film hydrodynamic conditions and the sealing surfaces are completely separated by a clearance. This clearance breaks down, however, when operating values of G fall to the left of G_{min} and the seal operates under mixed film conditions. The seal will run under contact or essentially dry conditions if operating values of G are reduced further. Operating values of G to the left of G_{min} result in increased friction and wear rates, figure 10.

• Seal Operates Hydrodynamically. One approach to seal design is to design for maximum life (minimum wear) and minimum losses at a sacrifice in leakage. This could be accomplished by selecting the seal-closing force so values of G during the majority of operation fall about G_{min} or slightly to the right of G_{min}, i.e., W_c is less than $\mu Vb/G_{min}$. The seal would operate most of the time with minimum clearance, which from equation 9 implies minimum leakage, minimum wear, and minimum coefficient of friction. Notice, however, that during start-up or shutdown, operating values of G will be less than G_{min} and the seal surfaces will come into contact. Thus, this approach is most readily applied to constant speed applications, where operating values of G are limited and where start-up and shutdown occur rapidly and infrequently. The seal can pump when operating hydrodynamically, as was noted earlier. From equation (9) and (14) it can be seen that putting a pressure differential across the seal is no guarantee that the leakage will be biased in the direction of the differential; the hydrodynamic pumping leakage may add to or hinder the hydrostatic leakage, or it may completely reverse the direction of the leakage. Notice also that zero pressure differential does not guarantee zero leakage, again because of the pumping leakage. It has been speculated that it may be possible to have the hydrostatic leakage "buck" the hydrodynamic leakage. To do this, as can be seen from equation (14), the pressure differential would have to change with μ, V, and W_c.

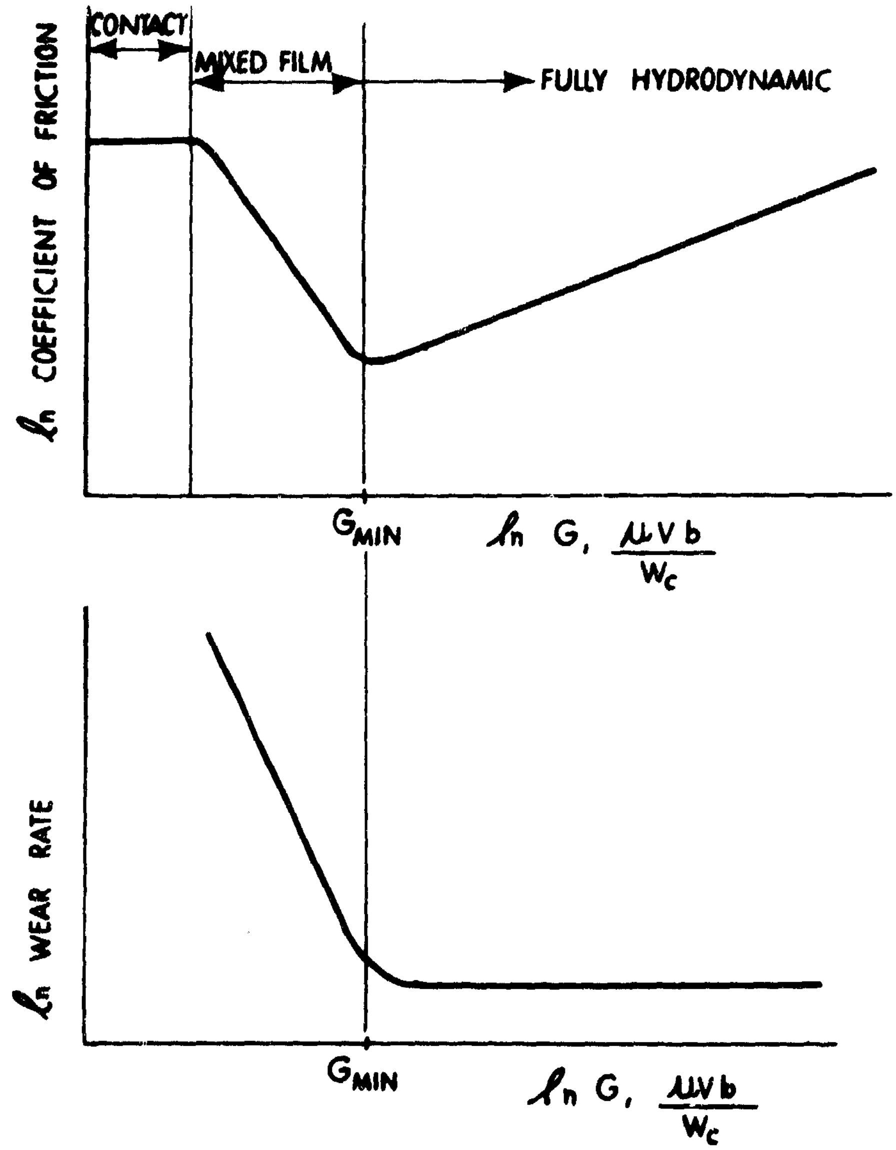


Figure 10 - Typical Performance Trends of a Contact Seal

● **Seal Operates under Mixed Film or Contact Conditions.** An alternate approach, which appears more suitable for the majority of deep submergence applications, is to try to eliminate leakage, by eliminating the clearance and sacrifice seal life. This could be accomplished by selecting the closing force so all operating values of G fall to the left of G_{min}, i.e., W_c is greater than $\mu Vb/G_{min}$. This approach requires careful selection of seal interface materials to minimize friction and wear, or risk destruction of the seal interface. The $\overline{P}V$ (rubbing factor)[15] criterion is commonly used to establish operating limits when this approach is used. The $\overline{P}V$ of the seal is the product of the unit load at the seal interface, $\overline{P}$ (not the applied pressure differential), expressed in pounds per square inch and the relative sliding velocity at the seal interface, V, expressed in feet per minute. Each combination of seal surface materials has a maximum acceptable pressure and velocity it can withstand without exceeding some allowable temperature, usually the temperature limit of one of the materials, or allowable wear rate. The limiting $\overline{P}V$ varies with the ambient temperature, the extent of lubricant at the interface, and the ability of the seal configuration to transfer heat away from the rubbing interface. For example, in a reciprocating seal configuration, a given material combination can usually operate at a higher $\overline{P}V$ than it can in a rotating configuration, because of the better heat transfer from the interface with a reciprocating configuration. If the $\overline{P}V$ of the seal is less than the maximum acceptable $\overline{P}V$ of the materials, it is assumed that the allowable temperature or wear rate will not be exceeded in operation and the materials will perform satisfactorily.

When the sealing surfaces are in true contact over the entire area of the seal interface, the $\overline{P}V$ and EBR of the seal are related by

$$\overline{P}V = \Delta P \times EBR \times V = \frac{W_c}{A_f} \times V \; . \qquad .....(19)$$

The magnitude of the pressure differential, i.e. whether zero or of the order of 5 psid, should have no effect on the leakage when the seal operates in contact. The problem then, for either design approach, is to select the proper seal closing force. Since there is no way to predict the value of G_{min}, the closing force must be determined by trial. The closing force in low differential pressure applications is determined primarily by the spring and elastic forces. It is usually a simple matter to change these forces on commercially available seals by changing the spring, or the working height of the spring, or by changing the squeeze or stretch on the elastic member. The seal manufacturer on the basis of his own experience, will usually have a good feel for the magnitude of closing force necessary. The seal user should draw from the manufacturer's experience.

Seal System Design Alternatives

Abrasive contaminant in the sealed fluids have an adverse effect on all contact seal configurations. Contaminants can physically foul the seal by inhibiting the motion of the sealing element. Abrasives can also get into the seal interface, causing rapid wear, short life, and leakage.

When abrasive conditions are severe, some secondary means for protecting the seal from abrasives is usually provided for in the seal system design. Since there is a high probability of having heavy concentrations of abrasives in the seawater near the bottom, some scheme for protecting the seal should be considered. Commonly used schemes are:

- Slingers or close fitting bushings. Slingers or close fitting bushings, figures 1 and 2, are sometimes used to keep contaminants from the seal. They are effective at keeping relatively large particles from the seal. It is questionable, however, if these devices keep the small particles (0 to 5 microns), which can get into the seal interface and cause abrasive wear, away from the seal.

- Multiple seal arrangement. Arrangements using two or more of the same, or two different seal configurations, are commonly used in industrial applications. The dual mechanical face seal arrangements used on the Annapolis electric drives, such as in figures 1 and 2, appear to be adaptations of commonly used industrial schemes. In most industrial applications, the pressure in the seal cavity is usually higher than on either side of the seals. In the deep-ocean adaptation, however, there is usually no pressure drop across either seal (a redundant arrangement) or a cascaded pressure drop from the machinery compartment to the sea (a cascaded arrangement). This difference in the way the seals are pressurized can cause problems, as will be considered when design features of the face seal are discussed. The rationale of the multiple arrangement is that the outboard seal, that seal between the seal cavity and the sea, functions in effect as a filter; only those particles which are smaller or of the same size as the interface clearance can enter the seal cavity and reach the inboard seal. The outboard seal restricts leakage from the real cavity and protects the inboard seal from abrasives, but the outboard seal becomes worn and is subjected to possible fouling in the process. It is usually thought that putting a small differential from the seal cavity to the sea will promote hydrostatic leakage and thus flush abrasives out of the outboard seal interface. This is not necessarily true, because of the balance of the seal and pumping effects. A dual seal arrangement, as used on the Annapolis motors, should also reduce the rate of sea-water contamination of the machinery compartment because of the longer leakage path between the sea and machinery. It has been suggested that a nonemulsifing fluid be used in the seal cavity; in this way, any sea-water leakage across the outboard seal can separate to the bottom of the seal cavity where it can be periodically drained. The dual seal arrangement, however, adds complexity to the system, (two seals and a compensator are needed for each shaft penetration) takes up more space, and results in more losses. Because of this, it appears impractical for applications with multiple shaft penetrations. It has been suggested that the outboard seal be replaced with a felt seal, figure 11. The rationale here is that the felt is porous enough to let seawater into and thus compensate the area between the seals. The felt is also an effective filter medium and will remove particles from the water as it passes through. The porous texture of the felt allows the particles to be imbedded in the felt without damaging the mating surface. This would simplify the seal system, since no compensator is needed for the area between the seals. It may be applicable to multishaft applications. Since the felt absorbs the contaminants, it would have to be replaced periodically.

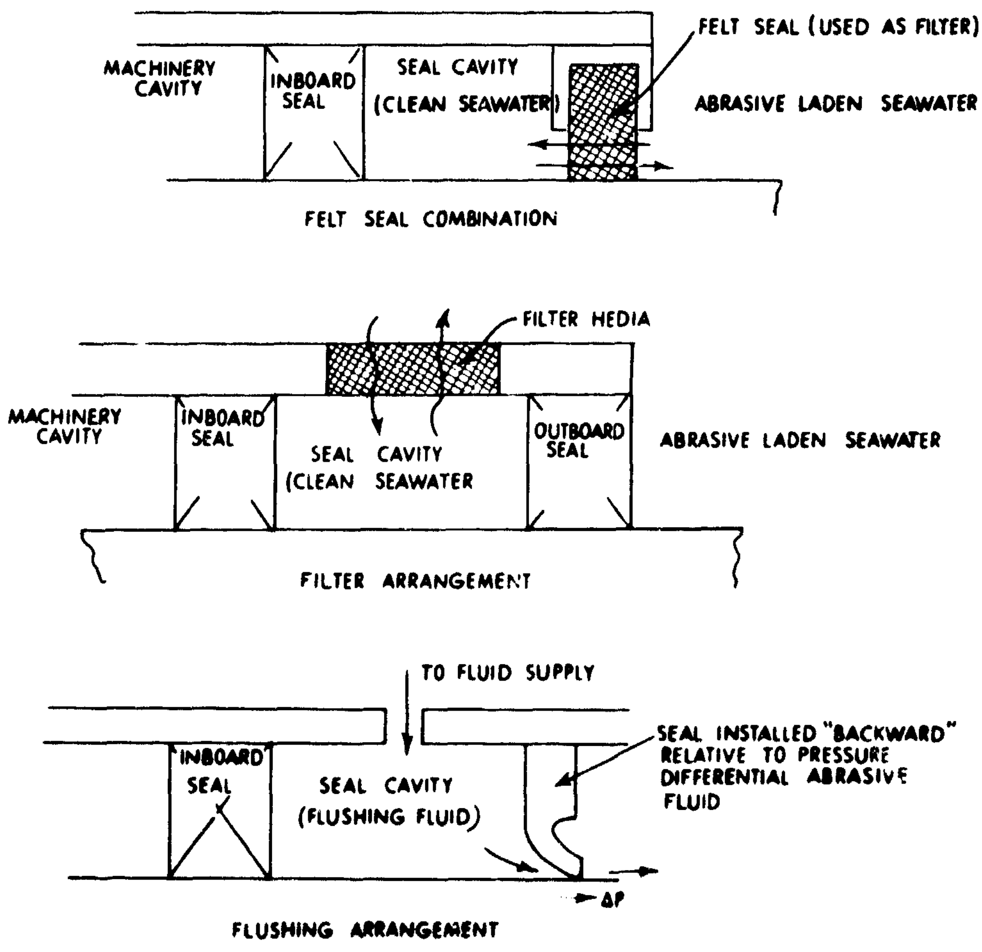


Figure 11 - Possible Arrangements for Abrasive Protection

• **Filters.** The seal cavity of a multiple seal arrangement can be vented to the sea through a filter medium, figure 11, similar to the filter arrangement used on some of the free-flooded machinery. Since the filter allows free passage of fluid, the seal chamber is pressure compensated and the seawater is filtered as it enters the seal compartment. The inboard seal is then protected from abrasives; the outboard seal would have to keep the clean water in the cavity and keep dirty seawater from entering the cavity.

• Multiple seal flushing arrangement. In some industrial applications, the seal facing the abrasive fluid will purposely be installed backwards, relative to the pressure differential, as shown in the example in figure 11. The pressure in the seal cavity is maintained high enough so that it forces the seal open (seal is underbalanced) and causes it to leak. In this way, clean fluid is continuously flushed through the seal cavity and out the outboard seal. This arrangement requires a continuous supply of flushing fluid. Since the supply of fluid on a deep-ocean application is usually limited by a desire to keep the compensator small, this arrangement is not applicable.

In summary, all contact-type seal configurations are conceptually the same; they are in effect bearings whose generated pressure and clearance is utilized to restrict leakage. Design of these seals is a tradeoff between wear and leakage.

CHAPTER III

SELECTION FACTORS

DIFFERENCES

The previous discussion considered the common features of the various contact seal configurations. Consider now the basic differences; the choice of one configuration over the other will usually be based on one or a combination of the following differences.

Mounting Convenience, Space Requirements

Contact seals are available in axial and radial configurations as shown in figure 6. The mechanical end-face seal is the most common example of the axial configurations wherein the seal closing force is parallel to and the leakage path is perpendicular to the axes of rotation. In a radial configuration, such as the elastomer lip seal, the seal closing force is perpendicular to and the leakage path is parallel to the axis of rotation.

The axial configuration usually requires some modification of the shaft, such as steps, grooves, or mounting sleeves, to mount and drive the mating face. It will not wear the shaft or require close tolerances or a fine surface finish on the shaft.

The radial configuration can usually use the shaft surface as a mating face but requires close tolerances and a fine plunge-ground surface finish on the shaft. It will wear the shaft and may require a hardened, replaceable wear sleeve, or shaft surface treatment, or some way to shift the seal axially to provide a fresh running surface on the shaft when the seal is replaced.

The face seal will generally require the most space, while the conventional O-ring, slipper seal, and spring loaded U-cups require the least space. The face seal will usually take up approximately the same or slightly greater radial space, but more axial space than the elastomer lip seal for shaft sizes up to about 4 inches in diameter.

Features of the shaft over which the seal must be assembled should be considered. Keyways, splines, threads or other shaft discontinuities may cause assembly problems with radial configurations and may require special assembly techniques to avoid damage to the seal.

Directional Sealing Ability

A unidirectional configuration is designed to seal only when the pressure differential across the device is in one specific direction. Sealing in a bidirectional configuration is independent of the direction of the pressure differential.

Configurations such as the conventional O-ring and O-ring-slipper seal are inherently bidirectional. The conventional elastomer lip seal, spring loaded U-cup and packing are inherently unidirectional configurations. A conventional face seal will also usually be unidirectional, but it can be designed to be bidirectional.

Care should be excerised to ensure that the seal is so mounted in the system that, if a pressure differential is applied to the seal, it is in the direction for which the seal was designed. If bidirectional pressures are anticipated, such as would occur if a vacuum filling procedure is used, consideration should be given to a bidirectional configuration.

Availability and Cost

Face seals are generally available in nonsplit configurations for shaft sizes from 0.125 to 4 inches in diameter. Split configurations will have to be designed for the specific application, will be more costly, and will require more care at assembly.

Elastomer lip seals, O-rings, O-ring-slipper, and spring loaded U-cup configurations are generally available for shaft sizes from 0.125 to 12 to 15 inches in diameter. Lip seals and O-rings can be split; the split is vulcanized or chemically bonded on assembly. Slipper and U-cup configurations are usually not split because they are usually made of Teflon which is difficult to bond.

Face seals will generally be more expensive than the other configurations. The costs of lip, slipper, O-ring, and U-cups are comparable.

The seal manufacturer should be consulted before the shaft, mounting, and space envelope dimensions are fixed to ensure that a seal for these dimension is available. Nonstandard sizes cost more and have longer delivery times. There are usually enough design/material options available for a standard size seal to make it adaptable to the particular requirements of the application.

Performance Limits

- Physical limits. Consideration should be given to the range of shaft motion, i.e., radial and axial runout, eccentricity, misalignment, and vibration the seal can withstand without suffering mechanical damage or without mechanical separation of the seal interface. An axial configuration can generally tolerate more radial shaft motion than a radial configuration; conversely, a radial configuration can generally accommodate more axial shaft-to-seal motion than the axial seal. The axial configuration requires close control of the axial shaft-to-mounting dimension so that the proper spring load is obtained. Conversely, the radial device requires close control of the radial shaft-to-mounting bore dimensions. However, it appears that unplanned-for shaft motions considerably smaller than the physical limits of the seal can affect the leakage of the seal. In this respect, radial and axial configurations have similar limits.

• $\bar{P}V$ Limits. As was noted earlier, it appears that pumping leakage can be minimized by designing the seal so that it operates under mixed film or contact conditions. Its ability to operate under these conditions depends primarily on the interface materials, particularly on the allowable $\bar{P}V$ of the interface combination. Configurations such as the face seal, slipper seal, and U-cups, utilize low-friction, low-wear, self-lubricating materials and can operate under these conditions. Configurations such as the conventional elastomer O-ring and elastomer lip seal, however, have a seal surface with inherently poor "dry" frictional properties; lubrication is essential if they are to operate successfully. Generally, the face seal will have the higher allowable $\bar{P}V$ among the various configurations. Specific operating limits for each configuration in the intended application should be obtained from the seal manufacturer.

• Other. Additional considerations are the sensitivity to contaminants in the sealed fluids, sensitivity to installation errors, and material compatibility with the sealed fluids and temperatures. These differences are due mainly to specific design features or options of the various configurations and will be considered in the next section.

DESIGN FEATURES AND APPLICATION CONSIDERATIONS

The seal user can select those design features and materials best meeting his application requirements from the manufacturer's literature. The choice of particular design features and materials can be left to the seal manufacturer, with the user specifying the application and performance requirements. The latter approach is recommended because of the diversity of design features and materials available and because a great deal of seal design and material selection is based on experience. Most seal manufacturers have an engineering consulting service and will supply recommendations on request. Most prefer to be consulted, particularly for unusual applications, and some manufacturers will purposely make their design literature vague or incomplete so that they must be consulted for full details. A list of seal manufacturers appear in Machine Design.[10]

Face Seal

The following discussion describes basic design features of the candidate contact seal configurations and points out some areas where application problems can occur.

The basic parts of a mechanical face seal[16] are shown in figures 12 and 13. The sealing element consists of a primary seal ring carrier, a primary seal ring, a secondary seal, springs and an antirotation device to prevent rotation or to drive the primary seal ring carrier. The primary seal ring has a raised, finely lapped annular surface, which is termed the seal face.

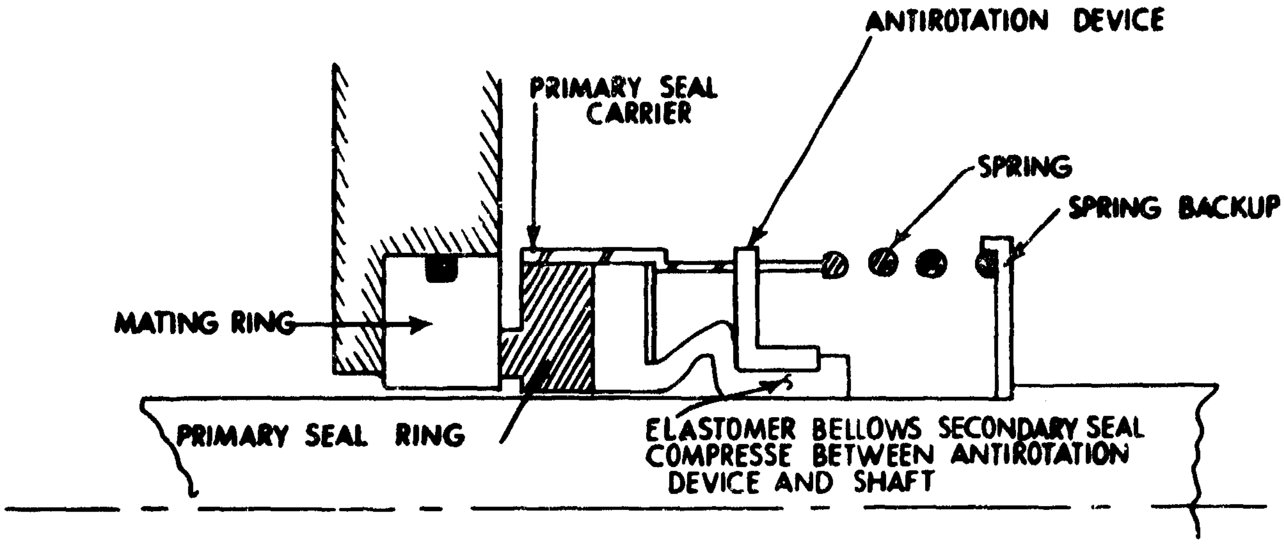


Figure 12 - Shaft-Mounted Face Seal
with Elastomer Bellows Secondary Seal

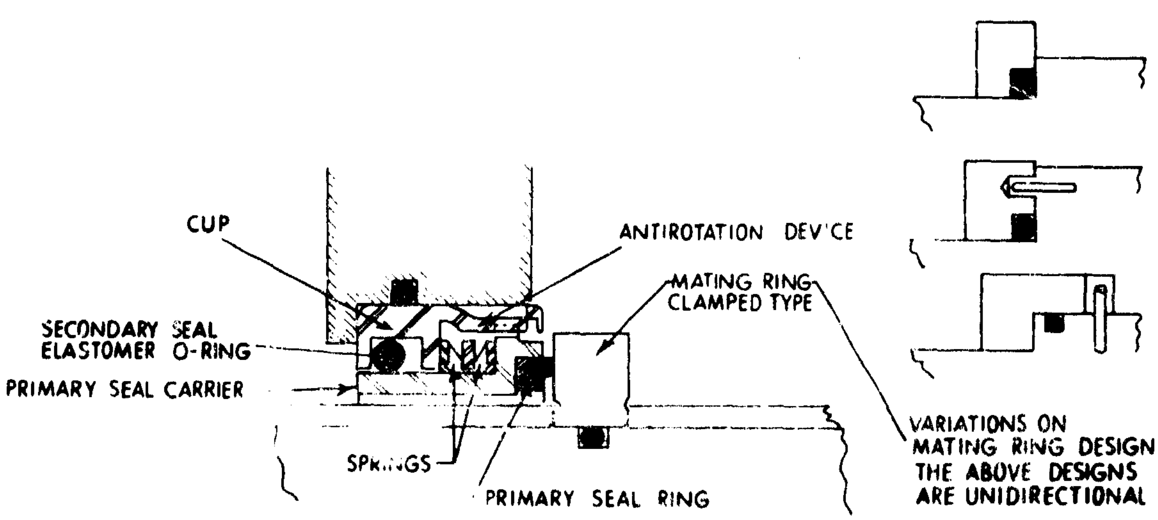


Figure 13 - Cup- or Package-Type
Mechanical Face Seal, Stationary Mounted

A mating ring provides a lapped mating face or surface. Primary sealing takes place at the interface of the seal and mating face. The secondary seal permits axial motion of the primary seal carrier and allows it to line up with the mating face, while providing a static seal between the housing and seal carrier. The sealing element and mating ring are usually supplied as a set by the seal manufacturer.

Commercial versions of face seals differ in:

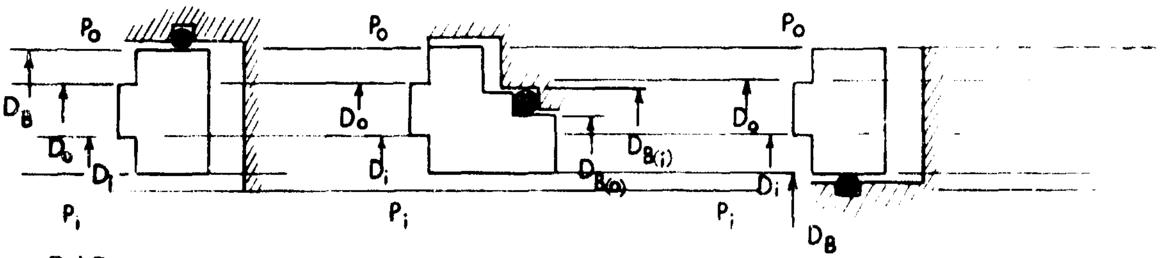
- The magnitude of the hydraulic balance ratio. The effective balance ratio, equation (2), can be expressed as

$$EBR = \frac{A_c}{A_f} + \frac{W_{SPRING} \pm W_{ELASTIC}}{\Delta P A_f} \quad . \qquad .....(20)$$

The first term[16] in this expression, A_c/A_f, is referred to as the hydrostatic or hydraulic balance ratio since it represents the ratio of the hydrostatic closing force to the maximum achievable hydrostatic opening force. Face seals are usually referred to in terms of percent of hydraulic balance. Hydraulically balanced seals, $50\% A_f < A_c < 100\% A_f$, are used in applications where differential pressure is high, usually over 100 psid. Hydraulically overbalanced seals, $A_c \geq 100\% A_f$, are usually used for low differential pressures. The reason for this can be seen from equation (19); for a given pressure differential, P, and sliding velocity, V, a balanced seal will have a lower $\bar{P}V$ than an overbalanced seal. Thus, manipulating the balance ratio is one way used to extend the $\bar{P}V$ operating limits of the seal. The hydrostatic forces are small at the low-pressure differentials used in deep-submergence applications; the term $W_{SPRING} \pm W_{ELASTIC}/\Delta P A_f$ will usually be much greater than A_c/A_f. Thus, the magnitude of the effective balance ratio is primarily determined by the spring and elastic forces and will be greater than 1.0; i.e., the seal will be overbalanced, regardless of whether the seal is hydraulically balanced or hydraulically overbalanced. An effective balance ratio greater than 100% is desirable since it should inhibit the static leakage of the seal as was discussed earlier.

- Directional sealing ability. Most face seals are <u>unidirectional</u>. The direction in which they seal is determined by the effective balance ratio, the type of secondary seal, and the way the seal is mounted. For example, a seal designed to be overbalanced when the differential is in one direction can become underbalanced, and the seal interface will open if the differential is reversed, figure 14. Reversed pressure may also unseat the secondary seal or unseat the sealing element and or mating ring from its mounting. It has been noted that in most of the Annapolis electric drive systems, which use two unidirectional face seals in a pressure cascaded arrangement, one of the seals is usually pressurized in the reverse direction, figure 15. Most face seals will usually accommodate a small reversed differential without harm; the magnitude of the reversed differential that can be tolerated depends on the spring load, and thus the setup working height of the spring, and particular features of the secondary seal and mounting. If, however, the reversed pressure gets higher than the seal can accommodate, because of initially presetting the compensator for too high a differential, or because of a poorly designed compensator or malfunction thereof, or from pressure or vacuum filling of the system, the seal will be forced open and leak; in some configurations mechanical damage can result.

P_O = Pressure at OD
P_i = Pressure at ID
D_O = Outside diameter of seal face
D_i = Inside diameter of seal face
D_B = Balance diameter; effective diameter of secondary seal



$D_B > D_O$
Unidirectional
Hydraulically underbalanced when $P_O > P_i$
Hydraulically overbalanced when $P_i > P_O$

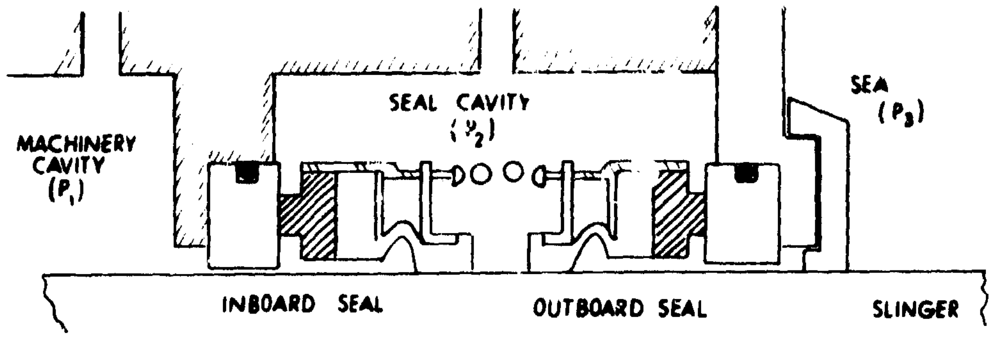
$D_i < D_B < D_O$
Bidirectional
Hydraulically balanced when $P_O > P_i$ and when $P_i > P_O$

$D_B < D_i$
Unidirectional
Hydraulically overbalanced when $P_O > P_i$
Hydraulically underbalanced when $P_i > P_O$

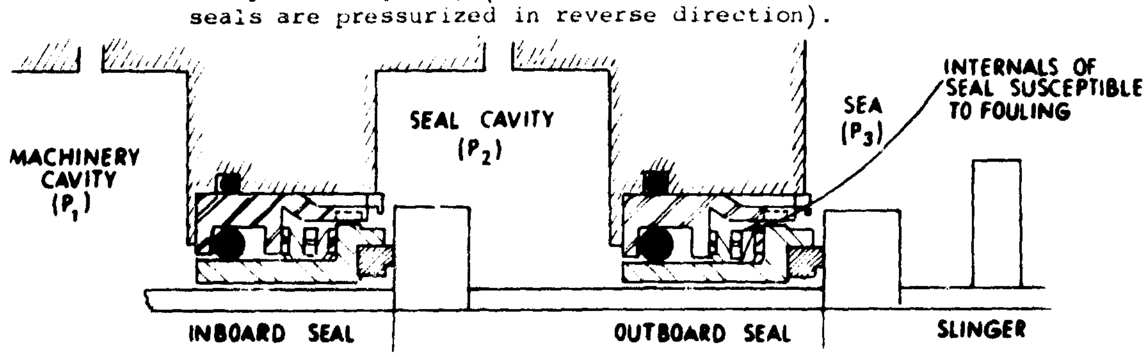
Figure 14 - Directional Sealing Ability of Face Seal

Thus, mount the seal so that the pressure differential is in the correct direction and aids the spring and elastic forces in closing the seal. If bidirectional pressures are anticipated, such as would occur in vacuum filling, consideration should be given to a face seal of bidirectional design. A bidirectional design is usually a little more costly but should result in increased reliability. Additionally, consideration should be given to ensure that the sealing element and mating ring are positively located in their respective mounting; it is preferable to mount them so that the pressure tends to hold the seal in place.

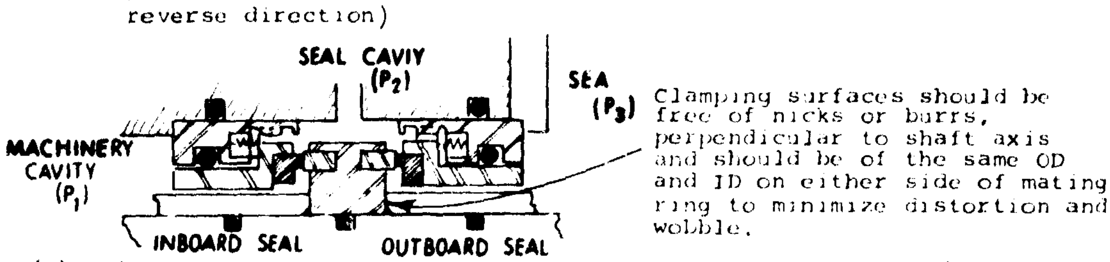
Item (a) - Arrangement using shaft mounted seals with common spring (if the pressure is cascaded i.e., $P_1 > P_2 > P_3$, inboard seal is pressurized in reverse direction).



Item (b) - Arrangement using stationary mounted seals with clamped mating ring (if the pressure is cascaded, both seals are pressurized in reverse direction).



Item (c) - Arrangement using common, clamped mating ring (if the pressure is cascaded, inboard seal is pressurized in reverse direction)



Item (d) - Arrangement using different size seals on stepped shaft (if the pressure is cascaded, both seals pressurized in designed for direction).

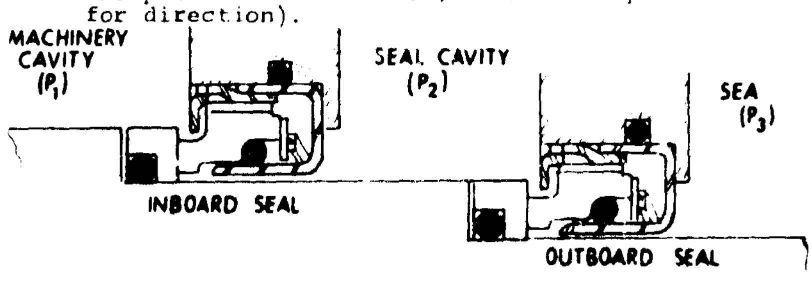


Figure 15 - Examples of Double Face Seal Arrangements in Use

• Secondary seal. An elastomer diaphragm, elastomer O-ring, or other types of pressure-actuated packing are commonly used as a secondary seal. The important consideration here is the compatibility of the secondary seal material with the fluids (and with fluids which may be used for cleaning, flushing or storage) and temperatures on either side of the seal. Swelling or embrittlement of the secondary seal can introduce secondary leakage paths or inhibit the motion of the primary seal carrier. Additional considerations are its flexibility, sensitivity to fouling, and sensitivity to the direction of the pressure differential. The temperature limit of a face seal is usually dictated by the secondary seal material. Face seals with a metallic bellows or piston-ring-type secondary seal are available for applications which are beyond the temperature or compatibility limits of elastomers. They are more expensive.

• Type and number of springs. The springs, particularly the spring load and spring rate, are very important at low pressures, since they apply the seal closing force. Seal designs using a single helical spring, multiple helical springs, wave springs, or a metallic bellows are available. The single helical spring design requires more axial space but has a lower spring rate and longer axial working range. Multiple springs, wave springs, and metallic bellows designs require less space but have higher spring rates and limited axial working range. The axial stack-up tolerances between the shaft and mounting must be controlled carefully when installing the seal so that the proper working height is obtained on the springs. Excessive axial tolerances can result in insufficient or no spring load and leakage. Insufficient axial tolerance can result in too high a spring load or bottoming of the springs. This will result in excessive wear or possible catastrophic mechanical failure of the seal. The closing force on a seal with a high spring rate will be more sensitive to variations in the working height of the spring than one with a low spring rate, and thus will require more care when assembling. Axial shaft play must be eliminated when assembling a face seal to ensure that the working height of the spring in operation is the same as when set up. Axial loads must be kept off the seal to prevent bottoming of the springs and overloading of the seal faces. Conversely, the seal will put an axial load on the shaft. The possibility of this loading affecting the machinery, particularly unloading the bearings, should be considered.

• Mounting arrangement, type construction. The sealing element can be mounted on the shaft, for a rotating seal, or in the housing, for a stationary seal. The basic construction features of the two types of mountings are usually as shown in figures 12 and 13. Both types have been used in deep-ocean applications.

• Shaft-mounted seal. The shaft-mounted seal commonly uses an elastomer bellows type of secondary seal and a single helical spring. The various pieces of the seal are separable and are held together, sandwichlike, in operation by the spring and hydrostatic forces.

1. Advantages. Major advantages claimed for this arrangement are:

a. Mounting convenience - requires little or no modification of the shaft. All that is usually needed is a collar or snap ring groove to back up the spring. Drive is usually by friction between the elastomer and shaft.

b. Since the parts are separable, a damaged, worn, or fouled part can be replaced or cleaned without replacing the whole seal.

c. Less sensitive to fouling because of its open construction.

d. Relatively long axial working range.

2. Limitations. Major limitations are its:

a. Requirement for considerable space.

b. Sensitivity to installation errors. The elastomer bellows is usually compressed between the shaft and primary seal carrier, to provide a static seal and a friction drive. The elastomer may adhere to the shaft prematurely, so tightly during assembly, that a large axial force may be necessary to move it into its proper axial position. This force must be transmitted to the elastomer bellows through the primary seal ring, which is usually carbon, by drawing up on the mating ring mounting flange, figure 12. The person assembling the seal will usually not realize how great a force he is applying to the primary carbon seal ring. The result is usually immediate breakage of the carbon, or excessively high face loads which eventually lead to failure. Care must also be taken in assembly and in operation (reversed pressure) to ensure that the primary carbon seal ring is not pulled out of position relative to the primary seal ring carrier. This can also result in carbon breakage. Too smooth or too small a shaft may result in slippage and leakage at the bellows shaft interface.

• Stationary-mounted seal. The stationary-mounted seal is usually of the "packaged" or capsule type of construction, figure 13. In this type of construction, the primary seal ring, primary seal ring carrier, springs, secondary seal, and antirotation device are contained in a cup or capsule. The capsule is pre-assembled by the seal manufacturer and is handled and installed as one unit.

1. **Advantages.** Major advantages of this arrangement are:

a. Compact size

b. Capsule, one-piece type of construction simplifies assembly and makes assembly less prone to error.

c. Versatility. The capsule type of seal is available with elastomer O-rings, metallic bellows, or other types of pressure actuated packings as secondary seals, wave springs, multiple helical coil springs, in hydraulic balanced or hydraulically overbalanced designs, and in unidirectional or bidirectional designs. ASLE Standard 68-5(T)[17] provides mounting and space envelope dimensions for these seals.

2. Limitations. Its major limitations are:

a. It will usually require modification, such as sleeves, steps, etc, to the shaft for mounting the mating ring. The more popular mating ring types and mountings are shown in figure 13. Dimensional information for these mating rings is given in ASLE Standard 68-5(T).[17]

b. It has relatively limited axial working range.

c. The closed-type construction makes the springs and secondary seal sensitive to fouling.

d. In most designs, figure 13, the cup is upset so that the internal parts cannot be removed without destroying the seal. Thus, if any part is worn or damaged or the seal becomes fouled, the entire seal must be replaced. However, variations of the cup design using snap rings or a similar device to hold the parts in the capsule, but still allow disassembly for replacement of the spring or secondary seal, are available.

• General Considerations. Consideration should also be given to the way the seal ring and mating ring are restrained from rotating and/or driven. This is particularly important in rotary applications with high stop-start rates or oscillating shafts. The starting torque of a face seal is greater than the running torque; thus, a relatively large reaction is taken at the drive or restraining device while starting. The torque reaction can sometimes cock the primary seal ring carrier or inhibit its axial motion. A condition termed bayonetting can sometimes occur if the restraining device is sharp or of such small area that high contact stresses occur; under these conditions, the restraining device can wear itself into and interlock with the primary seal carrier and inhibit its motion. The mating ring of the rotary seal and the cup of a stationary seal are usually kept from rotating by a press fit into a bore, by an O-ring about their outside diameter, or by a clamping arrangement, figure 15.

Clamping and press fits require care, since they can result in distortion or possible damage of the sealing surfaces. One or more O-rings around the outer diameter usually provide enough friction to keep the mating ring or cup from rotating. Consideration should be given to the relative rates of thermal expansion of the mounting bore and seal to ensure that enough interference exists over the range of operating temperatures to effect a static seal and provide sufficient friction force. Additional considerations regarding mounting:

1. Mount the seal so that the springs and secondary seal are in the cleaner, less corrosive fluid. This will minimize chances of fouling and corrosion on the springs and secondary seal. The mounting must also be consistent with the direction of the pressure differential.

2. The simpler part of the seal should be rotating to minimize churning losses and simplify balancing of the shaft.

- Interface Materials. Carbon-graphite is the most commonly used primary seal ring material because of its chemical inertness, good thermal conductivity, self-lubricating, and self-healing properties. The mating face is usually made of a hard, wear-resistant material; hard metal alloys, ceramics, and cermets are commonly used. The choice of mating material depends on the PV of the seal, the corrosive properties of the fluids, and abrasives in the fluids. Stellite has been used successfully in marine applications.

Radial Seals - General Considerations

- Mounting. The sealing element should be mounted so that it is stationary, usually in the machinery housing, rather than on the rotating shaft. This will keep inertia forces from altering the load on the sealing element and possibly unloading or overloading the seal surfaces. Installing a radial seal over a shaft with keyways, splines, or threads will usually require assembly aides to prevent damage to the sealing element. The sealing element should not have to support the weight of the shaft or housing during assembly because of possible damage to the seal.

- Mating surface finish. The mating surface of a radial seal, on the shaft or a shaft sleeve, will usually have to be supplied by the seal user. A caution which applies to all radial devices is the nature of this mating surface. It is important that machine lead, i.e., the generation of an ordered, helical surface finish, be avoided on the running surface. Machine lead will result in pumping and the direction of the leakage will be sensitive to the direction of shaft rotation. Machine lead can be avoided by plunge grinding the surface with these additional precautions:[18]

- Carefully dress the grinding wheel so that the lead is not generated on the wheel surface and thus transferred to work piece.

- Use nonwhole number ratios between the grinding wheel and work piece speed.

- Allow the wheel to spark out.

Refinishing of the running surface with polishing paper by field personnel should be discouraged. It usually results in the generation of helical surface finish.

- Alignment, tolerances on shaft diameter. The plane of the sealing element should be kept as perpendicular to the shaft axis as possible; tilting or skewing can also result in pumping.

The dimensions of the shaft and/or mounting bore are important since they will determine the spring and/or elastic load on the sealing surface.

- Compatibility of mating surface and sealing element material. Corrosion and chemical attachment or sticking at the seal interface can occur with some combinations of seal/mating surface materials. This problem is usually most severe in applications where relatively long-term static conditions are encountered. As was noted earlier, there is no fluid film at the interface under static conditions, so the seal and shaft surface are in intimate contact. A graphite/stainless steel combination[19] is not good in this respect. Since most dynamic seal materials usually contain graphite to enhance their frictional properties, and stainless steel shafts are commonly used in deep-ocean applications, particular care is necessary when selecting the combination of seal and shaft-surface material. It may be necessary to use a shaft sleeve of material different from that of the shaft, or to treat the shaft surface.

Elastomer Lip Seal or Oil Seal

A lip seal is commonly used to retain fluid in the sump of a machine where the static fluid level is below the level of the seal, or where the seal is not completely submerged. They are generally limited to pressure differentials below 15 psid.

The basic construction features are shown in figure 16. In an assembled seal, the metallic case, which adds rigidity to the seal and aids in mounting, and the elastomer lip element are mechanically held together. In a bonded seal, the elastomer is vulcanized to the metallic case, giving less chance of an internal leak. The elastomer lip may be loaded by a garter or finger spring or there may be no spring. The finger spring is less sensitive to fouling. The springless designs are used primarily for retaining very viscous fluids or grease, for excluding dust and dirt from a system, or where leakage is not critical. They will generally have a lower surface speed limit than those with springs because of their inability to follow eccentricity and dynamic runout.

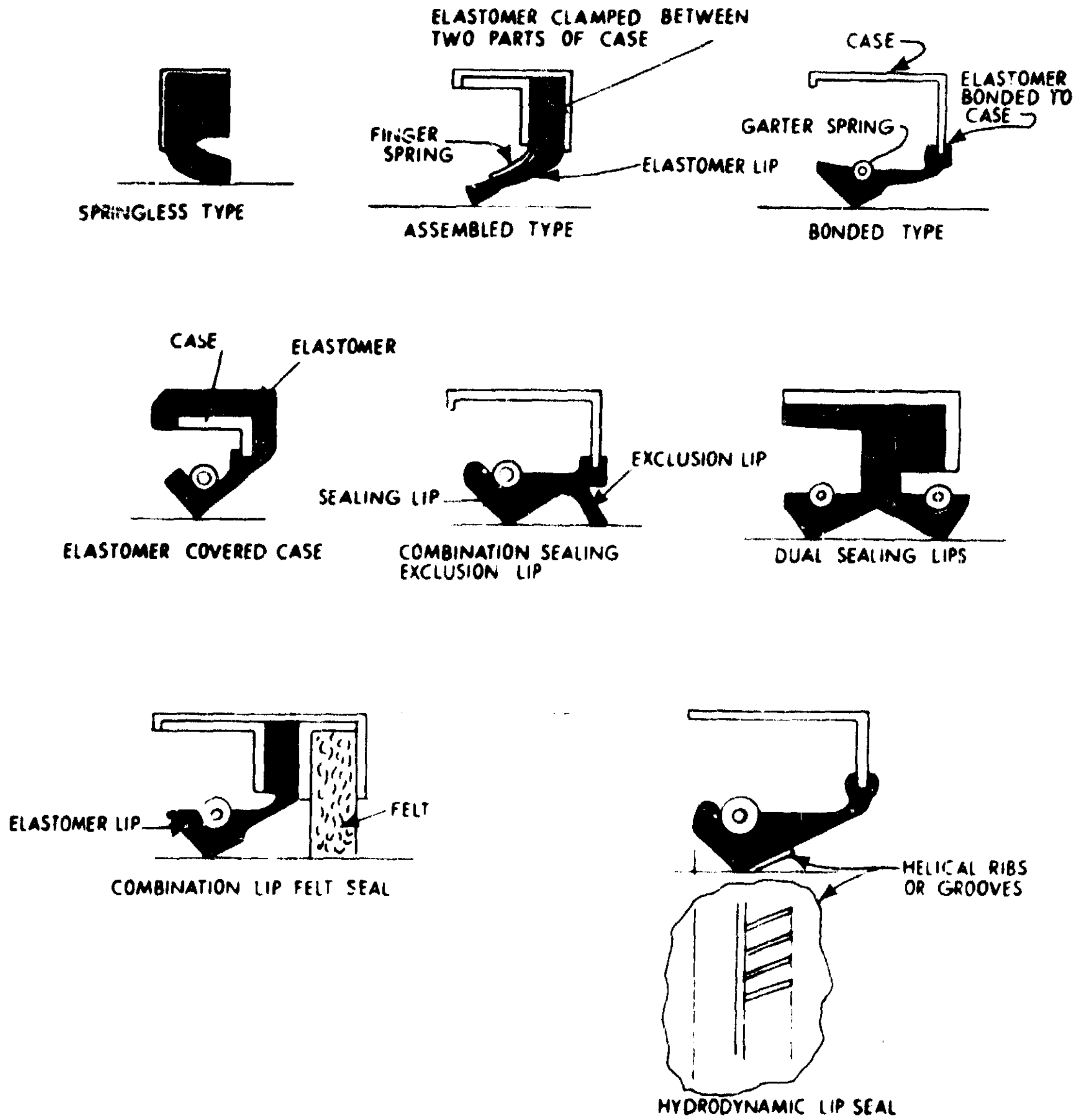


Figure 16 - Variations of Elastomer Lip Seal

Primary considerations for selecting a lip seal are:

- **Materials. The lip material must be compatible with the fluids and temperatures on either side of the seal, and with the mating shaft surface. The compatibility of various candidate compensating fluids with commonly used elastomers has been explained.**[9] The compatibility with fluids which may be used for flushing, cleaning, or storage should also be considered. Consult manufacturer for specific material recommendations. The abrasive resistance of the elastomer and mating surface are also important. Lip seal manufacturers generally recommend a shaft surface finish of 10 to 20 μin. center line average (CLA), free of machine lead. Finishes rougher than this will usually wear the elastomer. Finishes smoother than this can result in wear because they are too smooth to retain lubricant at the seal interface. A shaft hardness of at least Rockwell C30 minimum is generally recommended and some manufacturers recommend a hardness of Rockwell C-50-65 when abrasives are present. Most elastomers will not run dry, even for short periods of time. If the machinery is to be operated without compensating fluid during testing, the seals should be removed or some provision made to lubricate them. It is considered good practice to prelubricate the seals during installation. The case and spring material must also be compatible with the fluids, particularly the seawater.

- Mounting configurations. Consideration should be given to the manner of installing the seal. The configuration of the case and lip usually vary according to whether the seal is assembled on the shaft from the spring side or from the side opposite the spring. Various case and lip configurations are available to provide mounting and assembly convenience and to protect the spring and lip during assembly or operation. The seal is usually pressed into a bore in the machinery housing, usually a bearing bore. Special designs are available for applications where the seal must rotate. The interference fit between the seal and mounting bore keeps the seal from rotating and provides a static seal. Designs with elastomer covered cases are available to provide a static seal in rough bores or where the relative expansion of a metallic case and bore would destroy an interference fit. The bore into which the seal is pressed, or the tool used to press the seal into the bore should have a shoulder on it to ensure that the seal is assembled perpendicular to the bore and shaft axis. General-purpose single lip seals are unidirectional and will be overbalanced when the pressure is from the spring side.

- Lip configuration. Single and multiple lip designs are available. The multiple lip designs are essentially two or more seals in the same mounting case, where the functions of the various seals are usually different. For example, a dual elastomer lipseal with one lip spring loaded and the other nonloaded, figure 16, is usually recommended when dust and dirt are present on the nonloaded lip side. In this arrangement, the spring loaded lip retains the fluid in the machinery, i.e., does the sealing, while the unloaded lip protects it from abrasives. A dual lip design with both lips spring loaded, but with the lips facing in opposite directions is usually recommended for separating two fluids. In this arrangement, both lips are intended to seal fluid from the space between the lips. These multiple elastomer lip arrangements are not recommended for deep-ocean applications because there is no positive way to pressure compensate the area between the lips. It has been suggested that compensation be obtained by packing the area between the lips with grease. This is a questionable practice because the seal will probably run hot, which could cause deterioration of the elastomer, and there still is the possibility of having a pressure void because of incomplete filling or leakage of grease from between the seals. If multiple sealing elements are required, it would be better to use two separate, single elastomer lip seals and pressure compensate the area between them. As an alternative, a design combining a felt seal and elastomer lip in a common case, figure 16, may be applicable. The rationale of this arrangement was discussed earlier. A recent development is the hydrodynamic lip seal. It is similar in construction to a conventional lip seal but has the additional feature of having helixlike ribs or grooves molded into the lip on what is normally the air side of the seal, figure 16. In concept, it is similar to the viscous seal; any leakage past the lip is effectively redirected or pumped back. Such seals have been found more consistent and reliable than a conventional lip seal since they are less sensitive to variations in manufacturing, assembly, and operation which would normally cause leakage in a conventional lip seal. However, their suitability for deep-ocean applications, where fluids are present on both sides of the seal, is questionable because of their designed-in ability to pump.

Elastomer O-Ring

A conventional elastomer O-ring is a simple, compact, easy to mount (one piece groove), bidirectional device for sealing a rotating shaft. Special elastomer compounds and groove dimensions are usually recommended when used for rotating applications and are given in the manufacturer's design literature.[19]

A combination of relatively poor thermal and frictional characteristics and a tendency to stick some shaft materials result in relatively low allowable PV limits and high breakout torque.

The slipper and U-cup seal are usually preferred over the conventional O-ring in dynamic, either rotating or reciprocating, applications. They are generally used for applications with high differential pressures, but their simplicity and compactness make them attractive for deep-ocean applications where space is at a premium and the higher allowable PV's of a face seal are not required.

O-Ring Slipper or Cap Seal

The cap seal consists of a conventional elastomer O-ring or other elastomer support ring with a separate sleeve or cap of low friction, low wear material between the O-ring and dynamic surface, figure 17.

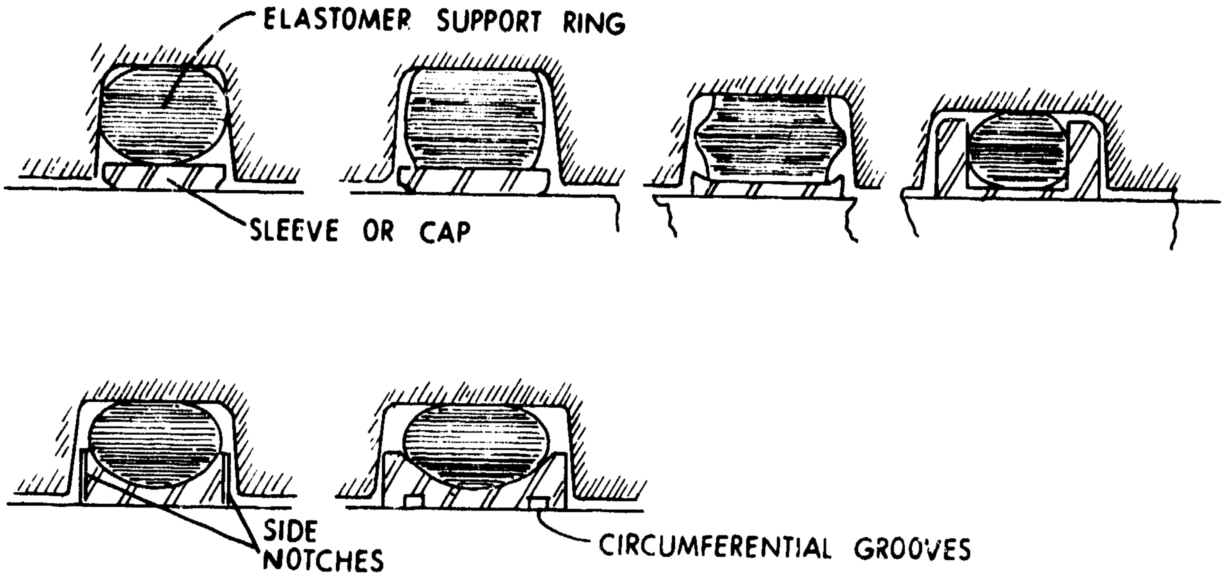


Figure 17 - Examples of Slipper or Cap Seals

The O-ring keeps the slipper from rotating, serves as a static seal between the slipper and groove, provides part of the seal closing force, and gives radial compliance to the seal. Relative motion takes place at the interface of the cap and shaft surface rather than on the surface of the O-ring. Since the frictional properties of the sleeve, which is usually made of Teflon or filled Teflon, are better than the elastomer, it will generally have higher PV limits and eliminate or reduce sticking and high breakout forces encountered with a plain O-ring.

Specific features requiring consideration are:

- Conformability of the slipper. The conformability of the slipper is particularly important in applications with low differential pressure because a large hydrostatic closing force is not available for forcing the slipper into conformity with the shaft. The elastomer provides the closing force in low-pressure applications. The conformability of the slipper is determined by its crosss-sectional shape, the hardness and squeeze (gland dimensions) of the elastomer, and by the slipper material. The slipper will usually have a rectangular, channel, or modified channel cross section, figure 17. The rectangular shape is usually preferred for rotating applications because of its better flexibility; the channel shapes are commonly used where extrusion resistance is required. Some manufacturers put circumferential grooves on the running surface of the slipper to increase its conformability. The slipper must not be so thin radially, however, that wear life is limited. The gland dimensions for rotating applications will usually provide for more radial squeeze on the elastomer than that for reciprocating applications.

- Materials. The slipper and elastomer material must be compatible with the fluids and temperatures on either side of the seal. The slipper is usually made of virgin or filled Teflon. Glass fibers, metal fibers, carbon and graphite are commonly used as fillers to enhance extrusion resistance, thermal conductivity, and friction and wear properties. The filled materials will generally have a higher allowable $\overline{P}V$. The slipper filler must be compatible with the shaft surface material. An important consideration when using these Teflon materials is the surface tension of the fluids surrounding the seal. To obtain low wear rates with these materials, it appears[20] that the Teflon must transfer itself to the mating surface. Some fluids, water is a good example, do not wet Teflon and inhibit the transfer of Teflon to the mating surface thus producing high wear rates. Most of these materials exhibit higher wear rates when lubricated with water than when completely dry. This problem also appears to depend upon the regime of operation of the seal, it being most severe when operating under mixed-film conditions. Under contact conditions, no fluid is present at the seal interface and Teflon can be freely transferred. When operating hydrodynamically, the surfaces are completely separated by a fluid film and the nature of the seal interface makes little difference on the wear rate. Carbon filled Teflon is usually recommended in a water environment.

- Mounting arrangement. Assembly of a cap seal into a radial configuration is usually easier if a two-piece gland is used. They can usually be assembled into one piece if special tools and techniques are used. Such a gland can also be mounted in an axial-type configuration, figure 6, to simplify the groove and assembly, but this will require a mating ring and close control over the axial stack-up tolerances, similar to that of a mechanical face seal. With some slipper configurations, cold flow of the slipper or too narrow a groove, can result in the slipper sealing at the sides of the groove. If the pressure is suddenly changed, this could trap high-pressure fluid behind the slipper, severely overbalancing the seal, or could keep pressure from penetrating behind the slipper, underbalancing the seal. This type of problem is usually cured by putting notches in the groove sides of the slipper. Too wide a gland could conceivably result in tilting or skewing of the slipper since the pressure differential may not be high enough to force the seal to the low-pressure side of the gland. The slipper seal will usually be bidirectional since the cross-sectional shape of the slipper and elastomer are usually symmetrical.

U-Cups

U-cup devices consist of a ring of low friction material, usually Teflon or a filled Teflon, with a U- or C-shaped cross section, and a spring within the U-section, figure 18.

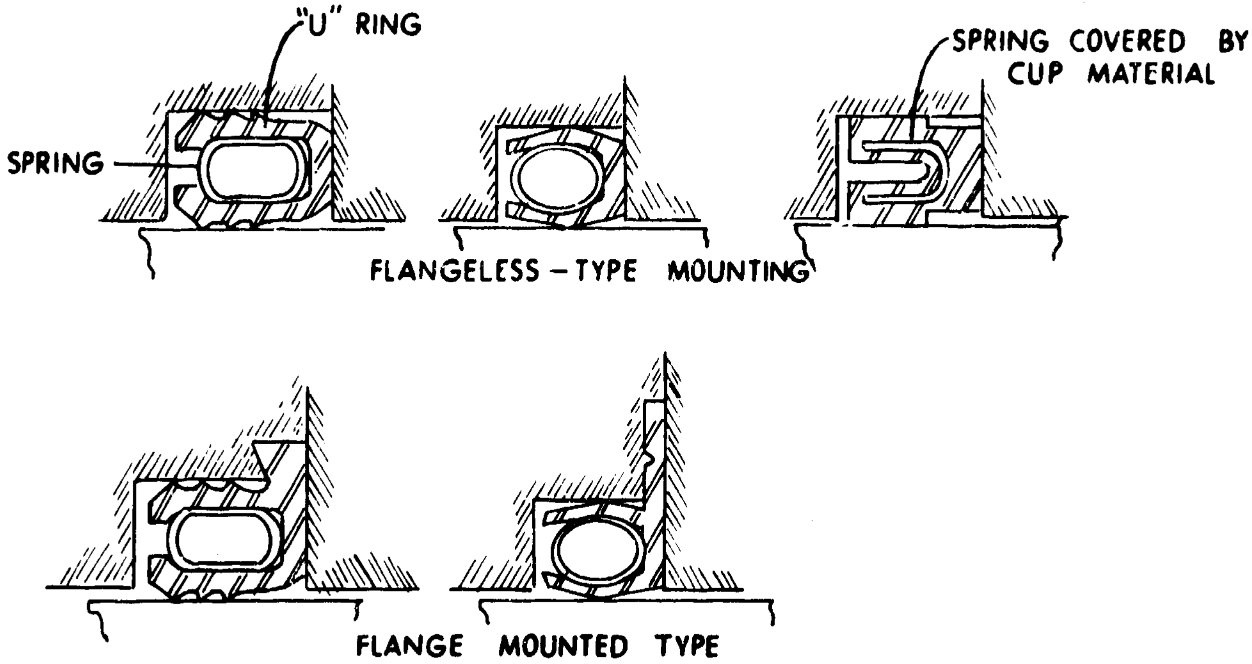


Figure 18 – Examples of U-Cup Seals

Some have liplike projections on the legs of the U to give a more conformable sealing surface. The radial load from the elasticity of the material, spring, and hydrostatic pressure, forces the outer leg against the groove to form a static seal and the inner leg against the shaft to from the dynamic seal. The base of the U is usually relatively thick compared to the legs and serves as an antiextrusion device.

Basic design variations on this device are:

- Mounting. U-cups are available with flange and flangeless-type mountings, figure 18. The flangeless type is generally used for static or reciprocating applications. The flange type is generally preferred for rotary applications because the flange provides positive retention against rotation and prevents skewing. The flange type requires a two-piece gland. U-cups are unidirectional and will be overbalanced when the pressure differential is from the open side of the U.

• Springs. In some designs, figure 18, the spring is imbedded and completely covered by the cup material. This protects the spring from corrosion and fouling. In other designs, figure 18, the spring is exposed to the fluids and can be separated from the cup. This arrangement has versatility since the spring load can be changed to meet the operating requirements. Precautions on material selection and groove dimensions are the same as for the slipper seal. Material-fluid compatibility is somewhat less critical because no elastomers are used.

CHAPTER IV

VALIDATE SEAL SYSTEM SELECTION PERFORMANCE DATA

Basic performance data, in particular the relative leakage rates of seawater into and compensating fluid out of the system, wear rates (life), power loss, and operating limits for seals in deep-ocean applications, are sparse. This has made it difficult to size compensators or set up operating and maintenance schedules.

Performance data from seal screening tests and from the electric propulsion motor tests at Annapolis and field operational experience will be incorporated in this chapter as such information becomes available. These data should give the seal user a feel for the level of performance obtainable from the seal so that realistic system designs and operational schedules can be formed.

ANNAPOLIS SEAL SCREENING TESTS

Screening tests are being conducted on off-the-shelf seals to provide a relative comparison of the performance of the various seal configurations, to uncover problem areas, and to provide basic performance data under simulated deep-ocean conditions.

The screening tests are being conducted in a simulated deep-ocean technology motor-seal configuration shown schematically in figure 19. This fixture allows flexibility in the type of seal configuration, seal system configuration (single or double seal), fluid, and pressure environment of the seal. The fixture is supported on a ball-bearing cradle so that the torque reaction; and thus, seal torque can be measured by a load cell. The shaft is 1.375 inches in diameter, sleeved up to 1.75 inches in diameter. Shaft speed is variable from 0 to ±3600 rpm. Acrylic cylindrical sections form the walls of the seal cavities and permit visual observation of the fluid in the various cavities. The system pressure is limited to about 500 psig.

Severn River water is circulated through the end cavity, to simulate the ocean environment, and then into a trap when any oil that has leaked past the outboard seal can be detected. The water cavity is sealed from the atmosphere by a high-pressure face seal.

- **Face seal performance.** Design features of the face seal selected for test are shown in table 2. This seal is similar to that used on several of the Annapolis electric drive systems. A similar type of face seal has been used in other deep-ocean applications. The seal was first tested with the system at ambient atmospheric pressure, in a single seal system configuration, table 2. Test data obtained to date for this seal are summarized in table 2. Notice in particular that the leakage rate of water into the system was, for all practical purposes, nil.

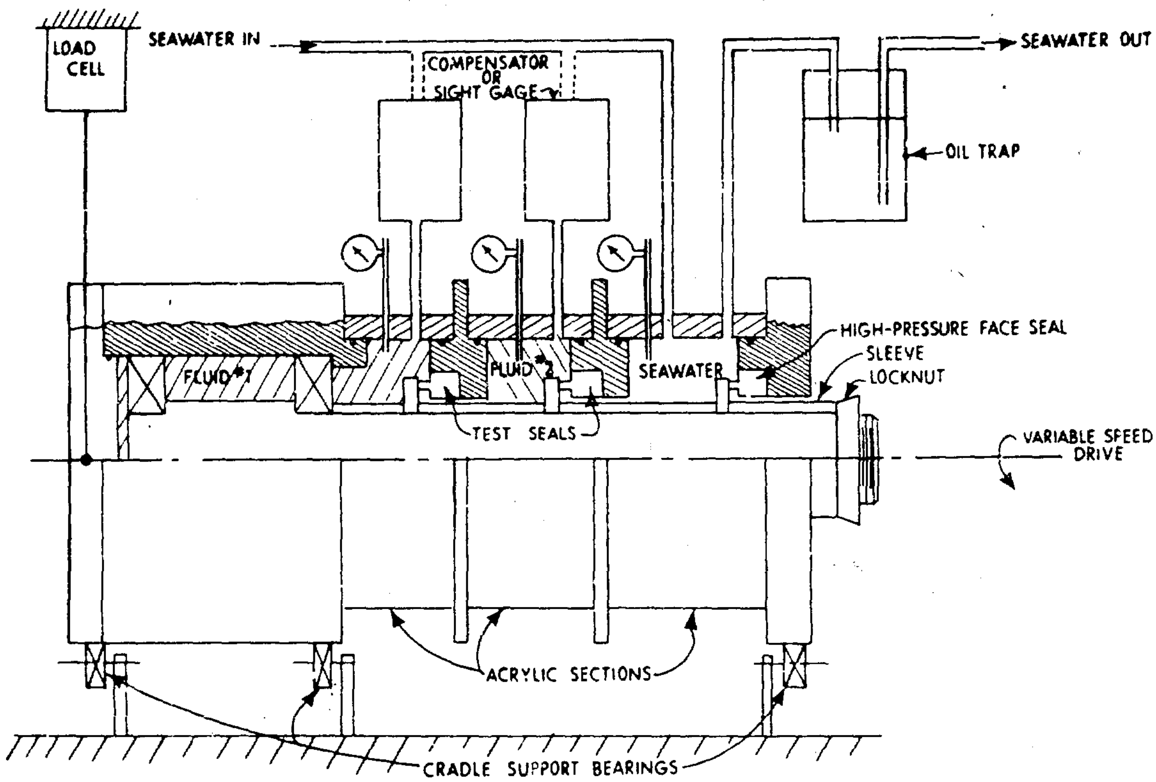


Figure 19 - Seal Test Fixture

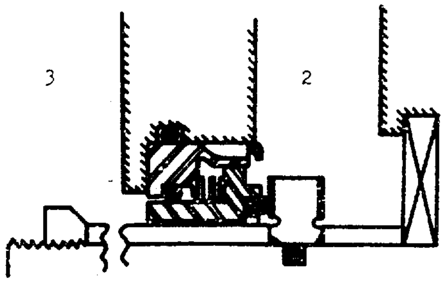
Notice also that speed and the magnitude of the pressure differential had no noticeable effect on the leakage. The leakage of oil into the water was also nil, except that a trace of oil was sometimes detected upon start-up, particularly after the seal had not operated for a few days. Radial runout on the shaft for these runs was less than 0.001 inch. The same seal was then run at the outboard side in a dual system configuration, table 3, with the the system at ambient pressure. Water and oil were, however, detected leaking past the outboard seal at this time, table 3, and the high-pressure seal was leaking water out and pumping air into the water cavity. The leakage was traced to radial runout of the shaft. Apparently during the reassembly into the dual seal configuration, the lock nut on the end of the shaft was overtightened, an error which distorted the shaft so that it was running out radially 0.013 to 0.020 inch at the drive end. Incidentally, the pumping of air into the water compartment was stopped by increasing the spring load (reducing its working height) on the high-pressure seal.

Table 2

Data for Face Seal in Single Seal System

<u>Seal Features</u>

1. Cup-type face seal with wave springs and elastomer O-ring secondary seal
2. Materials:
   Structure – Monel
   Seal face – carbon-graphite-resin
   Mating face – Stellite overlay on Monel
3. Spring load = 20 psi



<u>Notes</u>

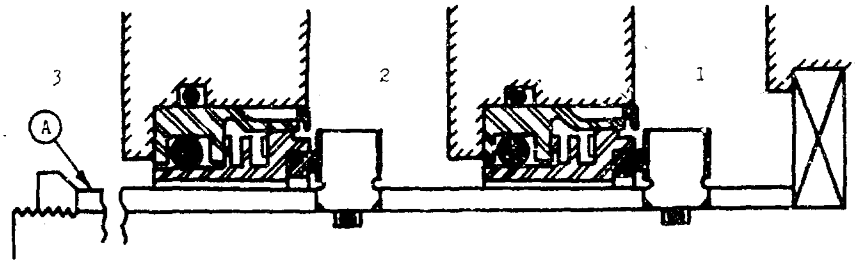
(1) None under steady-state conditions. However, a trace of oil in water was usually observed during starting, particularly after seal had set statically for few days.

(2) Water in oil determined by ASTM D-95-62 (water by distillation).

Fluid Environment			Shaft Speed	System Pressure	ΔP psig		Running Time	Average Leakage Rate cc/hour				Relative Torque	Average Wear Rate in./hour		TIR on Shaft
Fluid 3	Fluid 2	Fluid 1	rpm	psig	1 2	2 3	hours	(2) 3 2	2 3	2 1	1 2	oz.-in.	Seal Face	Mating Face	in.
Severn River water	MIL-H-5606 B	–	1800	Ambient	–	0	50	None	(1)			65			<0.001
			1800		–	5	50	Trace	(1)						
			100		–	0	50	Trace	(1)			20			
			100		–	5	50	Trace	(1)						
			±1800 1 Reversal per hour		–	0	8	None							
					–	5	8	Trace							
			+1800 2 Reversals per minute		–	0	0.5	None							
					–	5	0.5	Trace							
			±100 1 Reversal per hour		–	0	8	Trace							
					–	5	8	None							
			±100 2 Reversals per minute		–	0	0.5	None							
					–	5	0.5	None							
			0		–	0	43	None							
			0		–	5	48	None					(For 250 hours) 5.8	0.16	

Table 3
Data for Face Seal in Double Seal System

Seal Features
1. Same as Table 2
2. Spring load = 20 psi (for both seals)



Fluid Environment			Shaft Speed	System Pressure	ΔP psig		Running Time	Average Leakage Rate cc/hour				Relative Torque	Average Wear Rate in./hour		TIR on Shaft
Fluid 3	Fluid 2	Fluid 1	rpm	psig	1 2	2 3	hours	(2) 3 2	2 3	2 1	1 2	oz.-in.	Seal Face	Mating Face	in.
Severn River water	MIL-H-5606 B	MIL-H-5606 B	1800	0	0	0	19	0.02	0.545	0.105	0.0	72			0.013 to 0.020" at point A
				Test Stopped to Investigate Cause of Radial Runout											

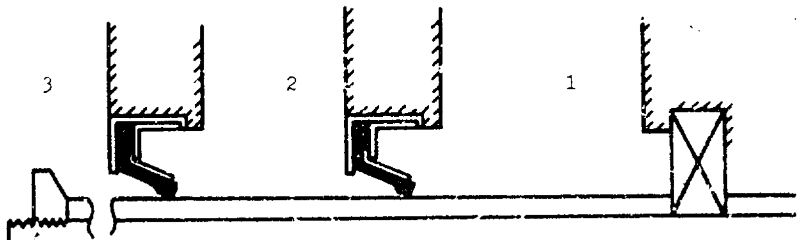
• Lip seal. A general-purpose, elastomer lip seal, table 4, is now under test. Data will be given when available.

• Other. An example of a slipper seal, spring-load U-cup and a dual seal system with an outboard felt seal and/or a filter arrangement are also planned for test. The slipper and U-cup seals have been tried with success on deep-ocean hydraulic actuator applications, to separate a hydraulic fluid and seawater. These applications, however, usually have oscillating or reciprocating shafts and operate with much higher pressure differentials.

NSRDC SUBMERSIBLE ELECTRIC DRIVE SYSTEMS

As was noted earlier, the pressure-compensated drive systems in the deep-ocean technology electric drive systems task usually use a double face seal arrangement. No serious problems with the seals have been reported to date. A trace of water is usually found in the seal cavity, but the level of contamination has not been a problem. Static leakage immediately after assembly has been reported in some instances. This has been due to contaminants getting between the seal faces at assembly. Sealing was restored by disassembling and cleaning the seal interface.

Table 4
Data for Elastomer Lip Seal in Double Seal System



Seal Features

1. General purpose, single lip "oil seal" with finger spring
2. Materials:
   Elastomer - general purpose elastomer (nitrile)

Shaft sleeve - 316 SS, super finished to 14 RMS OD = 1.748 inches

Fluid Environment			Shaft Speed	System Pressure	ΔP psig		Running Time	Average Leakage Rate cc/hour				Relative Torque	Average Wear Rate in./hour		TIR on Shaft
Fluid 3	Fluid 2	Fluid 1	rpm	psig	1 2	2 3	hours	(2) 3 2	2 3	2 1	1 2	oz.-in.	Seal Face	Mating Face	in.
Severn River water	MIL-H-5606 B	MIL-H-5606 B	100	Ambient	0	0	50	None	Trace	0.02	None	17			<.001
Severn River water	MIL-H-5606 B	MIL-H-5606 B	1800	Ambient	0	0	50	Trace	None	0.185	None	47			<.001

OTHER

Leakage levels of the order of 1 cc/hr or less, usually in the range of 10^{-2} to 10^{-1} cc/hr, have been reported with face seals in some component tests conducted by equipment manufacturers. These levels have not been troublesome.

Problems have been reported, but these have usually been traced to some operational procedure, or interaction with the pressure compensator which has resulted in unusual or very severe seal operating conditions.

CHAPTER V

NOTATION AND TECHNICAL REFERENCES

NOTATION

A_f	area of the seal interface
A_c	net closing area
B, S, D, E, E', E"	seal interface geometry terms
G	duty parameter $= \mu Vb/W_c$
$\overline{P}$	unit load at the seal interface
ΔP	hydrostatic pressure differential across the seal
Q(hs)	hydrostatic leakage
Q(hd)	hydrodynamic leakage
Q(c)	combined or total leakage
T	torque absorbed by the seal
V	relative velocity at the seal interface
W_c	total seal closing force
W_{SPRING}	component of closing force due to spring load
$W_{ELASTIC}$	component of closing force due to elastic elements of seal
W_o	total seal opening force
W_o(hs)	hydrostatic component of opening force
W_o(hd)	hydrodynamic component of opening force
b	width of the seal interface
f	coefficient of friction
h	mean interface clearance
μ	fluid viscosity at the seal interface

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13. ABSTRACT

This manual contains a procedure for selecting machinery seal for deep-submergence use.

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1. Attached is an updating of Chapter IV of the subject handbook. Please remove the existing Chapter and replace it with the new one.

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By direction

CHAPTER IV

VALIDATE SEAL SYSTEM SELECTION-PERFORMANCE DATA

Basic performance data for off-the-shelf seals in deep-ocean applications are sparse in particular the relative leakage rates of sea water into, and compensating fluid out of, the system. This has made it difficult to size compensators or set up operating and maintenance schedules.

Performance data from seal screening tests and from the submersible electric drive systems tested at NSRDC and field operational experience are incorporated into this chapter. These data should give the seal user a feel for the level of performance obtainable so that realistic system designs and operational schedules can be formed.

SEAL SCREENING TESTS AT NSRDC

Screening tests were conducted on an off-the-shelf mechanical face seal and elastomer lip seal. Details of the test seals, which were for a 1.75-inch-diameter shaft, are shown in tables 2, 3, and 4. The purpose of these tests was to provide an order-of-magnitude indication of the leakage levels obtainable with these devices when in a liquid/liquid environment, particularly to see whether hydrodynamic pumping leakage is a serious problem.

- Test fixture. Screening tests were conducted in a fixture simulating a single-ended pressure compensated motor, figure 19. The seals were mounted in stainless steel partitions which were spaced by transparent acrylic sleeves to form either two or three fluid cavities, depending upon whether a single or double seal system was simulated. Sight gages were used on cavities 1 and/or 2 to monitor the fluid levels. A pressure differential was obtained across the seal by applying air pressure above the fluid in the sight gage. Severn River water (brackish water) was circulated through the end compartment, cavity 3, to simulate the ocean environment, and then into a trap where any oil that had leaked past the outboard seal could be detected. The water cavity was sealed from the atmosphere by a mechanical face seal. Thermocouples indicated the fluid temperatures in each cavity.

The whole assembly was supported on a ball-bearing cradle so that the torque reaction, and thus torque to drive the seals and bearings, could be measured by a load cell.

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Data for Face Seal in Single Seal System

Seal Features

1. Cup-type face seal with wave springs and elastomer O-ring secondary seal, clamped-type mating ring

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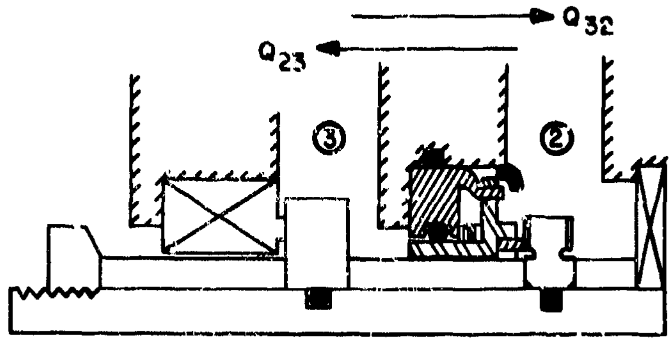
3. Spring load = 20 psi



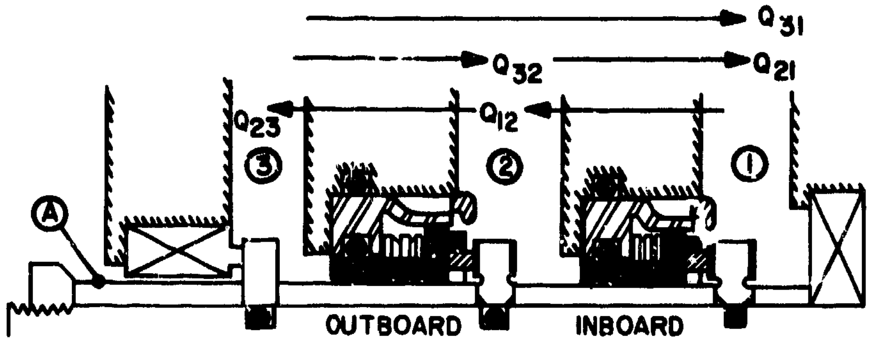
Run No.	Fluid Environment		Shaft Speed rpm	ΔP psig 2→3	TIR or Eccentricity on Shaft in.	Running Time hr	Average Leakage Rate cc/hr		Total Torque oz-in.
	Fluid 3	Fluid 2					Calculated $Q_{32} - Q_{23}$	By ASTM-D-95-62 Q_{32}	
1	River water	MIL-H-5606B	1800	0	<0.001	50	-0.08	<.001	65
2			1800	5		50	-0.06	<.001	90
3			100	0		50	-0.01	<.001	20
4			100	5		50	-0.05	<.001	33
5			±1800 1 reversal per hour	0		8	-0.20	None	
6				5		8	-0.40	<.001	
7			±1800 2 reversals per minute	0		0.5	-1.58	None	
8				5		0.5	-2.69	<.001	
9			±100 1 reversal per hour	0		8	-0.13	<.001	
10				5		8	-0.42	None	
11			±100 2 reversals per minute	0		0.5	-0.68	None	
12				5		0.5	-1.62	None	
13			0	0		48	None	None	
14			0	5		48	None	None	
Average wear rate for 250 hours of operation: seal face = 5.8 μin/hr; mating face = 0.16 μin/hr.									

Table 3
Data for Face Seal in Double Seal System

Seal Features

1. Same as table 2

2. Spring load = 20 psi (for both seals)



Run No.	Fluid Environment: Fluid 3	Fluid 2	Fluid 1	Shaft Speed rpm	ΔP psig: 1→2	2→3	TIR or Eccentricity on Shaft in.	Running Time hr	Average Leakage Rate cc/hr, Calculated: $Q_{32}-Q_{23}$	$Q_{21}-Q_{12}$	By ASTM-D-95-62: Q_{32}	Q_{31}	Total Torque oz-in.
15	River water	MIL-H-5606B	MIL-H-5606B	1800	0	0	TIR = 0.013 to 0.020 at A; 0.002 to 0.003 at test seals	19	-0.33	- 0.27	0.16	None	72
← Stopped test to investigate cause of radial runout →													
16	River water	MCS 1022	MIL-H-5606B	1800	0	0	<0.001	29.25	-0.40	- 0.07	Trace	Trace	84
17	↓	↓	↓	1800	5	5	↓	25	-0.52	- 0.35	Trace	↓	100
18	↓	↓	↓	90	0	0	↓	25	-0.04	+ 0.01	Trace	↓	44
19	↓	↓	↓	90	5	5	↓	25	-0.16	- 0.24	Trace	↓	65
← Axis of carbon face eccentric relative to shaft axis by 0.055 inch on following runs →													
20	River water	MIL-H-5606B	MIL-H-5606B	100	0	0	0.055	1	+0.04	- 0.5	<.01	None	
21	↓	↓	↓	100	5	5	↓	1	-0.45	- 1.37	<.01	None	
22	↓	↓	↓	1800	0	0	↓	1	+6.67	-17.052	5.18	None	
23	↓	↓	↓	1800	5	5	↓	1	+4.77	-16.92	5.92	None	
24	River water	MCS 1022	MIL-H-5606B	1800	0	0	↓	1	+5.5	-13.16	5.18	None	
25	↓	↓	↓	1800	5	5	↓	1	+3.58	-15.19	5.18	None	

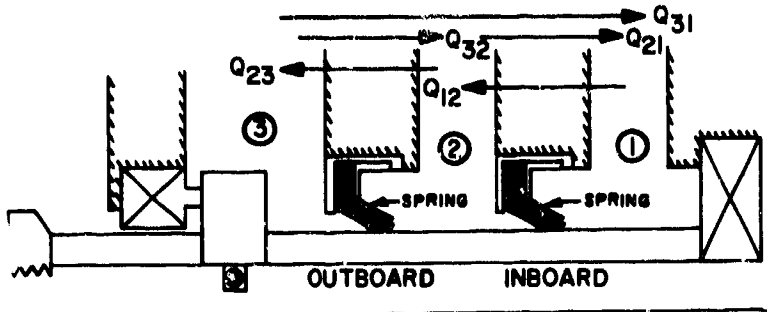
Table 4
Data for Elastomer Lip Seal in Double Seal System

Seal Features

1. General purpose, single lip "oil seal" with finger spring

2. Materials:
   Elastomer - general purpose elastomer (nitrile)

Shaft sleeve - 316 SS, super finished to 14 RMS OD = 1.748 inches



Run No.	Fluid Environment: Fluid 3	Fluid 2	Fluid 1	Shaft Speed rpm	ΔP psig: 1→2	2→3	TIR or Eccentricity on Shaft in.	Running Time hr	Average Leakage Rate cc/hr: Calculated $Q_{32}-Q_{23}$	$Q_{21}-Q_{12}$	By ASTM-D-95-62 Q_{32}	Q_{31}	Total Torque oz-in.	Reversed Pressure psig: 3→2	2→1
26	River water	MIL-H-5606B	MIL-H-5606B	100	0	0	<0.001	50.5	-0.08	-0.04	None	None	17	9.5	10
27	↓	↓	↓	1800	0	0	↓	50.5	-0.17	+0.05	<.001	None	47		
← Seal sat statically for 6 months, then disassembled for inspection. Wear on sleeve: at outboard seal = 0.4 µin/hr; at inboard seal = not measurable →														4.1	0.4
← Installed new set of seals and refinished shaft →														9.5	10
28	River water	MIL-H-5606B	MIL-H-5606B	100	5	5	<0.001	50	-0.10	-0.06	.004	None	99		
29				1800	5	5		50	-0.04	+0.03	Trace	Trace	119		
30				±1800 1 reversal per hour	0	0		8	-0.18	-0.12	0.09	None			
31					5	5		8	+0.04	-0.01	0.02	None			
32				±1800 2 reversals per minute	0	0		0.5	-0.27	-1.04	0.26	None			
33					5	5		0.5	-2.34	-1.98	None	None			
34				±100 1 reversal per hour	0	0		8	+0.05	+0.09	Trace	None			
35					5	5		8	-0.15	-0.05	None	None			
36				±100 2 reversals per minute	0	0		0.5	-0.72	-0.37	None	None			
37	↓	↓	↓		5	5		0.5	-0.5	-0.5	None	None			
38	River water	MCS 1022	MIL-H-5606B	1800	0	0	↓	25	+2.0	Trace	1.95	None	87		
39		MCS 1022		1800	5	5	<0.001	26.25	-0.11	-0.02	Trace	None	125		
40		Hoover #2		1800	0	0		25	-0.18	-0.02	Trace	None	88		
41	↓	↓	↓	1800	5	5	↓	25	-0.31	-0.02	Trace	Trace	121		
← Wear on sleeve: at outboard seal = 4.0 µin/hr; at inboard seal = not measurable →														6.2	7.3

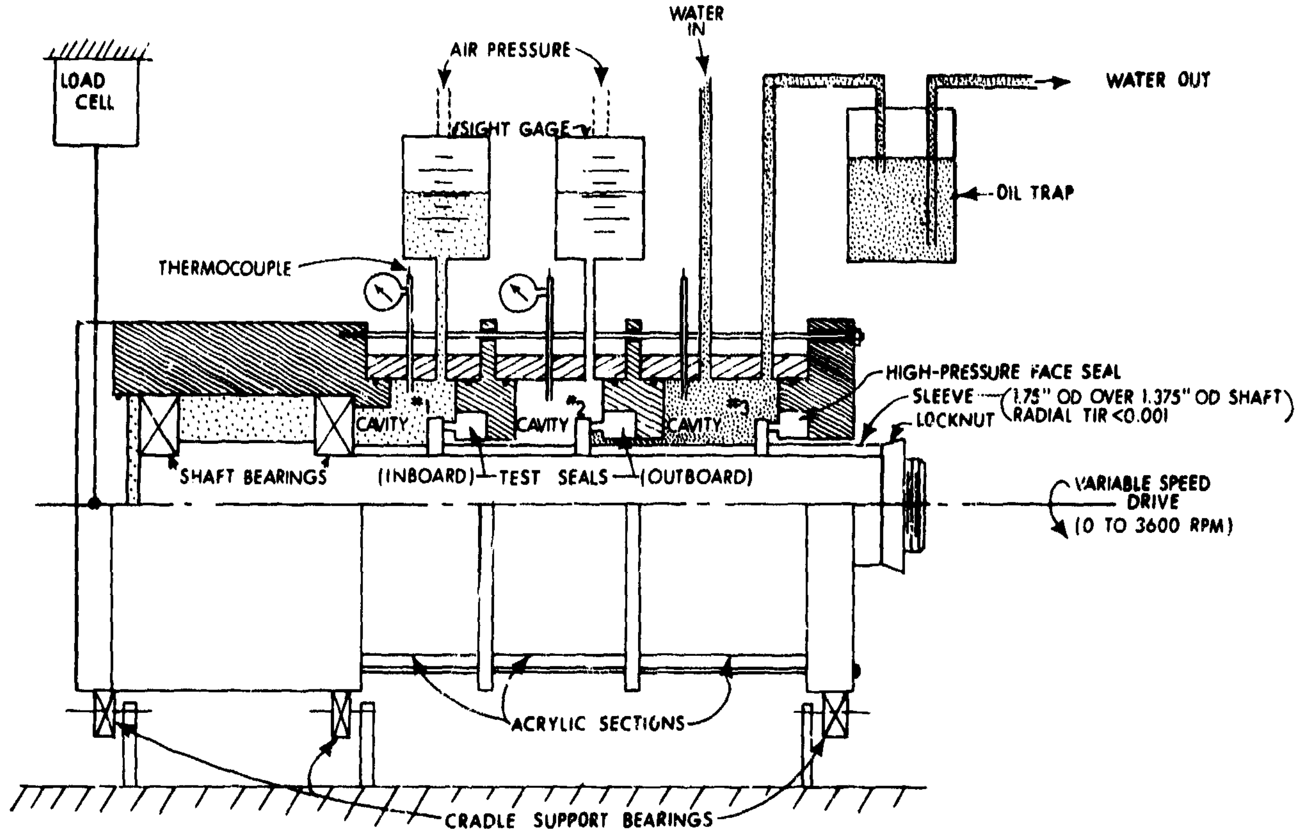


Figure 19
Seal Test Fixture

• Test procedure. All tests were conducted with the water cavity at ambient atmospheric pressure. Test conditions were as shown in tables 2, 3, and 4 and consisted essentially of a high and low speed steady-state run, with and without a pressure differential across the seal, and a series of transient start-stop-reversing runs. MIL-H-5606B, a fluid commonly used in deep-ocean applications,[9] was used as the compensating fluid; two other fluids, intended for specific deep-ocean applications, were also run.

The amount of water leaking into the compensating fluid was determined by ASTM-D95-62 (Water by Distillation). Relative leakage rates of the compensating fluid across the seals were calculated from the sight gage readings after correcting these readings for volume changes due to temperature.

The mechanical face seals were set up to give the face load (i.e., spring load) suggested by the manufacturer. An indication of the lip load on the elastomer lip seals was obtained by applying a reversed pressure differential across the seal: The lip seals were assembled in the test fixture and compensating fluid was put into the cavity on the spring side of either the inboard or outboard seal. Air pressure was then applied to the empty cavity on the side opposite the spring until air bubbles, which indicated the lip had lifted from the shaft, appeared in the compensating fluid. These pressures, noted in table 4 as Reversed Pressure, were used to monitor the lip load.

Eccentric operation has been associated with hydrodynamic pumping leakage effects in face seals.[2, 5, 6] To determine the order-of-magnitude of leakage produced by these effects, the acrylic sleeves were machined so that the axis of the carbon face, on both the inboard and outboard seal, was eccentric relative to the shaft axis by the width of the carbon face, 0.055 inch. This eccentricity exceeded the limit suggested by the seal manufacturer and is probably more than normally would occur in practice. However, it was hoped that exaggerating the eccentricity would give some indication as to just how serious hydrodynamic pumping leakage could be.

• Test results. Data for the tests are shown in tables 2, 3, and 4. The performance of the test seals when operated within the limits suggested by their manufacturer was characterized as follows:

- The leakage rates of water and compensating fluid across both the elastomer lip and mechanical face seal was generally low, of the order of 10^{-3} to 1.0 cc/hr.

- Rapid start-stop-reversing transients tended to produce slightly more leakage than steady-state running.

- The torque increased with speed and the pressure differential.

- Wear on the outboard seal was relatively greater than that on the inboard seal because of the more abrasive environment at the outboard seal.

- The lip load on the elastomer lip seal, as indicated by the Reversed Pressure in table 4, decreased noticeably with running time. This was probably due to loss of interference at the lip-sleeve interface caused by permanent set of the elastomer and wear of the lip-sleeve interface; note that the outboard seal had relatively greater wear than the inboard seal and also had a greater loss in lip load.

Eccentric operation of the face seal, particularly at high shaft speeds, table 3 (runs 20 through 25), produced greater leakage across both the inboard and outboard seal than when the seal was centered as was expected. The leakage of water across the outboard seal persisted even when a 5 psi differenital, from compensating fluid to water, was applied to the seal, table 3.

Excessive radial runout also produced leakage. On run 15, table 3, the locknut, figure 19, was accidentally overtightened, a mishap which distorted the shaft and caused it to run out radially 0.013 to 0.020 inch at the high-pressure seal and 0.002 to 0.003 inch at the test seals. This runout resulted in simultaneous outward leakage of water and ingestion of air across the high-pressure seal at the atmosphere/water interface.

Eccentricity and radial runout in a face seal produces a wiping action at the sealing interface. A similar wiping action occurs in a lip seal when the plane of the lip is not perpendicular to the shaft axis or when axial runout is present and would be expected to produce greater leakage.

SEAL PERFORMANCE ON NSRDC SUBMERSIBLE ELECTRIC DRIVE SYSTEMS

As was noted earlier, the pressure-compensated drive systems in the deep-ocean technology electric drive systems task usually employ a double face seal arrangement. No serious problems with these seals have been reported to date. A trace of water is usually found in the seal cavity, but the level of contamination has not been a problem. Static leakage immediately after assembly has been reported in some instances. This leakage has been due to contaminants that get between the seal faces at assembly. Sealing was restored by disassembling and cleaning the sealing surfaces.

One instance of spring failure, because of corrosion damage, was reported on the outboard seal of one motor. The sealing arrangement on this particular application was such that the springs on the outboard seal were exposed to sea water (see item (b) of figure 15). Inverting the seal, so the springs were in the compensating fluid rather than the sea, would have minimized chances of corrosion damage.

OTHER

Leakage levels of the order of 1 cc/hr or less, usually in the range of 10^{-2} to 10^{-1} cc/hr., have been reported with face seals in some component tests conducted by equipment manufacturers.[8] These levels have not been troublesome. Problems have been reported, but they have usually been traced to some operational procedure or interaction with the pressure compensator which has resulted in abnormal or very severe seal-operating conditions.

The slipper and U-cup-type seals have been used with success on deep-ocean hydraulic actuator applications, to separate a hydraulic fluid and sea water. These applications, however, usually have oscillating or reciprocating shafts and operate with much higher pressure differentials.