NUC TP 285

AD 741210

SEALS FOR THE OCEAN ENVIRONMENT

by
Leonard J. Martini
Weapons and Countermeasures Department

April 1972

Approved for public release; distribution unlimited.

37

NAVAL UNDERSEA RESEARCH AND DEVELOPMENT CENTER, SAN DIEGO, CA. 92132

AN ACTIVITY OF THE NAVAL MATERIAL COMMAND

CHARLES B. BISHOP, Capt., USN
Commander

Wm. B. McLEAN, Ph.D.
Technical Director

ADMINISTRATIVE STATEMENT

This report reviews the three basic types of seals that readily apply to the ocean environment, both at sea level and at ocean depths: linear seals, dynamic seals, and penetrant sealants.

This report was originally written as a design survey in partial fulfillment of requirements for an MSME degree at the University of Southern California. It has been reviewed for technical accuracy by R. K. Gottfredson, L. C. Peterson and R. T. Simeral of this Center.

Released by
D. D. AYERS, Head
Torpedo and Countermeasures
Division

Under authority of
C. G. BEATTY, Head
Weapons and Countermeasures
Department

UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Naval Undersea Research and Development Center San Diego, California 92132	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED 2b. GROUP

3. REPORT TITLE

SEALS FOR THE OCEAN ENVIRONMENT

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
Research report (June 1971–January 1972)

5. AUTHOR(S) (First name, middle initial, last name)

Leonard J. Martini

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
April 1972	36	12

8a. CONTRACT OR GRANT NO.	9a. ORIGINATOR'S REPORT NUMBER(S)
b. PROJECT NO.	NUC TP 285
c.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d.	

10. DISTRIBUTION STATEMENT Approved for public release; distribution unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Chief of Naval Material Washington, D.C. 20360

13. ABSTRACT

Three basic types of seals used to exclude the ocean environment are described: linear seals, dynamic seals, and general penetrant sealants. The section on linear seals culminates with a list of important design considerations for ideal flange seals. The section on dynamic seals reviews the theoretical aspects of designing rotary and reciprocal seals. An example design analysis, including sample calculations for a rotary O-ring spindle seal, is presented with emphasis on how environmental parameters affect dimensional characteristics. The section on general penetrant sealants introduces various types of rigid and flexible epoxy and elastomeric sealants used around electrical cables, hoses, tubes, etc., that penetrate walls of submerged structures.

DD FORM 1 NOV 65 1473 (PAGE 1)

S/N 0101-807-6801

UNCLASSIFIED
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Linear seals						
Dynamic seals						
General sealants						
Rotary seals						
Reciprocal seals						
O-rings						

SUMMARY

PROBLEM

To review the three basic types of seals used in the ocean environment.

RESULTS

A discussion is presented of the behavior of linear seals, dynamic seals, and general sealants at sea level and at ocean depths. An example design problem is presented and solved to show the relationships between rotary O-ring spindle seals and their environmental parameters.

RECOMMENDATIONS

Continued study of all types of seals used in marine applications is required to assure their most effective utilization.

CONTENTS

NOMENCLATURE

A	Contact area, in^2
C	Clearance between shaft and seal housing, in.
D	Depth of O-ring groove measured from shaft, in.
E	Young's modulus of elasticity, psi
f	Factor dependent on percent diametral reduction of O-ring due to peripheral compression
G	Maximum internal diameter of O-ring groove, in.
H	Internal diameter of seal housing, in.
I	Factor dependent on percent increase in cross-sectional diameter of O-ring due to peripheral compression
L	Unit loading, lb/in
N	Speed of rotating shaft, rpm
Q	Heating effects, Btu
$\overset{\text{RMS}}{\checkmark}$	Surface finish, (root-mean-squared)
S	Diameter of shaft
S_d	Stress due to groove depth, psi
S_{ic}	Increasing cross-section stress, psi
S_p	Stress due to differential pressure across O-ring seal, psi
S_{pc}	Peripheral compressive stress, psi
S_q	Stress due to heating at seal contact surface, psi
w	Width of O-ring groove, in.
W	Cross-sectional diameter of O-ring in free state, in.
W'	Increased cross-sectional diameter of O-ring due to peripheral compression in O-ring groove, in.
σ	Tensile stress, psi
ΔP	Differential pressure across the seal, psi
ϵ	Strain, in/in
η	Compressive stress factor; function of the speed of the rotating shaft

ACKNOWLEDGMENT

This is to acknowledge the technical assistance and support of Robert T. Simeral in the formulation of this report. The systematic procedure for designing a dynamic spindle seal as presented in this paper was derived from his 23 years of design experience.

INTRODUCTION

The solution to any seal problem depends on three major and interrelated variables: the operating conditions or environment, the mechanical design of the seal gland, and the seal material to be installed in this gland to prevent passage of the fluid being sealed. The various interrelations of these three variables account for the fact that there are so many different types of seals: felt radial seals, radial positive-contact seals, exclusion devices, clearance seals, split-ring seals, circumferential seals, axial-mechanical seals, simple compression packings, molded packings, diaphragm seals, static O-ring seals, dynamic O-ring seals, nonmetallic and metallic gaskets, and simple sealants (Ref. 1).

It is the object of this paper to describe those types of seals that readily apply to the ocean environment, both at sea level and at ocean depths. In the latter case, the extremes of low temperature, corrosive fluid, and great pressure require seal systems of maximum integrity and minimum maintenance. Three basic types of seals are used to exclude the ocean environment from an enclosed environment. For a submersible structure or vehicle, these three basic types of seals are: (1) linear seals, as on hatch covers and portals; (2) dynamic seals, such as rotary seals around propeller shafts and reciprocating seals on pistons; (3) general sealants around electrical cables and hoses that penetrate structural walls.

This report discusses these three basic types of seals. The discussion of linear seals presents the results of tests made on several elastomeric seal flanges and grooves of various configurations. The description of dynamic seals consists of a theoretical discussion of seal design related to seal function. The treatment of general sealants serves to introduce a description of a more economical method of solving seal[illegible] problems for limited-depth applications. Since the design of dynamic seals is the most critical problem, it has been given the greatest emphasis.

LINEAR SEALS

BASIC APPROACH

The basic solution of problems relating to linear-type seals has involved the use of rubber or elastomeric seals in grooves and flanges of various configurations. Most all hatch covers and portals open out (into the ocean environment), are hinged external to the structure, and thus take advantage of depth pressure to maintain positive squeeze on gasket type seals. Figure 1 shows a few configurations of hatch cover seals. The actual sealing material, whether rubber, elastomeric, or metal, must be compatible with the combined corrosiveness of seawater and all other metals and residues the seal may encounter. Long-term loading tests of seal systems (Ref. 2) and long-term deterioration tests of metals and elastomers in the deep ocean have been made (Refs. 3, 4, 5), but there is very little data available on the combined effects of material deterioration and long-term loading on seal system performance.

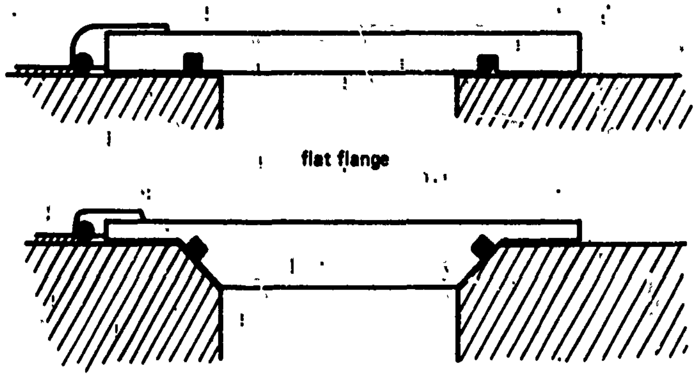


Figure 1. Hatch cover configurations.

In order to determine the combined effects of material deterioration and long-term loading on the performance of seals in the deep ocean, the Naval Civil Engineering Laboratory initiated a seal testing and evaluation program in November 1968 (Ref. 6). This program has completed three phases: (I) Mechanical Integrity of Flange Seal Systems, (II) Cyclic Loading of Flange and Hatch Seal Systems, and (III) Effects of Long-Term Hydrospace Exposure on Seal System Integrity. The following seven basic types of seal systems were chosen for evaluation because of their extensive use in shallow- and deep-diving submersibles, oceanographic instrumentation capsules and signaling devices, and internal pressure vessels:

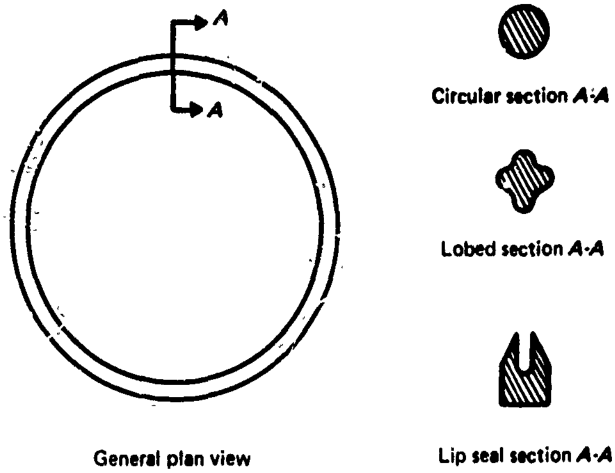
1. Conventional O-ring seal systems
2. Conventional O-ring seal systems with anti-extrusion devices, such as backup rings
3. Dovetail groove seal systems
4. Multiple lobed (quadring) seal systems
5. Multiple lobed (quadring) seal systems with anti-extrusion devices
6. Lip seal systems
7. Elliptical groove seal systems

Each of the first six types of seals was tested in both flat flange and angular flange configurations, while the elliptical groove was used in the angular flange seal only. The three basic types of seals are shown below.

EXPLICIT DATA

These basic types of seals in flat and angular flange configurations represent the majority of "hydrospace" seals. The findings of each of the three test phases are meaningful in the design of flanges and hatch-type seals:

I. Mechanical Integrity of Flange Seal Systems. All seal systems withstood hydrostatic pressures of 5,000 psi (11,250 ft depth in seawater) without leakage or seal



extrustion for pierods of 16 hours. The silicone rubber adhesive compound used to bond the base of the lip seal to the flange was "extruded to some degree in each test of this system. However, since the extrusion was in the direction of both the high- and low-pressure sides of the seal and was not accompanied by leakage, this extrusion was not considered to be an indication of failure of the seal system" (Ref. 7).

II. <u>Cyclic Loading of Flange and Hatch Seal Systems.</u> All flange and hatch seal systems withstood 20 test cycles of 16 hours at 5,000 psi, 8 hours at 0 psi in sodium chromate-inhibited fresh water with no leakage except for an O-ring in an elliptical groove. It was concluded that the elliptical-shaped groove is such that a standard O-ring is forced deep into the groove when submitted to high pressures. When the pressure is then reduced (cyclic loading), the O-ring can not recover its shape and, thus, the seal leaks at lower pressures.

III. <u>Effects of Long-Term Hydrospace Exposure on Seal System Integrity.</u> Twenty seal systems were exposed at a depth of 5,900 ft in the Pacific Ocean for 189 days. Seal systems utilizing carbon steel, 6061-T6 aluminum, or carbon steel with welded overlays of corrosion-resistant metals, *i.e.*, nickel-copper 400, nickel-molybdenum-chromium alloy "C" or nickel-chromium-molybdenum alloy 625 as flange faces were tested. The efficiency of galvanic protection was determined by attaching zinc anodes to a representative set of seal system test jigs. Five seal test jigs leaked: the carbon steel, angular flange, elliptical groove and O-ring seal with and without anodes; the carbon steel, angular flange, dovetail groove and lobed-ring seal with and without anodes; and the carbon steel, angular flange, dovetail groove, lobed-ring seal and backup ring. Because these "five seal test jigs showed no corrosion in the seal groove or seal mating surface under the seal, the failures were attributed to mechanical failure, insufficient design squeeze on O-rings and extrusion of backup rings, not failure from flange deterioration" (Ref. 8). Carbon steel flanges corroded more than the aluminum flanges, and corrosion was less when the crevice between the flanges was small and when galvanic protection was used. The seal flanges utilizing corrosion-resistant metal overlays were uncorroded after exposure.

IDEAL TYPES OF FLANGE SEALS

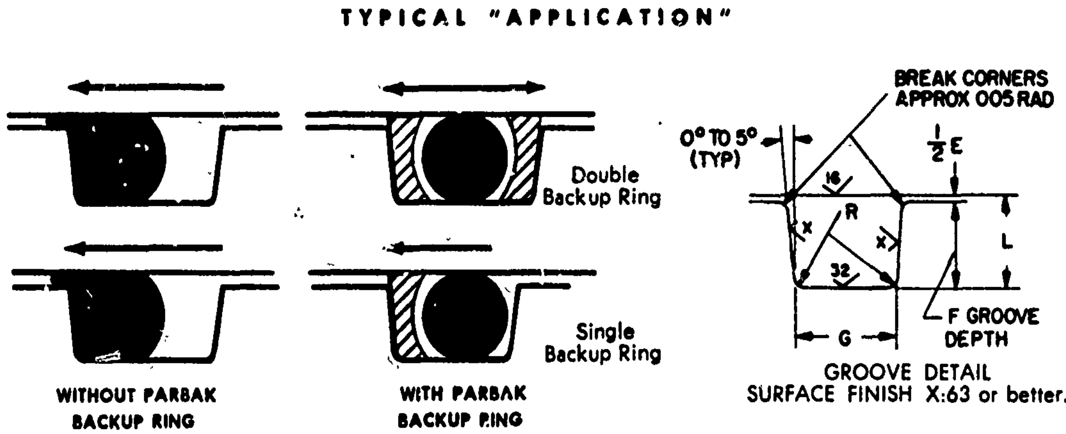
As indicated by data above, the ideal flange seal should reflect these design considerations:

1. Size of flange or hatch cover should be minimized to reduce amount of general corrosion and probability of mechanical leakage.
2. Corrosion-resistant metals or overlays should be used to eliminate corrosion at sealing faces.
3. Crevices between hatch covers and structure walls should be minimized to reduce corrosion at the flange.
4. Galvanic protection should be used to reduce corrosion.
5. The use of standard O-ring seals in elliptical grooves should be avoided, especially when cyclic loading is involved.
6. The use of lobed-ring seals in dovetail grooves should be avoided.
7. Backup rings that must be fitted to specific configurations and/or spliced should not be used.
8. Lip seals that must be bonded to the base flange, especially in high-pressure applications, should not be used.

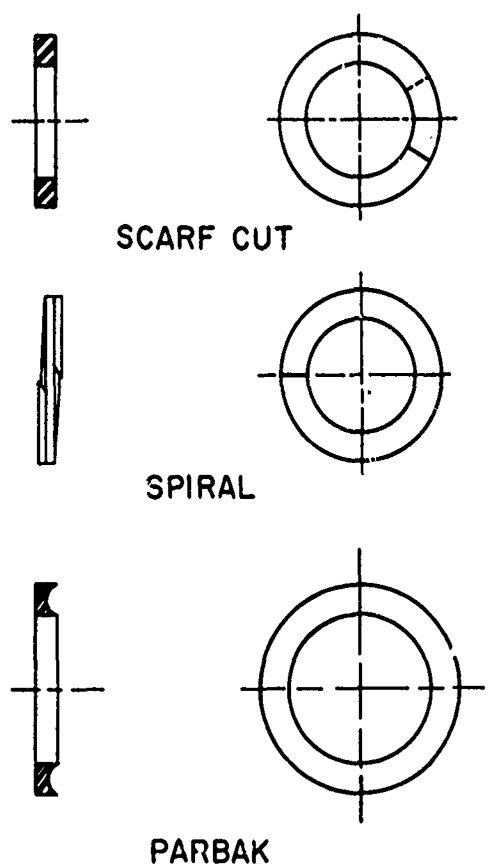
The two most important design parameters to consider when designing any type of external static seal as used in hatch covers are pressure and corrosion. Proper groove design and seal installation in accordance with the many available seal manuals (Refs. 9, 10, 11) will prevent seal extrusion by depth pressure, while proper selection of galvanically compatible materials will minimize corrosion.

Figure 2 shows a typical application of backup rings to prevent O-ring extrusion; but from the data noted, it is seen that backup rings are not advantageous in applications where the seal is continually broken and resealed (opening and closing of hatch covers). In order to avoid the use of backup rings and still maintain a high-pressure seal, the usual procedure used at the Naval Undersea Center is to reduce the groove angle recommended (from 0 to 5 deg) to 0 to ±½ deg and reduce the clearance, E, to the minimum possible. Reducing what would be the clearance between the flange faces also reduces corrosion. Reducing the size of crevices limits the flow of new oxygenated seawater to potentially corrodible metals. Initial corrosion in a small gap soon depletes the dissolved oxygen in the seawater within the crevice, and this oxygen is not replaced due to the slow diffusion of oxygen-rich seawater into the crevice from outside (Ref. 8, p. 5). The oxygen-poor seawater is not so aggressive toward steel and aluminum as the oxygen-rich seawater outside (Ref. 3). Angular flange seals are more susceptible to crevice corrosion than flat flange seals, because it is much more difficult to hold close tolerances on larger surface areas.

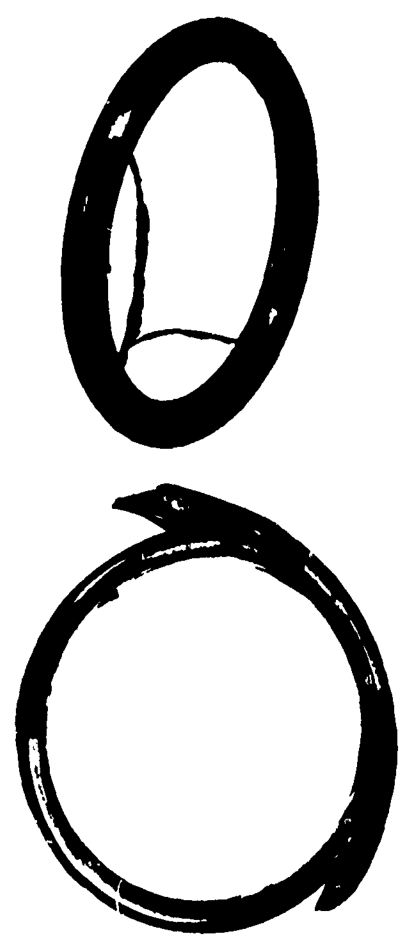
Figure 3 shows the flat and angular flange-type of hatch cover seals, while Fig. 4 shows flanges made of noncorrosive overlays, such as nickel-chromium-molybdenum alloys. Dimensions are shown in order to depict relative sizes of important features.



(a) Parbak backup rings



(b) Typical backup rings



(c) Result of O-ring extrusion

Figure 2. Application of backup rings. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

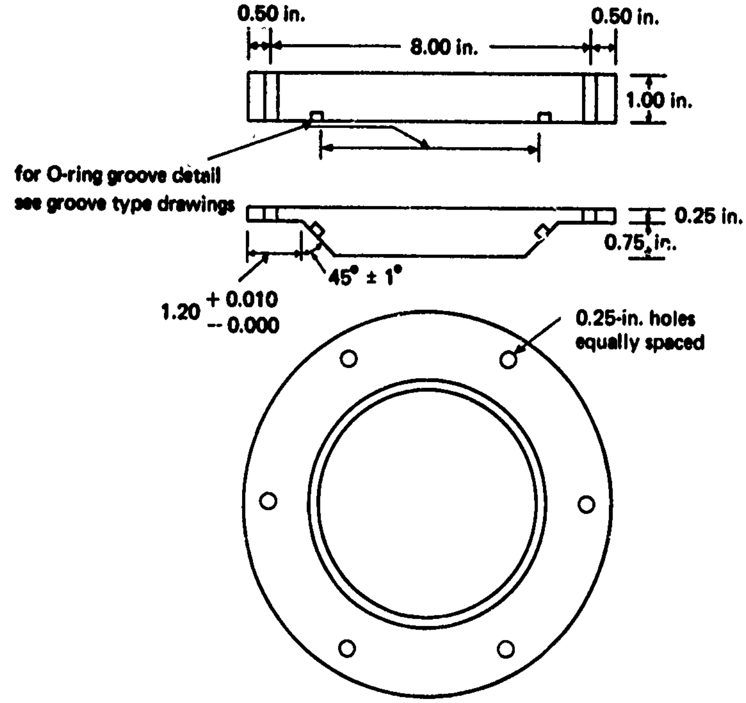


Figure 3 Flat and angular flanges.

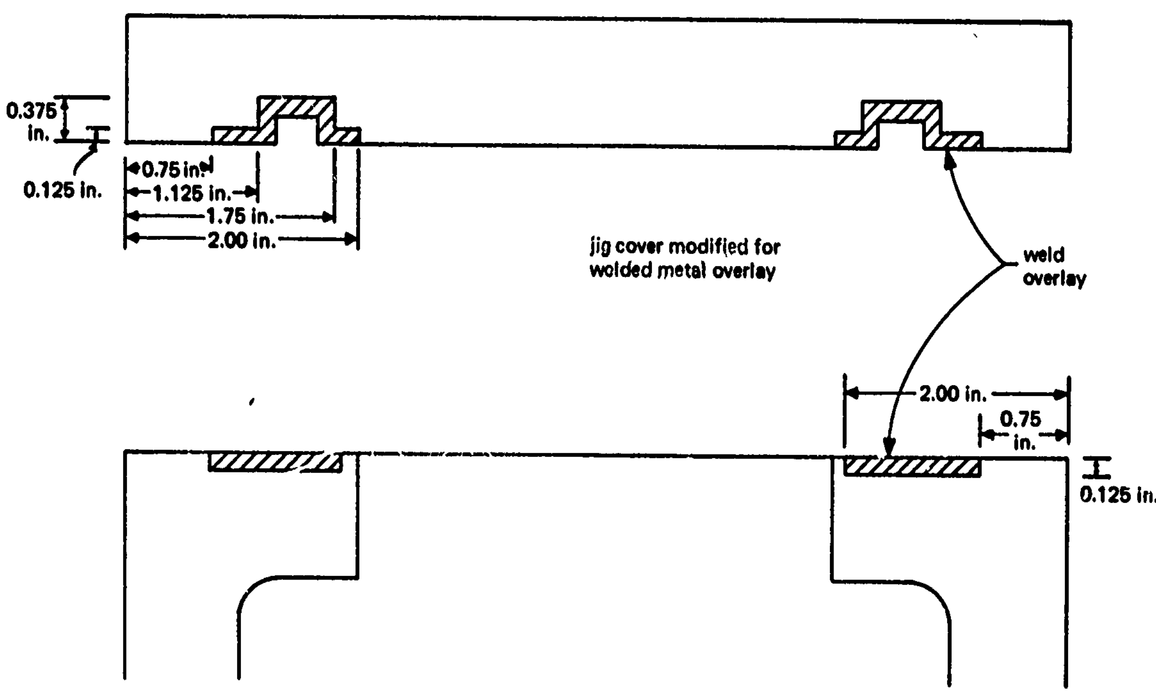


Figure 4. Jig bottom modified for welded metal overlay

METAL SEALS – "A NEW APPROACH"

Metallic gaskets and metal O-rings have always been used in high-pressure, permanently sealed (non-cyclic) applications, and many types are available (Ref. 1, pp. 90–101). The recent approach in deep underwater vehicles is to make the metallic sealing principle inherent in the design of hatch covers. On the deepest diving U. S. submarine, the *Dolphin*, the main hatch cover (actually a main door) has an angular type flange with highly polished mating surfaces. Positive seal is maintained by slight elastic deformation of these angular surfaces by depth pressure, very much like a conical cork in a bottle. Of course, this type of hatch cover or flange seal is very expensive, and more care must be taken in its use and maintenance than is required with other types.

MOVING OR DYNAMIC SEALS

THEORY OF O-RING ROTARY SEALING APPLICATIONS

Once it was considered impossible to use O-rings in a seal for a rotating shaft except at very low rotational speeds. This inapplicability of O-rings to rotary motion was blamed on the Gow-Joule effect. According to this phenomenon, when rubber is stretched, then heated, it tries to contract, and its modulus of elasticity and, thus, stiffness or ability to carry load increases with rise in temperature. If the rubber is under constant load, it will contract; if under constant strain, it will exert greater stress.

When an O-ring is used in the usual way, by stretching slightly around the shaft, the friction of the rubber against the rotating shaft generates heat, which causes the rubber to contract about the shaft. As the rubber O-ring contracts about the shaft, unit loading at the shaft–O-ring sealing surface increases; thus, more friction and heat is generated, and the Gow-Joule effect intensifies. The cycle of friction, heat, and contraction of the ring is repeated until rapid failure of the seal occurs.

If the O-ring has peripheral tensile stresses due to its being stretched over the shaft, failure will occur in a few minutes at shaft speeds above 200 fpm (Ref. 9). The differential pressure across the O-ring seal also increases the unit loading at the sealing surface and will cause premature failures at even slower shaft speeds. The amount of tensile stress allowable for a rubber seal of any type that rubs directly on a shaft depends upon contact area, differential pressure across the seal, unit loading, surface finish of the shaft, and the properties of the rubber material used. A design formula for the relationship between the peripheral tensile stress and the environmental conditions would be of the form:

$$\sigma = \epsilon E$$

$$\sigma = f(A, \Delta P, L, \overset{\mathrm{RMS}}{\checkmark})E$$

Notice that the peripheral tensile stress, σ, on the rubber O-ring will be reduced if the contact area and the unit loading caused by the expansion of the ring on the shaft are reduced. If the reduction of the two latter factors is great enough, the peripheral tensile stress can be alleviated to the point of peripheral compression; that is, the unit load becomes compressive ($-L$) instead of tensile. If the net forces on an O-ring result in peripheral compression even under the severest environmental loads, the Gow-Joule effect will not occur. This is because the Gow-Joule effect occurs only if the rubber O-ring is under tensile stress (Ref. 10). Therefore, a rotary seal will operate if the induced peripheral compressive stress is always greater than tensile stresses induced by differential pressure, ΔP, and the unit loading, L.

Figure 5 shows how a rotary shaft seal can be designed to induce peripheral compression on the O-ring. The dimensions of the O-ring groove and seal housing are such that when an oversized O-ring is installed, the O-ring is forced against the bottom (maximum depth) of the groove, away from the shaft, as in Fig. 5d. For efficient running conditions, the O-ring should have about 5 percent peripheral compression, $[(\mathrm{OD} - G)/\mathrm{OD}] \times 100\%$, and a cross-sectional squeeze usually between 2 and 4 percent when the shaft is installed, $[(W' - D)/W'] \times 100\%$. When the O-ring is installed in the groove (Fig. 5d), the cross-sectional area of the O-ring increases because its outer diameter is decreased by an amount $\mathrm{OD} - G$. The O-ring tends to "snake" in the groove, a phenomenon opposing the increase in cross-sectional area. If the amount of "snake" is excessive, the cross-sectional area of the O-ring will not increase sufficiently to produce the desired amount of peripheral compression. Therefore, the groove width, w, is usually kept to a minimum. Since the groove width must allow for swell caused by fluid adsorption and thermal expansion caused by heating, there is an allowable minimum value of w such that the volume of the groove is at least 5 percent greater than the volume of the O-ring seal (Ref. 11, pp. 3-9).

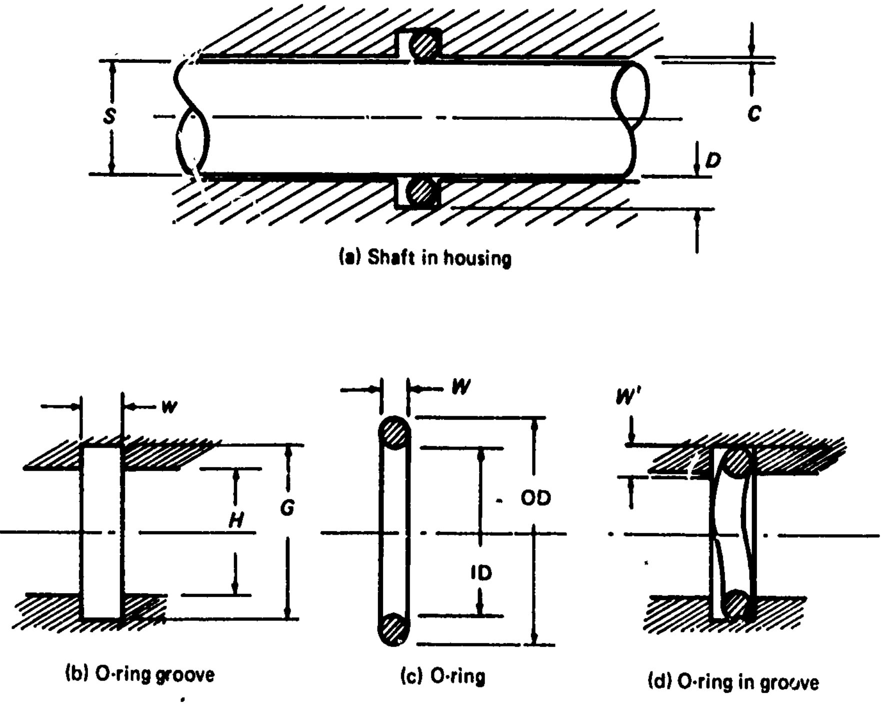


Figure 5 Rotary shaft seal.

The environmental parameters upon which each of the groove dimensions depends can best be defined by a relationship table (Table 1) and accompanying example (Fig. 6). The example problem represents an actual design developed by the Naval Undersea Research and Development Center. It is given here to show how the relationships in Table 1 apply to each other. For instance, the clearance between the shaft and seal housing, C, should be small enough to insure against O-ring extrusion by differential pressure but large enough to allow the seal housing to "float" and take up eccentricities without contacting the shaft. The clearance is determined from non-extrusion data like those shown in Fig. 7. The diametral clearance in Fig. 7 equals twice the dimension shown in Fig. 6. In the design example, the clearance chosen was slightly greater than that given in Fig. 7., *i.e.*, $2C = 0.0050$ instead of 0.0038, thus increasing the effectiveness of the floating housing.

Table 1. Relationships Between Dimensions and Environmental Parameters.

Dimension	Environmental Parameter	Interrelationships and Functions
Seal housing		
C	Differential pressure, ΔP	$C \propto \frac{1}{(\Delta P)^n}$ (Fig. 7)
H	Shaft size, S Differential pressure, ΔP	$H = S + 2C$
D	O-ring cross-sectional diameter, W Shaft speed, N	$D = \eta W$ $\eta \propto N^n$, $\eta = 0.90$ to 0.99
G	Shaft size, S Shaft speed, N	$G = S + 2D$
O-ring		
OD	Shaft size, S	$OD = fG$
ID	Shaft speed, N	$f = \left(1 + \frac{\text{\% diametral reduction}}{100\%}\right)$ $ID = OD - 2W$
W	Shaft speed, N	$W \propto \frac{1}{N^n}$, design chart, or minimum W'
W'	Shaft speed, N	$W' = IW_{min}$ $I = \left(1 + \frac{\text{\% increase in cross section}}{100\%}\right)$
f and I	Shaft speed, N Differential pressure, ΔP	Figure 9, % diametral reduction vs. % increase in cross section

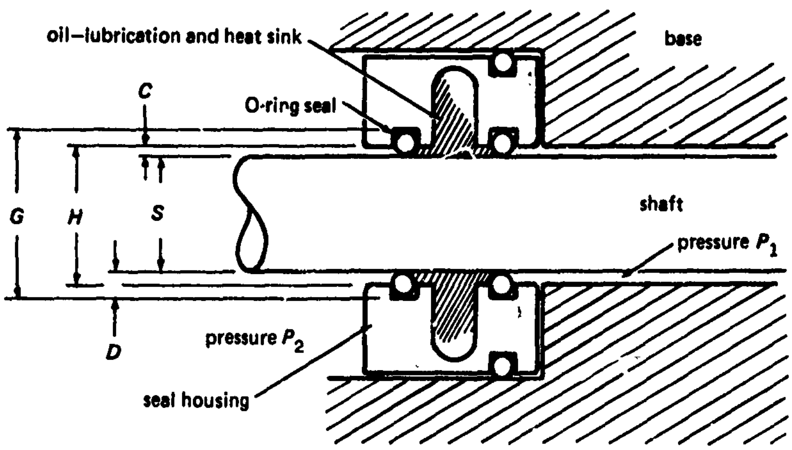


Figure 6. Spindle design problem–rotary O-ring.

Environmental Parameters

Given: shaft diameter, $S = 2.0625 \pm 0.0005$
shaft speed, $N = 2{,}200$ rpm
differential pressure $= \Delta P = P_2 - P_1 = 2{,}000$ psi

Component Dimensions

To determine:

1. C, clearance
2. H, housing diameter (± 0.0005)
3. D, groove depth
4. G, groove diameter (± 0.001)
5. O-ring size required, OD and ID
6. O-ring cross-sectional increase in groove
7. Actual squeeze of O-ring

CALCULATIONS

1. C, clearance: make minimum to insure against O-ring extrusion by differential pressure, ΔP. See Fig. 7.

   $C = 0.0025$ nominal

2. H, housing diameter $= S + 2C$

   $H = 2.0625 + 2(0.0025) = 2.0675 \pm 0.0005$ diameter

3. D, groove depth: $D \approx (0.90 \text{ to } 0.99)W$, where W = O-ring cross-sectional diameter. Select smallest W to provide least sealing area. Use a factor of 0.95 because shaft speed is high, 1,150 fpm. Therefore,

$$D = (0.95)\,(0.070 \pm 0.003) = 0.0665 \text{ nominal depth}$$

4. G, groove diameter $= S + 2D$

$$G = 2.0625 + (2 \times 0.0665) = 2.196 \pm 0.001 \text{ diameter}$$

5. O-ring size required:

(a) $\text{OD} = f \times G$, where the factor

$$f = \left(1 + \frac{\%\text{ of diametral reduction}}{100\%}\right)$$

Usually $f = 1 + 0.08 = 1.08$, therefore

$$\text{OD} = 1.08\,(2.196) = 2.372 \text{ diameter}$$

(b) $\text{ID} = \text{OD} - 2W$

$$\text{ID} = 2.372 - 2\,(0.070) = 2.232 \text{ diameter}$$

Therefore use nearest size O-ring

$$\text{ID} = 2.239 \text{ by } W = 0.073 \pm 0.003 \text{ Parker No. 2-35}$$

6. O-ring cross-sectional increase in groove:

(a) $W'_{min} = I \times W_{min}$, where the factor

$$I = \left(1 + \frac{\%\text{ increase in cross section}}{100\%}\right)$$

Usually $I = 1.022$, therefore

$$W'_{min} = 1.022\,(0.070 - 0.003) = 0.0685$$

(b) $W'_{max} = 1.022\,(0.070 + 0.003) = 0.0746$

7. Actual squeeze of O-ring:

(a) $\text{Minimum} = W'_{min} - \left(\frac{G_{max} - S_{min}}{2}\right)$

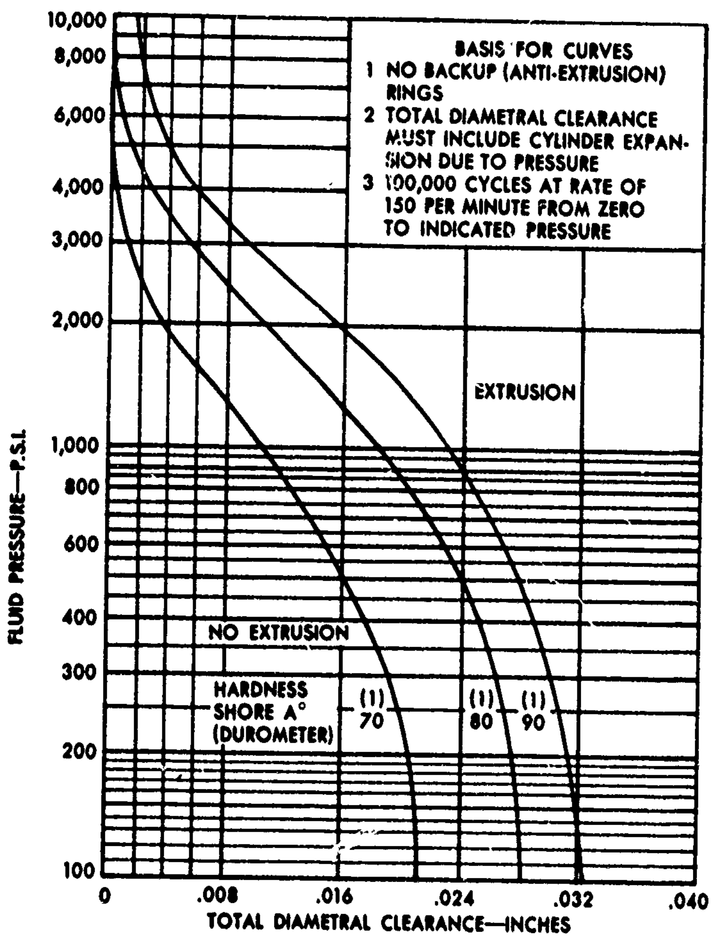
$$\text{Minimum} = 0.0685 - \left(\frac{2.197 - 2.062}{2}\right)$$

$$\text{Minimum squeeze} = 0.001$$

$$\text{(b) Maximum} = W'_{\max} - \left(\frac{G_{\min} - S_{\max}}{2}\right)$$

$$\text{Maximum} = 0.0746 - \left(\frac{2.195 - 2.063}{2}\right)$$

$$\text{Maximum squeeze} = 0.0086$$



Curves from "O-Ring Seals in the Design of Hydraulic Mechanisms" by D. R. Pearlman of Hamilton Standard, Propeller Division, United Aircraft. Paper presented at SAE Annual Meeting, January, 1947.

Figure 7. Non-extrusion data. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

The groove depth is a function of the cross-sectional diameter of the O-ring chosen to provide the least sealing area around the shaft. The cross-sectional diameter or O-ring width, *W*, can be selected according to shaft speed, as listed in Ref. 10 (see Fig. 8 and Table 2); but notice this table is only for differential pressures up to 800 psi. As differential pressure across an O-ring increases, the O-ring deforms more and the sealing area around the shaft increases. Therefore, the minimum O-ring width should be used for high differential pressures. The O-ring width is then multiplied by a compressive stress factor, η, to determine the groove depth, *D*. This factor determines the amount of compressive stress induced within the O-ring and is a function of the shaft speed. The greater the shaft speed, the greater this factor should be, because high shaft speeds develop more heat at the sealing surface area, and thus more heat must be removed by the lubricant. This is discussed more fully under "Heating and Lubrication," p. 20.

Parts 5 and 6 of the calculations in Fig. 6 contain factors *f* and *I* that are related to the compressive stress factor, η, used in Part 3. The theoretical interrelationship of *f* and *I* is depicted in Fig. 9. The equations used to determine this plot appear in the appendix. Thus, as indicated in Fig. 9, an 8-percent reduction in O-ring diameter would result in a cross-sectional diameter increase of between 2 and 4 percent. The factor used in the problem contains a 2.2-percent increase in cross-sectional diameter, the probable minimum O-ring width when installed in the groove. Using the minimum percentage increase in cross-sectional diameter insures that the O-ring selected will provide a positive minimum squeeze even when the shaft is not rotating, although the actual squeeze should be checked as was calculated in Part 7 of the design problem (Fig. 6).

The determination of the most efficient *I*-factor (percent increase in cross-sectional diameter) depends mainly on one environmental factor, *i.e.*, the amount of heat encountered by the O-ring seal. Heat is generated at the sealing surface of the rotating shaft; it may be transferred from the fluid being sealed, or from closely located bearings that generate heat, and also through the base or seal housing, depending on the application. The heat generated at the sealing surface of the rotating shaft is usually the most critical since it is caused by localized friction produced by relative motion and intensified by differential pressure forces across the O-ring seal. The most efficient *I*-factor (percent increase in cross-sectional diameter) must result in a seal with the maximum amount of peripheral compression to oppose tensile stresses, which give rise to the Gow-Joule effect. These relationships are shown in Figs. 10a and 10b, which depict the stresses applied to the rotary O-ring and the resulting cumulative stresses within the O-ring. The applied stresses must be balanced by the resultant stresses within the O-ring. The magnitude and direction of the stress are respectively depicted by the length and orientation of the vectors.

Figure 10a is a three-dimensional sketch of an O-ring in peripheral compression. The peripheral compression acts circumferentially within the O-ring to oppose the externally applied radial forces caused by the O-ring groove. Therefore, the internal compressive (peripheral) forces shown acting on the cross-sectional plane, $(F_c)_3/2$ and $(F_c)_4/2$, oppose half of the externally applied radial forces $(F_r)_3$ and $(F_r)_4$, respectively. Even though the cross-sectional plane represents an infinitesimal section of the O-ring, the entire O-ring volume is in peripheral compression. The peripheral compressive forces, or stress, induce

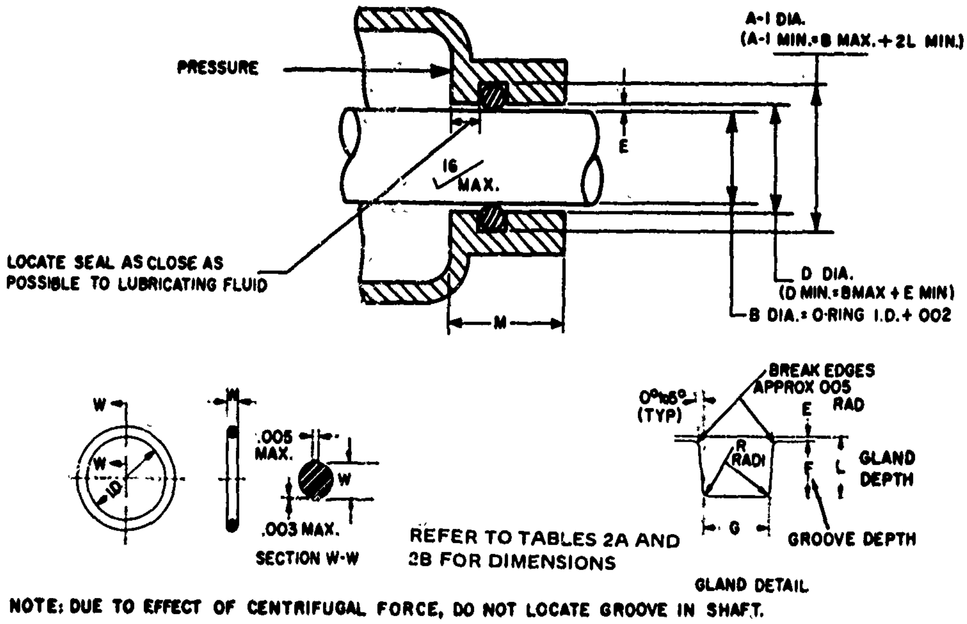


Figure 8. Rotary O-ring seal gland. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

Table 2a. Design chart for rotary O-ring seal glands, 800 psi max[c]. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

O-Ring Size Parker No. 2-	W Cross Section		Maximum Speed fpm (a)	L Gland Depth	G Groove Width	E(c) Diametrical Clearance	Eccentricity Max (b)	M Bearing Length Min. (c)	R Groove Radius
	Nominal	Actual							
004 through 045	1/16	0.070 ±.003	1500	0.065 to 0.067	0.075 to 0.079	0.012 to 0.016	0.002	0.700	0.005 to 0.015
110 through 163	3/32	0.103 ±.003	600	0.097 to 0.099	0.108 to 0.112	0.012 to 0.016	0.002	1.030	0.005 to 0.015
210 through 258	1/8	0.139 ±.004	400	0.133 to 0.135	0.144 to 0.148	0.016 to 0.020	0.003	1.390	0.010 to 0.025

(a) Feet per minute = 0.26 × Shaft Diameter (inches) × rpm.
(b) Total indicator reading between groove OD, shaft, and adjacent bearing surface.
(c) If clearance (extrusion gap) must be reduced for higher pressures, Bearing Length M must be no less than the minimum figures given. Clearances given are based on the use of 80 Shore Durometer minimum O-ring for 800 psi max.

Table 2b. Rotary O-ring seal gland dimensions, 800 psi max†. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

O-ring size	Dimensions				B	A-1	D	G
					OD (shaft)	Groove diameter (gland)	Throat diameter	Groove width
Parker No. 2-	ID	±	W ±.003	Mean OD (Ref)	+.000 −.001	+.003 −.000	+.003 −.000	+.004 −.000
001								
002								
003								
004	0.070	0.004		0.210	0.072	0.202	0.084	
005	.101	.004		.241	.103	.233	.115	
006	.114	.005		.254	.116	.246	.128	
007	.145	.005		.285	.147	.277	.159	
008	.176	.005		.316	.178	.308	.190	
009	.208	.005		.348	.210	.340	.222	
010	.239	.005		.379	.241	.371	.253	
011	.301	.005		.441	.303	.433	.315	
012	.364	.005		.504	.366	.496	.378	
013	.426	.005		.566	.428	.558	.440	
014	.489	.005		.629	.491	.621	.503	
015	.551	.005		.691	.553	.683	.565	
016	.614	.005		.754	.616	.746	.628	
017	.676	.005		.816	.678	.808	.690	
018	.739	.005		.879	.741	.871	.753	
019	.801	.005		.941	.803	.933	.815	
020	.864	.006	0.070	1.004	.866	.996	.878	0.075
021	.926	.006		1.066	.928	1.058	.940	
022	.989	.006		1.129	.991	1.121	1.003	
023	1.051	.006		1.191	1.053	1.183	1.065	
024	1.114	.006		1.254	1.116	1.246	1.128	
025	1.176	.006		1.316	1.178	1.308	1.190	
026	1.239	.006		1.379	1.241	1.371	1.253	
027	1.301	.006		1.441	1.303	1.433	1.315	
028	1.364	.006		1.504	1.366	1.496	1.378	
029	1.489	.010		1.629	1.491	1.621	1.503	
030	1.614	.010		1.754	1.616	1.746	1.628	
031	1.739	.010		1.879	1.741	1.871	1.753	
032	1.864	.010		2.004	1.866	1.996	1.878	
033	1.989	.010		2.129	1.991	2.121	2.003	
034	2.114	.010		2.254	2.116	2.246	2.128	
035	2.239	.010		2.379	2.241	2.371	2.253	
036	2.364	.010		2.504	2.366	2.496	2.378	
037	2.489	.010		2.629	2.491	2.621	2.503	
038	2.614	.010		2.754	2.616	2.746	2.628	
039	2.739	.015		2.879	2.741	2.871	2.753	
040	2.864	.015		3.004	2.866	2.996	2.878	

†For pressure over 800 psi consult Table 2a and the Design sections of Parker Seal Co. *O-Ring Handbook*.

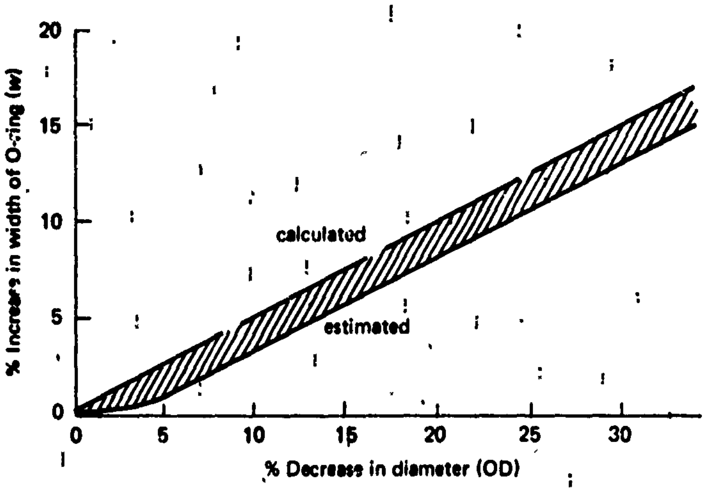


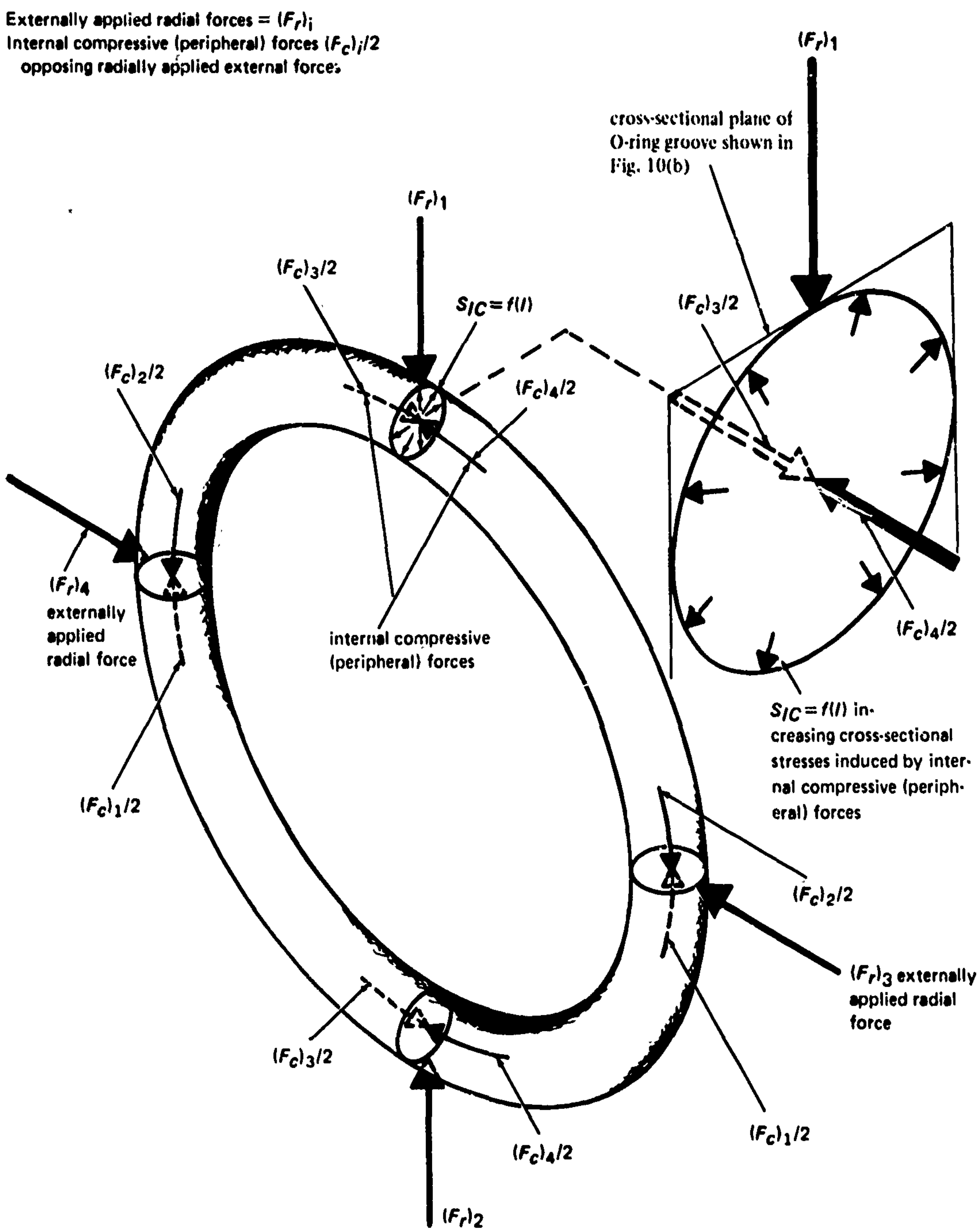
Figure 9. The theoretical relationship of f and I.

cross-sectional stress, S_{Ic}, in the plane. These induced cross-sectional stresses actually increase the cross-sectional area of the O-ring (phenomenon of Poisson's ratio).

Figure 10b is an elaboration of the applied and induced stresses shown in the cross-sectional plane of the O-ring depicted in Fig. 10a. The first views show the indirectly applied peripheral compressive stress as a function of the percent of diametral reduction, f. Applying this stress causes an increase in cross-sectional area (shown in the adjacent view) and a corresponding resultant stress throughout, dependent on I. (The relationship between f and I was shown in Fig. 9.) The second set of views shows the shaft installed, cumulative stresses after the addition of the stress resulting from the depth of the groove being about 95 percent of the original O-ring width, and the consequent internal stresses applied by the O-ring. The third set of views shows the application and resultant stresses of differential pressure, *i.e.*, the function of the O-ring to seal a fluid under pressure. The stresses resulting from both groove depth and differential pressure are equally distributed inside the O-ring.

This equalized distribution follows from the fact that a rubber seal is considered to be an incompressible, viscous fluid having a very high surface tension (Ref. 11). Although these two stresses are distributed evenly within the O-ring seal, they act to flatten the O-ring against the shaft, thereby increasing the contact area of the seal and the unit loading on the shaft. This increase in unit loading increases the frictional forces as the shaft rotates and would produce the Gow-Joule effect if not compensated by the opposing stress (S_{Ic}) caused by increased cross section.

The Gow-Joule effect is shown in the last set of views only to depict what the S_{Ic} opposes and to indicate that heating effects, Q, are a function of the differential pressure, groove depth, and shaft speed. It must be remembered that as long as the net stresses inside the O-ring are opposed by the initially induced peripheral compressive stress, S_{pc}, the Gow-Joule heating effect cannot occur and the O-ring will not even start to contract around the rotating shaft.



Note: The internal compressive (peripheral) forces at the plane oppose the external radial forces applied 90° from the plane shown, *i.e.*, $(F_c)_3/2$ opposes $(F_r)_3$.

Figure 10(a). O-ring in peripheral compression.

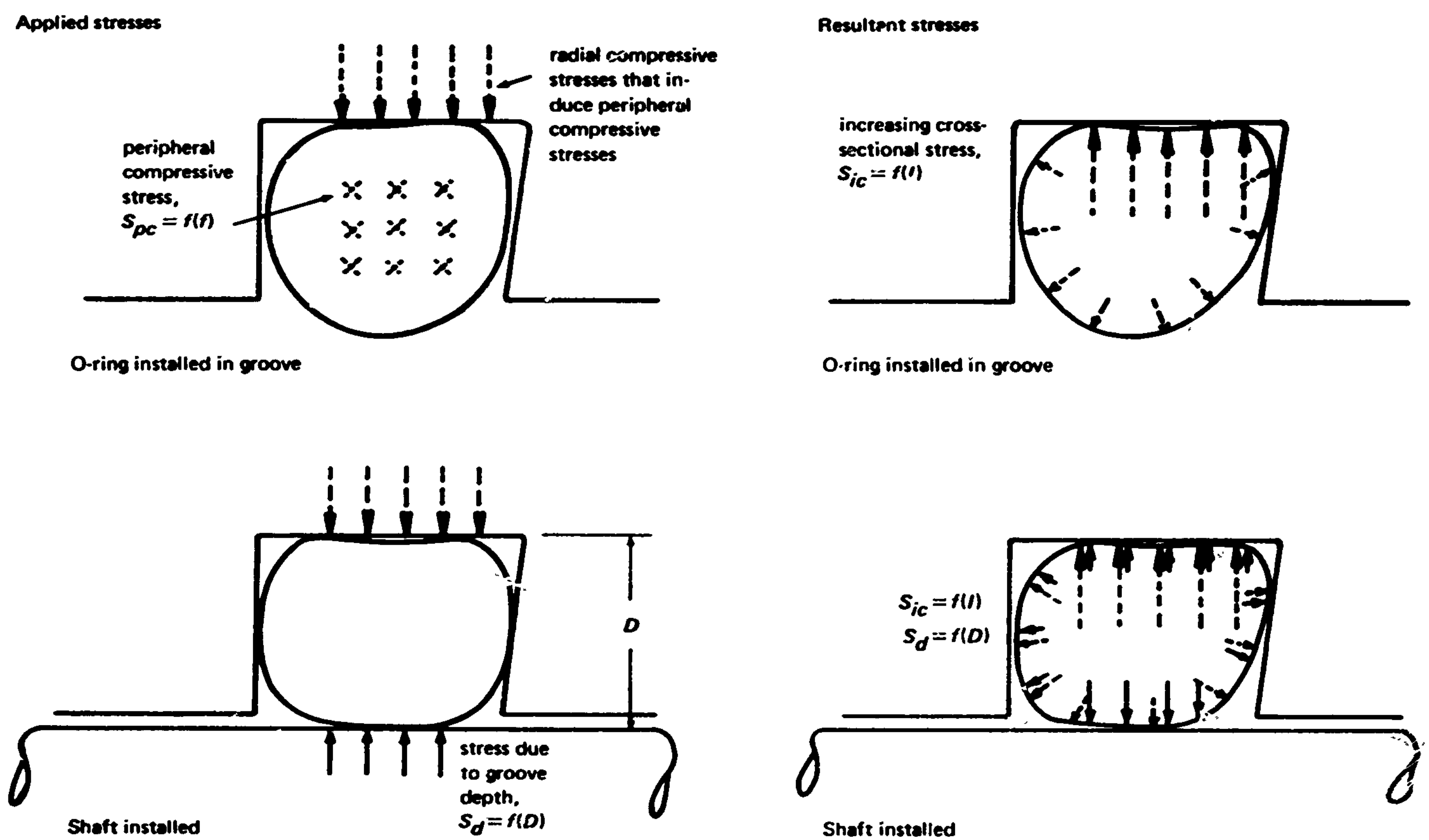


Figure 10(b). Rotary O-ring seal--design stresses.

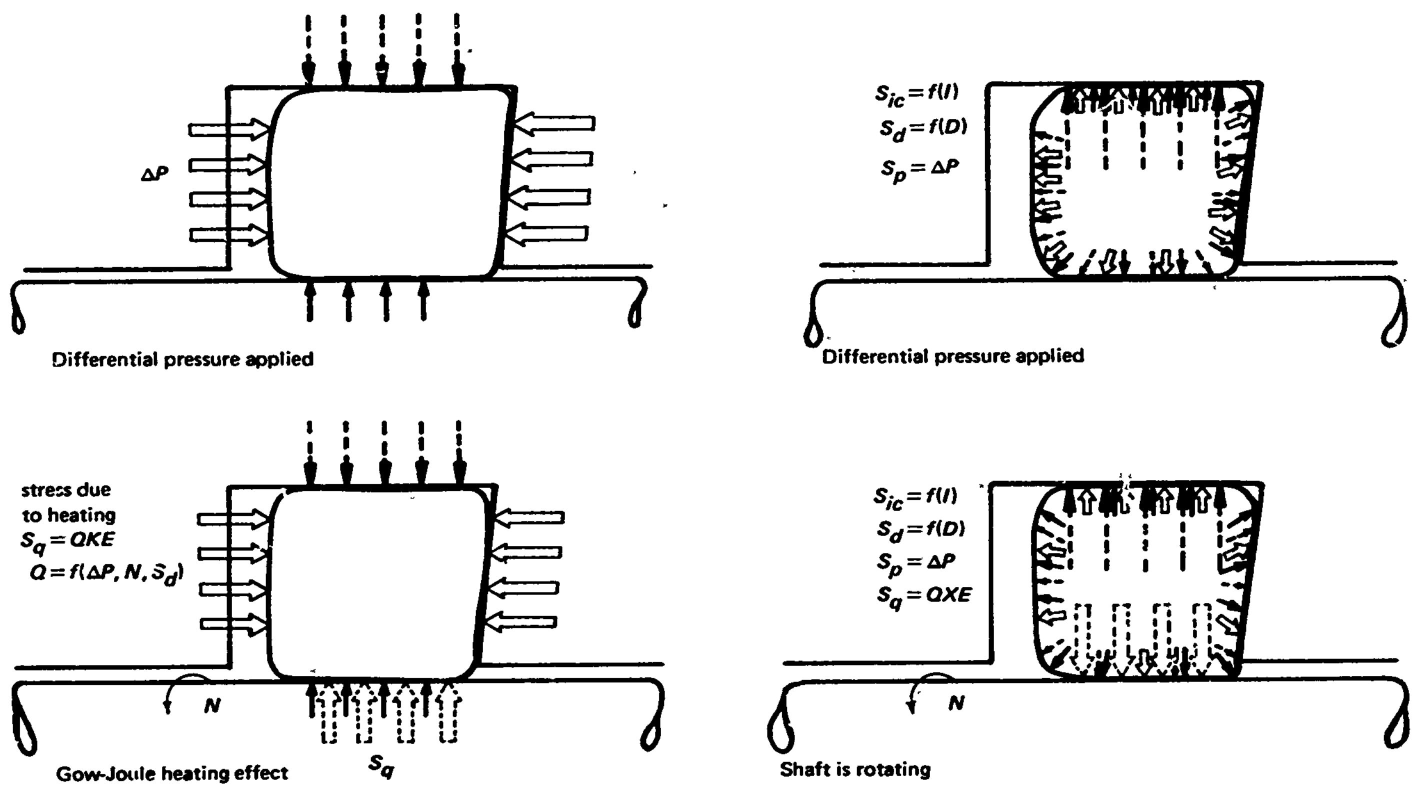


Figure 10(b) (Contd.)

In the last view, it becomes obvious that the S_{ic} stresses should be made maximum to oppose the stresses that would otherwise give rise to the Gow-Joule effect. In order to obtain this maximum cross-sectional stress, the maximum amount of peripheral compression, f, must be applied to the O-ring. Theoretically, this can be accomplished by pushing an O-ring with the largest possible outside diameter into the groove. The limiting factor is the mechanical difficulty in compressing a large O-ring into a small groove without detrimental effects, such as "snaking," scratching, etc. Realistic maximum values of f vary from 1.07 to 1.10, meaning 7 to 10 percent of diametral reduction. This means the maximum possible percentage increase in cross-sectional width is 5 percent (see Fig. 9).

HEATING AND LUBRICATION

As noted, the amount of heat generated at the sealing surface of the moving shaft and stationary O-ring is increased by an increase in O-ring squeeze, *i.e.*, a decrease in the groove depth, D, (see Fig. 6). It was also noted that an increase in differential pressure applied across the O-ring increases the amount of heat generated. This heat must be dissipated to prevent overheating of the O-ring and the metal surface of the rotating shaft. This is accomplished through lubrication. The lubricant shown in the annulus of the spindle design (Fig. 6) thus provides a heat sink, while also reducing that portion of heat generated by normal asperity interference.

The importance of lubrication, especially in our application of high shaft speed and high differential pressure, cannot be over emphasized. The extreme pressure (EP) oil used in the rotary seal shown in Fig. 6 is a special formula of "Wynn Oil." High-load-carrying films from EP oils have a "cushioning effect" on the normal stresses of asperities. This is accomplished by the EP oil maintaining a sufficiently thick solid-like boundary film, which has a lower elastic modulus than the asperities that this film separates (Ref. 12).

In addition to mechanically separating asperities between bearing surfaces, boundary lubricants function through chemical phenomena. There are two main chemical reactions that must be controlled in order to optimize the benefits from a boundary lubricant: (1) beneficial decrease in fracture properties of metal surfaces; (2) detrimental increase in corrosive wear. A properly balanced boundary lubricant contains oxygen, water, and load-carrying additives that control the amount or rate of conversion of bearing metals to friable corrosion products. These friable products will break into smaller wear particles before causing extensive damage to the confining surfaces. The detrimental effect occurs when too much friability leads to corrosive wear. Therefore, a trade-off of boundary film properties is required (Ref. 12).

RECIPROCATING SEALS

Ocean systems very frequently incorporate the use of reciprocating hydraulic, pneumatic, and other fluid-actuated types of systems, such as pistons, sliding flanges, extension devices, etc. Rubber and elastomeric O-ring seals are most successfully used in these applications because they are inexpensive and reliable if used properly. The following discussion is presented as an aid in avoiding their misuse. Again, the discussion will consider how the ocean environment and specific application affects the design parameters of reciprocating seals, and the interrelationship of failure and cause of failure.

Applications and Design Parameters

Extreme depth pressure and corrosion are the most critical aspects of the ocean environment affecting the design parameters of reciprocating seals. Extreme pressure requires minimum clearance between the O-ring housing and piston shaft to insure against O-ring extrusion (as seen in rotary seal applications). Side loads on a piston or rod can cause the clearance in the O-ring gland to be on one side only. If adequate O-ring squeeze has not been provided, leakage will result, and if excessive clearance is created, extrusion of the O-ring may result. High side loading on a piston will cause uneven friction on the seal, and if high enough, the rod or barrel will be galled or scored. Shock pressures, such as created by sudden stopping of a hydraulic damping cylinder, are many times greater than the actuation pressures required for normal use and must be considered as critical design parameters. In many applications, a mechanical lock or brake should be provided to reduce and/or take up shock loads, especially after the final piston position (relative to the cylinder) has been attained.

Corrosion and related contamination by sand, silt, and other sediments can be very detrimental to the sealing efficiency of O-rings in underwater hydraulic cylinder applications. Equipment having rods exposed to this hostile environment during operating cycles should be fitted with scraper and/or wiper rings which prevent dirt and corrosive products from reaching the O-ring seal and seal housing. To reduce galvanic corrosion, the usual type of bearing materials (babbit, bronze, etc.) that most often are dissimilar to the major types of structural materials (steel, stainless steel, aluminum, etc.), can be replaced by inert materials, such as nylon and teflon. Designers using such polymeric materials must consider and allow for relatively high coefficients of thermal expansion and, in the case of teflon, cold flow.

In most piston/cylinder applications, the O-ring groove is machined into the male element (piston), since it is usually an easier process and the complications involved in design of a groove in the female part (peripheral compression, as already discussed under the theory of rotary seals) are eliminated. There are applications, however, where it is functionally more efficient to put the groove in the cylindrical part instead of the piston. If the frictional force of the moving metal surface across the O-ring is in the same direction as the direction of differential pressure, the O-ring will tend to be dragged into the gap more easily and thus extrude at a much lower (30 to 40 percent) than normal pressure (Ref. 10, pp. 6-9). By placing the groove in the opposite metal part, the friction will work against pressure and reduce the susceptibility to "spiral failure."

Failure and Causes

Spiral failure is a unique type that sometimes occurs on reciprocating O-rings. This type of failure appears as a spiral or corkscrew cut halfway through the O-ring cross section. The seal has the appearance of being twisted while being cut with a knife. This type of failure is shown in Fig. 2c.

A properly used O-ring slides during all but a small fraction of any reciprocating stroke and does not normally tend to twist or roll. Figure 11 shows why a properly used O-ring does not tend to roll or twist: (1) the differential pressure across the O-ring, produces a holding force within the groove, due to friction on a larger area, greater than the pulling force produced by the sliding surface (rod or cylinder wall) opposite the groove; (2) the surface finish of the sliding surface is made smoother than that of the groove in order to reduce friction at the sliding surface; (3) running friction between moving parts is always lower than the breakout friction between non-moving parts; (4) the torsional resistance of the O-ring tends to resist twisting.

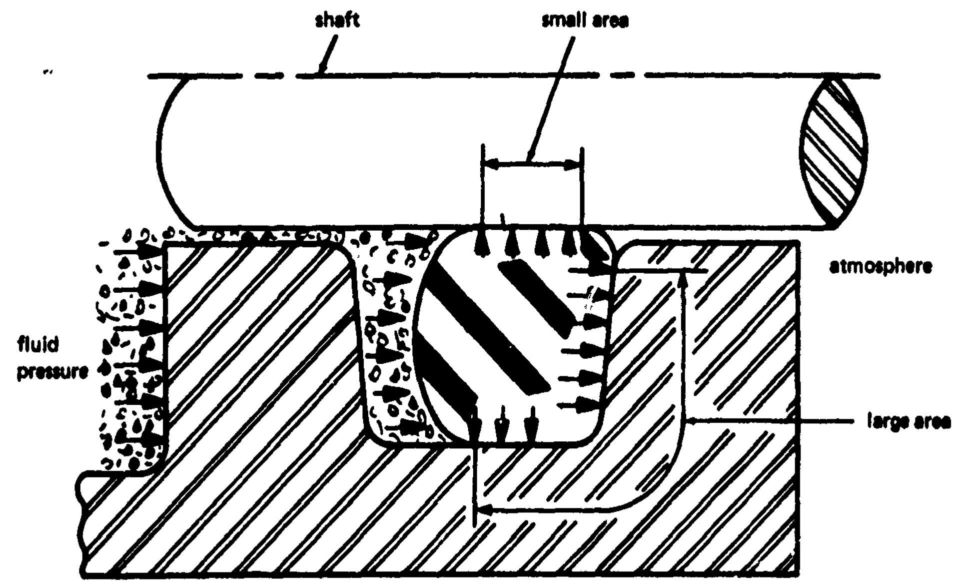


Figure 11. Action of fluid pressure to prevent rolling of O-ring. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

The conditions which cause spiral failure are those which cause some segments of the ring to slide and others to roll simultaneously. Spiral failure occurs when an excessively twisted O-ring is subjected to relatively high pressure. Pressure applied to a twisted portion of a seal magnifies the stress beyond the elastic limit of the rubber. Rapid stress-aging then causes a rupture of the O-ring to start adjacent to the clearance gap. Slight flexing motion or working of the O-ring apparently causes the rupture to penetrate about half-way through the cross section (Ref.10, pp. 6–13). The operational factors which contribute to spiral failure of a seal are speed and length of stroke, pressure differential, temperature of operation, side loads, and contamination or gummy deposits on metal surfaces, whereas the design factors which contribute to spiral failure are lack of lubrication, squeeze, shape of groove, surface finish of gland, type of metal rubbing surface, concentricity of mating metal parts, stretch of O-ring, and improper O-ring installation.

Rubber O-rings are not recommended for applications where the cylinder rod has a stroke of more than 12 in. of unsupported length unless extra precautions are taken. Usually, the longer a stroke of a cylinder rod, the greater the eccentricity, bending, side load, and, in general, the tendency to produce factors which contribute to wear and/or spiral failure. To minimize the consequences of eccentricity, bending, and side loads of a long-stroke cylinder rod, floating glands are most often used. The object of a floating gland is to allow the piston or rod bearing (containing the O-ring groove), to pivot, adjust or float a small amount, offsetting misalignment. The seal housing used in the rotary O-ring problem (Fig. 6) is actually a floating gland or floating seal.

Investigations made by the Parker O-Ring Company have disclosed that spiral failure occurs very often when reciprocating speeds are less than 1 fpm. The apparent reason for failure at slow speeds is that the sliding or running seal friction created is very high and comparable to breakout friction. This high friction tends to excessively twist the O-ring, especially on low or balanced pressure components, and thus spiral failure occurs. O-ring seals are not recommended, therefore, for speeds less than 1 fpm when the differential pressure is less than 400 psi.

The two main design factors that contribute to spiral failure of reciprocating seals are lack of lubrication and groove shape. Lack of lubrication between the O-ring and the sliding rod tends to increase the relative friction and may result in excessive twisting of the O-ring and, eventually, spiral failure. If a V-shaped groove is used, the hydraulic holding force is reduced because the area on the side of the V-groove is less than at the bottom and sides of a square groove, as shown in Fig. 11. Therefore, we can see that when an unlubricated rod or surface is actuated through a seal contained in a groove of reduced area, twisting of the seal and spiral failure has a greater probability of occurring.

GENERAL SEALANTS

The use of general sealants around electrical cables, hoses, tubes, etc. that penetrate walls of submerged structures is limited to applications involving relatively low pressures. It is at shallow depths, then, that general sealants such as epoxies, polyesters, and similar thermosetting resins are used.

General sealants are classified as either rigid or flexible. The hard-setting or rigid sealants are characterized by their inability to flex. They crack if flexed and are often difficult to remove. Common rigid sealants are those based on compounds of epoxies, polyesters, acrylics, polyamides, and polyvinyl-acetates. Table 3 lists the sealant bases for rigid systems and some of their characteristics (Ref. 1, p. 104). It can be seen from the table that the epoxies and polyesters provide a variety of formulation viscosities and, thus, methods and forms of application, such as injection molding, gun extrusion, brushing, spraying, or troweling into cracks and voids.

Nonrigid sealants stay flexible after curing and are all elastomeric based. Their range of flexibility varies considerably, as does hardness, from Shore durometer 10A to 85D. Some

Table 3. Hardening sealants, rigid types. Reproduced from MACHINE DESIGN, June 11, 1964, by permission.

Sealant Base	Sealant Form	—Curing Characteristics—	Curing Time and Temperature Range[1]	Pot Life[2] (hr)	Application Methods[3]	Toxicity During Application	—Remarks—
Epoxy	One-part liquid	Thermosetting resin requires elevated temperature for curing	1-16 hr @ 180-350 F	½-8	G_M, G_P, K	None	Minimum shrinkage, but normally very brittle. Available in thixotropic form.
	Two-part liquid	Cures at room temperature because of addition of curing agent (accelerator), or hardener. No fumes released during epoxy polymerization	1-8 hr @ 75 F some formulations require 4 to 6 days ½-3 hr @ 150 F	½-3	G_M, G_P, K, S	None. Some hardeners are toxic and can cause dermatitis.	Formulations available which are easy to apply because of low viscosity. Available in thixotropic form. Normally very brittle.
	Powder	Thermosetting resin requires elevated temperature cure	24 hr @ 250 F 2 hr @ 320 F 5 min @ 500 F	Unlimited	Powder placed in joint and heat applied. Injection molding principle can also be used	None Fishy odor	Brittle; high durometer. High production application if injection molding techniques used.
Modified Epoxy	Two-part liquid modified with polysulfide polymer	Thermosetting resin cures at room temperature because of addition of curing agent or hardener. No fumes released during cure	½-8 hr @ 75 F	1/6-2	G_M, G_P, K, S	None	
Polyester	Two-part liquid	Thermosetting resin. Catalyst curing agent added at time of use.	½-24 hr @ 75 F. Less time at higher temperature	1/6-4	G_M, G_P, K, B	None. Some hazard in handling catalyst	Low shrinkage and limited flexibility.
	One-part liquid	Thermosetting anaerobic resin. Remains liquid in presence of oxygen. Contact with metal acts as a catalyst to accelerate curing, which must take place in absence of oxygen.	1-24 hr @ 75 F	Unlimited if kept in contact with oxygen	G_M, B, K, squeeze bottle	None	Easy to apply. Used as liquid gaskets and for pipe thread sealing. Can be formulated in various viscosities and shear strengths, to match specific applications.
Acrylic	Made in one part emulsion, one-part solvent relase, and two-part systems		Varies, depending on formulation and curing system	——	G_M, B, S	None	More commonly used as adhesives, but have limited sealing applications
Oleo-Resin	One-part putty or paste	Thermoplastic type material, normally cures by solvent evaporation	14 days or more @ 75 F to reach specific hardness	Very long	G_M, G_P, K	None Oily odor	Easy to extrude. Requires long time to reach measurable physical properties. High degree of shrinkage
Asphalt and Bituminous	One-part putty or paste	Thermoplastic type material, normally cures by solvent release	Several weeks @ 75 F	Very long	G_M, G_P, K	None	Easy to handle and extrude. Available in black only. Can cause staining and tendency to cold flow.
	Two-part flow-type paste	Thermoplastic type with catalyst curing agent	1-8 hr @ 75 F	¼-¾	G_M, G_P, K, B, S	None Strong odor	Same as one-part material.

[1]Ranges given are representative. Formulation and addition of solvents or thinners can alter these values.

[2]Values given are for standard conditions—75F and 50 per cent relative humidity. Times represent limits obtainable by formulation or addition of solvents. Pot life for specific formulations varies with temperature and humidity.

[3]Application methods code: G_M—Manually operated gun for extruding sealant. G_P—Pressure operated gun. B—Brush. K—Spatula, putty knife, trowel, squeegee S—Spray.

of these sealants are true rubbers, and they can be compounded to resist a variety of environmental conditions. These nonrigid sealants are either true elastomers that return to their original shape after being deflected, or they are characterized by taking a slight permanent set after being deformed. Typical elastomeric sealants are listed in Table 4. They are categorized by method of curing. From the remarks in the table it is obvious that flexible sealants would not be adequate underwater, since most are easy to extrude. They can and are used in electrical cables as the insulating layers around and between the conductors.

Most potting compounds are basically sealer formulations and are considered to be sealers if their primary function is to keep moisture out of electrical connectors or electronic assemblies. This is the most common function of sealants in marine applications, that of providing the sealing mechanism for most underwater "through-cables," *i.e.*, underwater electrical connectors and cable assemblies. There are many types of electrical connectors, plugs, and sockets, cable-glands, receptacles, etc., all of which use some type of molding material, such as the common flexible sealants already mentioned, epoxies, polyurethane, polythene, and neoprene, or more exotic materials, such as glass-filled diallyl phthalate.

Various companies make bulkhead connectors that provide completely corrosion-resistant feed-through assemblies for electrical conductors. This means that electrical connection between the exterior and interior of a submerged vessel is conveniently accomplished with high reliability by mass-produced products. O-rings are used to provide the water-tight seal against the mounting bulkhead and the connector body, while the sealing of electrical pins, conductors, and feed-out leads within the connector is obtained by the types of molded sealants already mentioned. Some connectors can withstand pressures of 20,000 psi (for use at maximum ocean depth), while others are depth limited but have other advantages, such as being pressure compensated to enable the connector to be plugged or unplugged underwater.

Although this discussion of general sealants has presented only a few relevant facts, the wide range of applications for general sealants has been implied. The use of general sealants in the underwater environment deserves much more study.

Table 4. Nonrigid sealants, flexible types. Reproduced from MACHINE DESIGN, June 11, 1964, by permission.

Sealant Base	Curing Characteristics	Curing Time and Temperature Range[1]	Pot Life[2]	Application Methods[3]	Toxicity During Application	Remarks
TWO-PART CHEMICAL-REACTION SYSTEMS[4] (Catalyst cured)						
Polysulfide	Thermosetting liquid polymer. Cures at room temperature. Elevated temperature reduces cure time.	16-24 hr @ 75 F	1-6 hr	G_M, G_P, K, B, S	None[5]	Economical; handles easily. Primers sometimes used for substrates in critical applications. Wide range of properties; good flexibility. A versatile sealant.
Polyurethane	Thermosetting elastomer. Cures at room temperature.	16-24 hr @ 75 F 6 hr @ 180 F	3-6 hr	G_M, G_P, K, B, S	None	Economical; somewhat difficult to handle. Has high strength and good abrasion resistance. Normally requires vacuum degassing and primer system.
Silicone	Thermosetting elastomer; room-temperature curing. Should be post cured at high temperature if elevated temperature service is anticipated.	24 hr @ 75 F 6 hr @ 180 F	3-6 hr	G_M, G_P, K, B	None	Excellent for high-temperature service; good thermal barrier. High in cost. Normally requires primer system.
Modified Epoxy	Thermosetting resin; room temperature or heat activated curing.	2 hr @ 75 F	50 min	G_M, G_P, K, B, S	Can affect allergic individuals	Normally modified with plasticizers or other ingredients to impart various degrees of flexibility. Polysulfides and polyamides often used. Economical; handles easily.
Viton and Neoprene	Thermosetting, room temperature curing.	24 hr @ 75 F	3-4 hr	G_M, G_P, K, B, S	None	Viton good for high-temperature service. Costly; difficult to handle; requires priming. Neoprene difficult to handle.
ONE-PART CHEMICAL-REACTION SYSTEMS[6] (Water-vapor catalysts cured)						
Polysulfide	Thermosetting liquid polymer; cures at room temperature. Curing accelerated by heat and/or moisture.	14-21 days @ 75 F	1 yr @ 80 F[7]	G_M, G_P, K	None	Very easy to handle. Tacks out rapidly. Can be applied without primer. Supplied only in thixotropic form.
Polyurethane	Thermosetting elastomer; cures at room temperature. Curing accelerated by heat and/or moisture.	14-21 days @ 75 F	3-9 mo	G_M, G_P, K	None	Excess moisture can cause blowing or sponging. Normally requires primer.
Silicone	Thermosetting elastomer; cures at room temperature. Curing accelerated by heat and/or moisture.	7-14 days @ 75 F	6 mo	G_M, G_P, K	None	Easy to extrude. Can be supplied in transparent color. Normally requires primer.
Acrylic	Thermoplastic type resin that cures at room temperature.	21 + days @ 75 F	3-9 mo	G_M, G_P, K	None Strong odor	Sometimes requires preheating to handle. Low cohesive strength. Generally in the mastic or nonhardening family.
SOLVENT RELEASE SYSTEMS[8]						
Neoprene	Thermoplastic elastomer; sets at room temperature.	7-14 days @ 75 F	6-12 mo[9]	G_M, G_P, K, B, S	None[10]	Fairly easy to apply. Available in colors. Good weathering characteristics; high shrinkage. Poor package stability.
Hypalon	Thermoplastic elastomer; sets at room temperature.	7-14 days @ 75 F	6-12 mo	G_M, G_P, K, B, S	None	Similar to neoprene. Somewhat better weathering resistance.
Butyl	Room curing thermoplastic resin.		6-12 mo	G_M, G_P, K	None	Easy to apply. High degree of tenacity. Relatively costly. Requires long set time to achieve required strength.

[1]Ranges given are representative. Formulation and addition of solvents can alter these figures somewhat.

[2]Values given are for standard conditions—75 F and 50 per cent relative humidity. Times represent limits obtainable by formulation or addition of solvents. Pot life for specific formulations varies with temperature and humidity, especially for water-vapor catalyzed sealants. In general, an increase in room temperature and humidity will shorten pot life, and a temperature-humidity decrease will result in increased pot life.

[3]Application methods code: G_M—Manually operated caulking gun for extruding sealing. G_P—Pressure operated caulking gun for extruding sealant. B—Brush. K—Spatula, putty knife, trowel. S—Spray.

[4]All two part systems require mixing before use. Sealants with relatively long pot lives can be mixed in large batches. Short pot life materials must be mixed in small quantities or dispensed by automatic meter and mixing equipment. Some sealants once mixed, can be held for extended periods by storage at low temperature.

[5]Normally, two-part sealants are nontoxic and do not release fumes during cure. However, some catalysts or curing agents, used to mix with the sealant, can be harmful, especially to allergic persons

[6]One part systems can be applied directly from the can or tube. Moisture content of atmosphere affects cure time to a greater extent than the two part sealants. In general, most one part sealants will set up if relative humidity is at least 20 per cent. Lower relative humidities will extend set-up time and curing time. Cure can be accelerated by adding water to the sealant surface.

[7]Pot life for one part sealants is considered to be the shelf life, (if sealant is kept capped) since no mixing is required for use. Once the sealant has been exposed to water vapor, curing will start. Therefore, it is important to use the material as soon as possible and keep can or tube tightly closed after use to prolong pot or shelf life.

[8]Solvent release systems require long cures, but many formulations will set-up and be tack free in a relatively short time. Thickness of the sealant layer will affect cure time as well as exposure to atmosphere. A tightly confined solvent sealant may take months to cure.

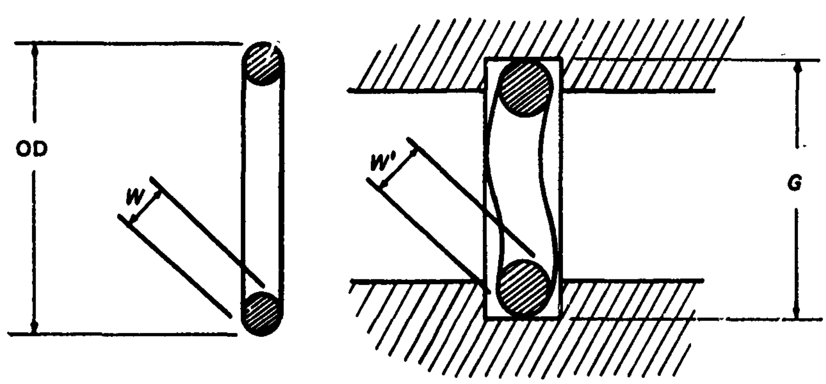
[9]Pot life is considered to be shelf life if sealant is kept tightly closed when not in use.

[10]Fumes are released during cure, but at relatively low rates. Sealant supplier should be consulted for recommendations regarding vapor release of specific formulations and degree of toxicity.

APPENDIX

DETERMINATION OF RELATIONSHIP BETWEEN PERCENT OF DIAMETRAL REDUCTION OF O-RING AND PERCENT INCREASE IN CROSS-SECTIONAL WIDTH

Consider an O-ring being installed into an O-ring groove of smaller diameter



$$\% \text{ of diametral reduction} = \left(\frac{\text{OD} - G}{\text{OD}}\right) \times 100\% = \frac{d(\text{OD})}{\text{OD}} \times 100\%$$

$$\% \text{ increase in cross-sectional width} = \left(\frac{W' - W}{W}\right) \times 100\% = \frac{dW}{W} \times 100\%$$

Relationships:

1. Conservation of mass (if cross section is considered to be always circular):

$$\frac{\pi W^2}{4}(\text{OD} - W)\pi = \frac{\pi (W')^2}{4}(G - W')\pi$$

therefore

$$W^2(\text{OD}) - W^3 = (W')^2 G - (W')^3$$

$$W^2(\text{OD}) = (W')^2 G + W^3 - (W')^3$$

2. Since OD and G differ much more than W and W' and since W^3 is very close to $(W')^3$, we can ignore the last two cubed terms. They are of the order 0.0003, compared to OD and G, which are of an order greater than 0.210. For example, if we inserted typical values:

$$W^2(\mathrm{OD}) = (W')^2G + W^3 - (W')^3$$

$$(0.070)^2(2.372) \approx (0.075)^2(2.196) + (0.070)^3 - (0.075)^3$$

$$0.0049(2.372) \approx 0.0056(2.196) + 0.00034 - 0.00042$$

$$0.0116 \approx 0.0123 - 0.00008$$

The last term of the equation is small compared to the other terms and therefore can be ignored, giving

$$W^2(\mathrm{OD}) = (W')^2G$$

This states the square of O-ring width times the diameter is a constant

$$W^2(D) = K$$

Take the derivative

$$2WdW(D) + W^2d(D) = 0$$

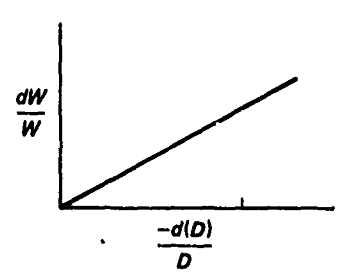
therefore

$$2WdW = -W^2d(D)/D$$

$$2\frac{dW}{W} = \frac{-d(D)}{D}$$

3. Therefore (with cross sections always circular)

$$\frac{\%\text{ increase in } W}{\%\text{ decrease in } D} = \frac{dW/W}{-d(D)/D} = \frac{1}{2}$$



4. Because the O-ring cross section does not remain circular when the O-ring is pressed into the smaller groove, the minimum width W' will be less than that calculated through Step 3. Thus, the actual W' can be estimated to be about 2.5 percent less using a logical analogy from Fig. 12.

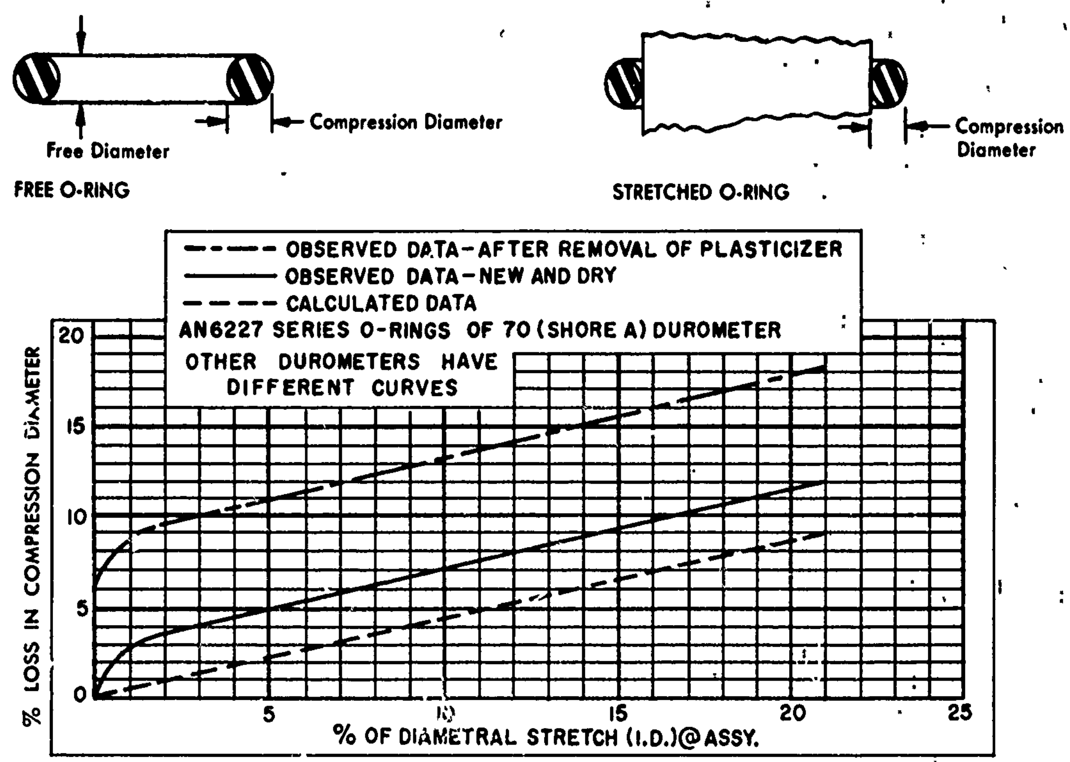
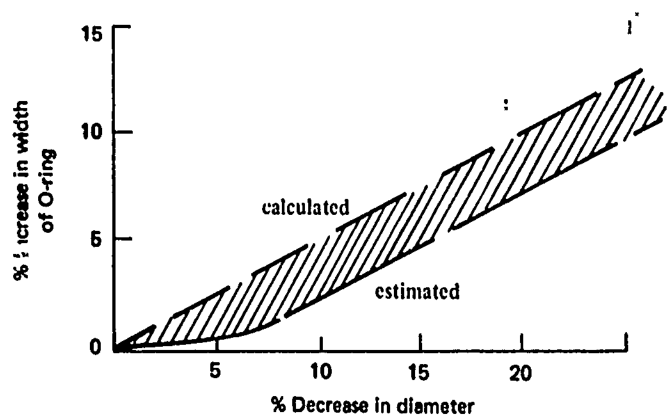


Figure 12. Loss in compression diameter due to stretch. In the stretched condition, an O-ring cross section is no longer circular. It is often necessary to compensate for the loss in squeeze resulting from the reduced "compression diameter." Dimensional changes in the "free diameter" do not affect the seal. Reproduced from Parker Seal Co. *O-Ring Handbook*, October 1967. By permission.

Therefore, the actual data would probably fall between the calculated and estimated curves shown below.



REFERENCES

1. *Machine Design*. "Seals - Reference Issue," A Penton Publication, Cleveland, Ohio, June 1964, p. 2, index.

2. Aeronautical Systems Division. Air Force Systems Command Report 56-272, Part VI, Design Data for O-Rings and Similar Elastic Seals; by George R. Trepus *et al.* Wright Patterson Air Force Base, Ohio, May, 1961.

3. U. S. Naval Civil Engineering Laboratory. Technical Report R-504, Corrosion of Materials in Hydrospace, by Fred M. Reinhart. Port Hueneme, Calif., Dec. 1966.

4. U. S. Naval Civil Engineering Laboratory. Technical Report R-428, Deep Ocean Bio-deterioration of Materials - Part III., by J. S. Muraoka. Port Hueneme, Calif., Feb. 1966.

5. *Ibid.* Part IV, Jan. 1966.

6. Naval Civil Engineering Laboratory. Technical Note N-999, Seal Systems In Hydrospace, Phase I, II, III, by J. J. Jenkins & F. M. Reinhart. Port Hueneme, Calif., Nov. 1968.

7. *Ibid.* Technical Note N-999, Phase I, Integrity of Flange Seal Systems, p. 3.

8. *Ibid.* Technical Note N-1072, Phase III, Effects of Long-Term Hydrospace Exposure on Seal System Integrity. (189 days at 5900 ft), p. 5.

9. Precision Rubber Products Corp., Handbook of O-ring and Dyna-seal Packings, Dayton 7, Ohio, 7th Edition, 1956, p. 11.

10. Parker Seal Company. Parker O-Ring Handbook, Culver City, Calif., 1968, p. 6–20.

11. Parker Seal Company. Seal Compound Manual, Culver City, Calif., 1964, p. 1–2 under "Basic Principle."

12. Richard S. Fein. Boundary Lubrication in Texaco Publication *Lubrication*, Texaco Inc., New York, N. Y., v. 57, no. 1, 1971, p. 12.