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LARGE DIAMETER METAL SEALS

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Nuclear Rocket Applications

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LARGE DIAMETER METAL SEALS FOR NUCLEAR ROCKET APPLICATIONS

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INTRODUCTION

Large diameter metal seals are used in the reactor pressure vessel of nuclear rocket engines. Seals having nominal diameters over 40 inches are located at the main closure of the pressure vessel and at the pressure vessel-nozzle interface. Their environment includes cryogenic conditions imposed by the liquid hydrogen propellant, and high intensity nuclear radiation from the reactor. This paper describes development of several types of metal seals with functional characteristics that are successfully meeting the conditions introduced by the "large diameter problem."

THE LARGE DIAMETER PROBLEM

For purposes of this paper any diameter exceeding 24" is considered to be large. Environmental effects can produce large diameter problems in smaller joints. As diameters increase to several feet, a number of conditions arise which are difficult to control, complicating the design of a suitable joint and seal. These conditions include the following:

1. Dimensional tolerances must be increased. Temperature control during machining is mandatory. Eccentricity and out-of-round problems are magnified.
2. Surface roughness and waviness are more difficult to control.
3. Assembly clearance must be increased. "Feel" is lost when assembling bulky and heavy components.
4. Thermal effects are more pronounced.
5. Flange loading per unit circumference increases directly with diameter, for a given pressure, resulting in greater separation for a feasible cross sectional area. At 1000 psi, this loading for a 50 inch diameter becomes 12,500 lb/in.
6. Economic considerations prohibit scrapping a large expensive part because of minor imperfections on a sealing surface.

NUCLEAR ROCKET REQUIREMENTS

Nuclear rocket applications introduce additional conditions of cryogenic temperatures associated with liquid hydrogen and the effects of intense nuclear radiation flux. Cooling requirements due to heating from gamma rays place limits upon material thicknesses, bolt location, and other design factors. Relative flange rotation, relative radial deflection, and bolt elongation or clamp deflections cause a gap and radial shear at the seal when the system is pressurized. While improvements are being made in the structural matching of mating components to minimize this flange separation and shear, gaps of .020 inch to .085 inch are encountered. In small diameter joints, it is possible to accommodate the intended seal by providing flange rigidity, dimensional fits, and sealing surface conditions (1). In large diameter joints, the structural design is perfected as much as possible, but the seal must accommodate adverse conditions which inevitably remain. Merely scaling up designs used for small diameters is not satisfactory.

Development of metallic seals presented in this paper is directed toward achieving the following functional characteristics:

1. Provide a generous clearance for initial seal placement.
2. Yield the seal during assembly, following elastic deflection, adjusting its dimensions to those of the groove, thereby relieving the tolerances that would otherwise be required for seal and mating components.
3. Provide sufficient compressive rigidity in the seal to produce adequate flow of material at the sealing surfaces to accommodate surface conditions and affect an initial seal. Surface conditions encountered include roughness between 32 and 63 rms, waviness of .003 inch, scratches and slight dents.
4. Respond to movements between flanges to maintain sealing when system is pressurized.

The seals described in following paragraphs are machined from 6061-T4 aluminum cylinders and lead plated (thicknesses from .002 to .005 inch). The material, selected for thermal compatability with aluminum pressure vessels, is rolled into a cylinder from plate, welded, and heat treated. Certain dimensions of seals and flange grooves tested are included to illustrate design concepts. Specific applications require design variations.

WAVE RING SEAL

A wave ring gasket described by Freeman (2) and others, is a type of "lens ring gasket" in a pressure energized class having a radial interference fit at assembly of one part in 2,000 to 10,000. The conventional wave ring gasket was developed for small rigid high pressure vessels. The Wave Ring Seal (Fig. 1) developed during this work adapts the "lens ring" principle to large diameters. Rounded lobes provided, fit into a cylindrical seal groove in each flange member. Radial compression of the seal during assembly applies radial loading along these lobes forming initial seal lands. When pressure is applied, this radial loading and corresponding sealing stress increases due to pressure energizing, maintaining the seal. The seal lands ride the cylindrical grooves during separations of up to .085 inch.

A 50 inch nominal diameter seal, initially placed in a locating bore, is radially compressed .043 inch (Fig. 1b) by a guide-in chamfer as the seal is pressed into place. This represents an interference fit of one part in 600. A portion of this compression occurs as yielding, the balance remaining as elastic deflection. The lead coating yields and forms an initial seal. In practice the seal is pressed into one flange member first. The seal cross section rotates, the diameter at the exposed lobe remaining near its original value. This demonstrates the capability of the seal to adapt to differences in mating groove diameters.

X-SEAL

The X-Seal was developed for use where seal grooves may be provided in both mating interface members (Fig. 2). A tongue and groove configuration (Fig. 2d) is also used, with vent holes providing pressure energizing. The initial placement of the seal (Fig. 2a) produces a gap of .054 inch. As the flanges are drawn together, the seal legs first deflect elastically, then yield in bending and move into the corners. At this stage, the gap is approximately .022 inch. As the flanges are drawn into contact, compressive yield and buckling occur in each seal leg. This crush assembly sequence first causes the ring to become sized to an intimate fit, adapting to variations in groove diameters and depth. Secondly, the relatively high compressive loading mates narrow lands on the seal legs to surface waviness, while the lead plating flows into surface asperities, providing a positive initial seal without disturbing the harder flange material. When system pressure is imposed, the flange surfaces remain in contact to some threshold pressure. At higher pressures, a gap opens between the flange surfaces (Fig. 2c). This gap is approximately proportional to pressure increase relative to the threshold pressure. As this gap occurs, the X-Seal legs on the low pressure side of the groove deflect as required and remain forced against the groove corners by distributed pressure loading. This pressure load is borne by the narrow seal lands, thereby maintaining an adequate sealing stress through the action of pressure energizing. The soft lead plating serves as lubricant and sealant.

Seal groove runout and concentricity tolerance between mating flanges in "blind" assemblies may prevent the X-Seal from entering both grooves properly. In some cases, this may be corrected by an entrance guide chamfer on one of the grooves. The Extended X-Seal (Fig. 2b) provides a more positive solution. The groove in the upper flange member is widened by .060 inch and the height of the seal is increased by .030 inch. The upper pair of legs, being extended in length, reach into the corners of the wide groove when assembled.

Loading and unloading deflection characteristics were investigated for the X-Seal and Extended X-Seal using an Instron Test Machine. Five inch long segments were compressed between plates containing corresponding segments of seal grooves. Results for typical samples tested are shown in Fig. 3. It may be observed that elastic bending occurs until point "A" is reached, at a load of 400 to 600 lb/in. Bending yield then progresses through all legs until point "B" is reached at a load of about 700 lb/in. At this stage, all legs have reached the corners of the groove, and upon further closing of the plates, compressive yield progresses through the four legs until the seal plates close to zero gap (point "C") at a load of about 1800 lb/in.

OMEGA SEAL

A typical pressure-energized metal seal has two cantilever legs which deflect within their elastic range during assembly. The resulting load applied at the seal lands is depended upon to provide an initial seal. A soft metal or plastic coating is usually employed. With sufficiently narrow seal lands and providing the flange separation is within the elastic deflection range of the seal, the initial seal is maintained when pressure is imposed. The thickness of the legs must be a compromise between a large value for stiffness to provide load sufficient to form the initial seal and a small value for maximum elastic deflection. Proper functioning requires finishes generally better than 32 rms, close control of flatness, and flange separations held to less than a few mils.

The Omega Seal (Fig. 4) is a pressure-energized type with features adapting it to large diameter use. This seal is being developed for use in joints where a seal groove may be provided in only one flange. Upon assembly the free ends of the seal legs close together forming the characteristic omega shape permitting direct compressive loading through the seal lobes for the purpose of forming the initial seal. At the same time, the cross section is of sufficiently low rigidity to yield and bend upon at low pressures, permitting it to follow flange separations. This opening of the seal may be limited to elastic deflection in applications where the possibility of fatigue exists.

The initial placement of the seal illustrated produces a gap of .092 inch (Fig. 4b). Elastic deflection during assembly compression is approximately .006 inch. The "backbone" then yields, permitting the legs to close into contact at their free ends with a flange separation of .023 inch (Fig. 4c). The outer seal diameter increases during this phase, moving into contact with the outside diameter of the groove. The complete closing of the flange (Fig. 4d) then forms contact lands approximately .030 inch wide. As these lands are formed, the base material yields to conform to dimensional variations and surface waviness while the lead coating flows into surface asperities.

When pressurizing, several vent notches provided permit gas to flow into and pressurize the internal void of the seal when the legs are in the "closed" configuration (Fig. 4e). Tests indicate that the seal "backbone" is held in place against the outside of the groove by friction during pressurization and that flange separation is followed by the free leg which contacts the flat flange member (Fig. 4f). Fig. 5 is an idealized diagram showing the total sealing stress versus pressure and the corresponding sealing stress components due to pressure and deflection loads. For illustration purposes, the seal is assumed to have an elastic deflection for the active leg of .003 inch at an elastic limit load of 15 lb/in, an effective leg length of .180 inch, and a seal land width of .030 inch. The flange is assumed to begin separating at a pressure of 200 psig with a .003 inch per 100 psi rate (.024 inch at 1000 psi). A four stage sequence occurs during flange separation. The pressure component of loading increases with pressure throughout this sequence:

1st Stage: Zero gap at seal. The elastic component of loading at the seal land is constant.

2nd Stage: Flange separation begins at 200 psig. As the gap opens the elastic component decays to zero.

3rd Stage: The elastic component goes negative and increases in this direction until the seal reaches its yield point.

4th Stage: The plastic component as the seal yields in bending remains at a constant negative value.

The total sealing stress at any time is the algebraic sum of its components. It is important that it exceed pressure by a suitable margin at all times. Results indicate that the sealing stress must be maintained at a minimum of twice the pressure to prevent excessive leakage and must greatly exceed the yield stress of the seal coating to maintain a positive seal. It should be noted that the initial seating load for the seal decays with the slightest separation of the flanges and is therefore not included as a component in the above discussion.

The allowable flange separation for a given size of seal cross section depends upon leakage limits and the number of operating cycles expected. For relatively large gaps, degradation of the soft sealant coating with repeated cycling and possible fatigue of the base material must be considered.

Loading and unloading deflection characteristics obtained for five-inch long Omega Seal segments using an Instron Test Machine are shown in Fig. 6. Elastic bending occurs until point "A" is reached, at a load of approximately 20 lb/in. Bending yield then progresses until the free ends of the legs come in contact at point "B" with a load of 28 lb/in. Compressive yielding then progresses until the seal plates close to zero gap (point "C") at a load of about 750 lb/in. The unloading curve indicates an elastic spring back from point "D" of about .008 inch, slightly greater than the .006 inch recovery noted for tests with complete seals.

TESTING

Scale Model Vessel. The Scale Model Vessel is basically a two-thirds scale model of a typical KIWI-B pressure vessel, fabricated for use in structural and seal testing at room and cryogenic temperatures. Seal grooves for the Wave Ring Seal and the X-Seal are provided at the closure joint between the two halves of the vessel. Leakage collected in a manifold groove is measured by a metering unit consisting of a low level pressure switch and solenoid dump valve. A recording of pressure in the metering unit provides continuous indication of instantaneous leakage rate.

Wave Ring Seals and X-Seals of 36 inch nominal diameter were tested alternately with gaseous and liquid nitrogen to pressures of 250 psig (flange separation — .004 inch). In addition, X-Seals were tested hydrostatically to 600 psig (flange separation — .021 inch). No leakage was encountered with the X-Seals. Wave Ring leakage was less than 10 cc/sec, improving with repeated cycles. Results at cryogenic temperatures were not significantly different than corresponding room temperature results.

Full Scale Pressure Vessel. Structural and proof tests, conducted during development of pressure vessels, provide a limited method of testing metallic seals. A Wave Ring Seal of 50 inch nominal diameter, and in one instance an X-Seal, is used at the vessel closure. An X-Seal, Extended X-Seal, or Omega Seal of 40 inch nominal diameter is used at the pressure vessel-nozzle interface.

Testing at pressures to 1000 psig was hydrostatic. Satisfactory seal performance for each type of seal was noted by visual observation during tests conducted primarily for structural data. One series of tests included gaseous nitrogen leakage checks at pressures up to 250 psig, limited by safety considerations. The leakage rate for one Wave Ring Seal was less than 2 cc/sec and for a second seal was up to 30 cc/sec. No measurable X-Seal leakage occurred. Flange separations up to .085 inch were measured at the Wave Ring Seal location and up to .042 inch at the X-Seal location. No Omega Seal leakage occurred with flange separation up to .010 inch.

Metal Seal Test Assembly. The Metal Seal Test Assembly (Fig. 7) is a large diameter, low volume fixture designed to provide for testing flexibility and saftey during gas tests to pressures over 1000 psig. Details of the construction are shown in the sectional view of Fig. 8. Major parts are four annular rings. The two outer rings (seal plates) hold the seal being tested, while the inner rings support the seal plates and seal them at their inner diameters. The lower seal plate is captive, while the upper plate floats. Thirty-two tie rod assemblies run vertically through the center of the pressurizing annulus (inboard of the test seal). These tie rods are in effect stiff springs. By interchanging tie rod and spacer materials (aluminum and steel) and varying the initial tightening loads, the seal plates may be controlled to separate at various pressures and at various rates with respect to pressure. Flange separations from zero to greater than .040 inch are attainable. A leakage collection groove is located outboard of the test seal, with leakage confined by an o-ring (confines leakage for plate separation up to approximately .050 inch). Leakage is measured using the metering unit described previously, or by water displacement.

A series of Omega Seals, of a 40 inch nominal diameter, were gas tested at pressures to 1200 psig. One portion of this testing was conducted for comparative purposes at zero seal plate separation with leakage collected over water. When testing with nitrogen gas, the seal was bubble tight until 800 psig at which point a slight disturbance of the initial seal occurred. Leakage increased to 2 cc/sec at a maximum pressure of 1000 psig. After two additional pressure cycles, leakage decreased to 0.6 cc/sec. On testing with a seal plate separation of .020 inch, maximum leakage noted (at intermediate pressures) was 7.5 cc/sec during early pressure cycles. This leakage decreased to 2 cc/sec at 1000 psig. After 40 cycles, maximum leakage reached 35 cc/sec, decreasing with pressure increase to 5 cc/sec at 1200 psig. At a seal plate separation of .033 inch, rapid wear at the sealing surfaces resulted in leakage ranging to 300 cc/sec, decreasing with pressure increase to 15 to 25 cc/sec at 1000 psig.

The general behavior of the Omega Seals tested may be described as follows:

1. No leakage occurs at pressures up to that at which a gap begins to open, disturbing the initial seal.

2. As a gap begins to open, leakage commences and at first increases with pressure. The seal interface is progressively disturbed.

3. The leakage levels off at about 600 psig at which point the pressure energizing effect produces yield stress in the lead coating on the seal lobes. The re-establishment of sealing results in decreased leakage with further pressure increase.

Field Tests at NRDS (Nuclear Rocket Development Station). Field tests are conducted at NRDS located near Mercury, Nevada. Cold flow and powered reactor experiments conducted to date using liquid hydrogen have subjected metallic seals to temperatures from cryogenic to above ambient, and a high nuclear radiation flux. Prior to each test run, gas leakage checks are conducted with helium at pressures to 300 psig. Seals of each type described have been used with no indication of leakage during reactor runs.

DISCUSSION

The seals tested provide a generous clearance for placement, initially deflect elastically at assembly, and then yield to adjust dimensions to those of the groove. The X-Seal and the Omega Seal provide compressive rigidity to produce an adequate flow of material to affect an initial seal. All of the seals respond to pressure loading and follow flange separations. Thus, all of the desired functional characteristics are met with the exception that the Wave Ring Seal in some cases may not provide compressive rigidity sufficient to produce a positive initial seal.

A Wave Ring Seal, sliding in a cylindrical groove is capable of functioning over a relatively large range of flange separations. Waviness or "low spots" in the flange groove can lead to inadequate local contact at pressures below those where pressure energizing is effective. Pressure cycling generally results in effective sealing.

The X-Seal, while requiring appropriately matched grooves or tongue and groove in mating flange members, is capable of following flange separation, rotation, and shear movements when pressurized, with a minimum of disturbance of the sealing contact surfaces. When pressure is relieved, the flanges recover and reform the seal to the initial assembled shape ready for the next pressure cycle.

The Omega Seal greatly simplifies mating problems between flange members, as it requires a groove in only one member of a joint. Some leakage is encountered with present designs at intermediate pressures due to sliding movements at the sealing land. This effect may be reduced by increasing the length of the seal legs.

Surface finishes generally ranged from 32 to 63 rms, with local roughness in some zones up to 80 rms.

CONCLUSIONS

The seals described in this paper successfully meet severe operating conditions associated with large diameter joints in nuclear rocket engines. Dimensions are used to the extent felt necessary to demonstrate seal concepts. These concepts may be applied without fundamental change to virtually unlimited diameters. For a seal groove width of only .252 inch, the X-Seal has demonstrated effective sealing for flange separations up to .040 inch. For the same groove width, the Omega Seal is effective for separation up to .020 inch and is usable with reduced performance up to .030 inch. The Wave Ring Seal has demonstrated effectiveness for flange separations up to .085 inch, although .060 is a preferred limit for a seal of the design tested. Seal design for specific applications must be preceded by a determination of the effective flange separation pressure, maximum separation, flange rotation, and shear movement.

ACKNOWLEDGEMENTS

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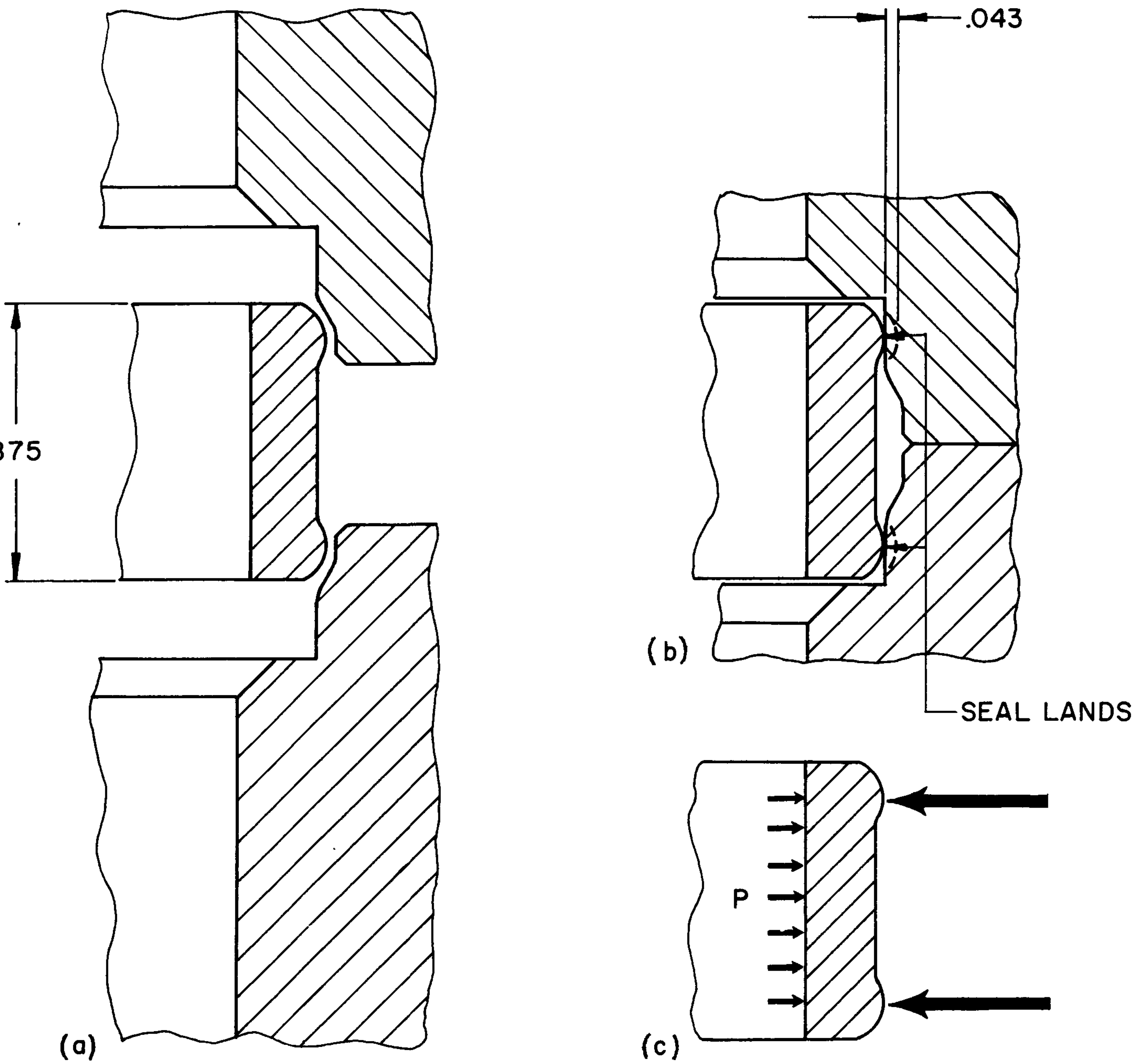

FIG. 1. WAVE RING SEAL : (a) Positioned; (b) Assembled; (c) Pressurized

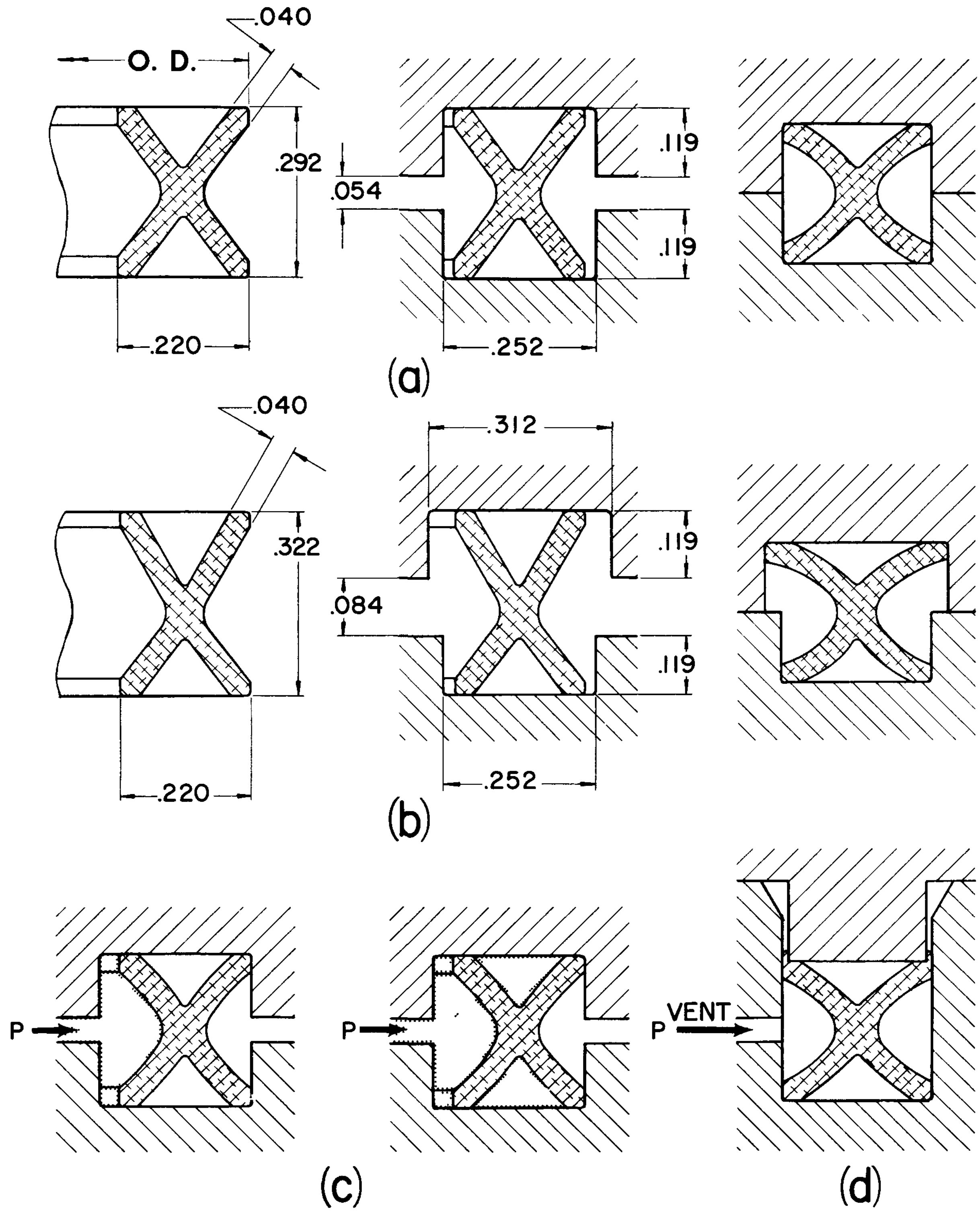

FIG. 2. X-SEAL

(a) Positioned and assembled
(b) Extended X-Seal, positioned and assembled
(c) Pressurized, with flange separation
(d) Tongue and groove configuration

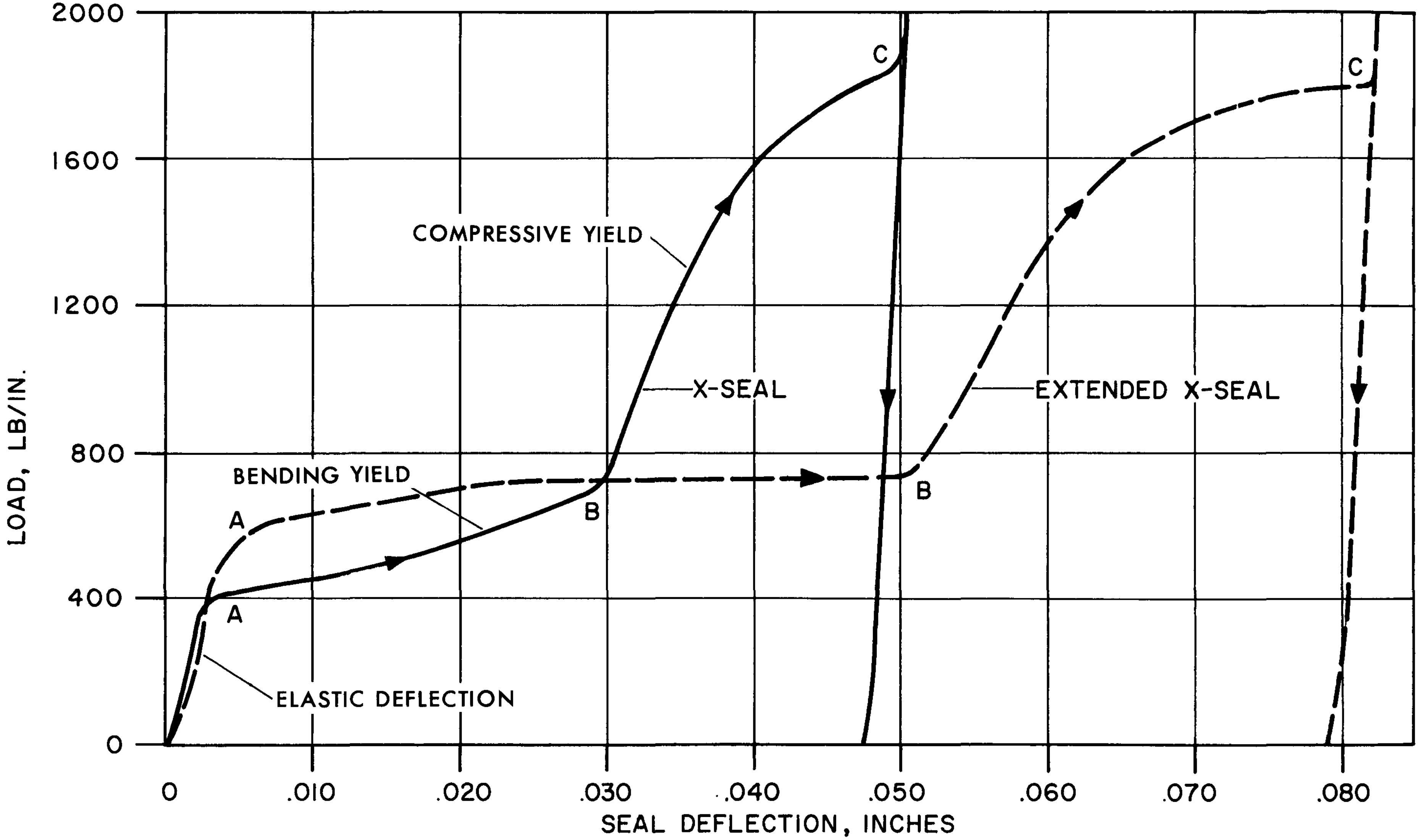

FIG. 3. X-SEAL ASSEMBLY LOAD AND UNLOAD CYCLE

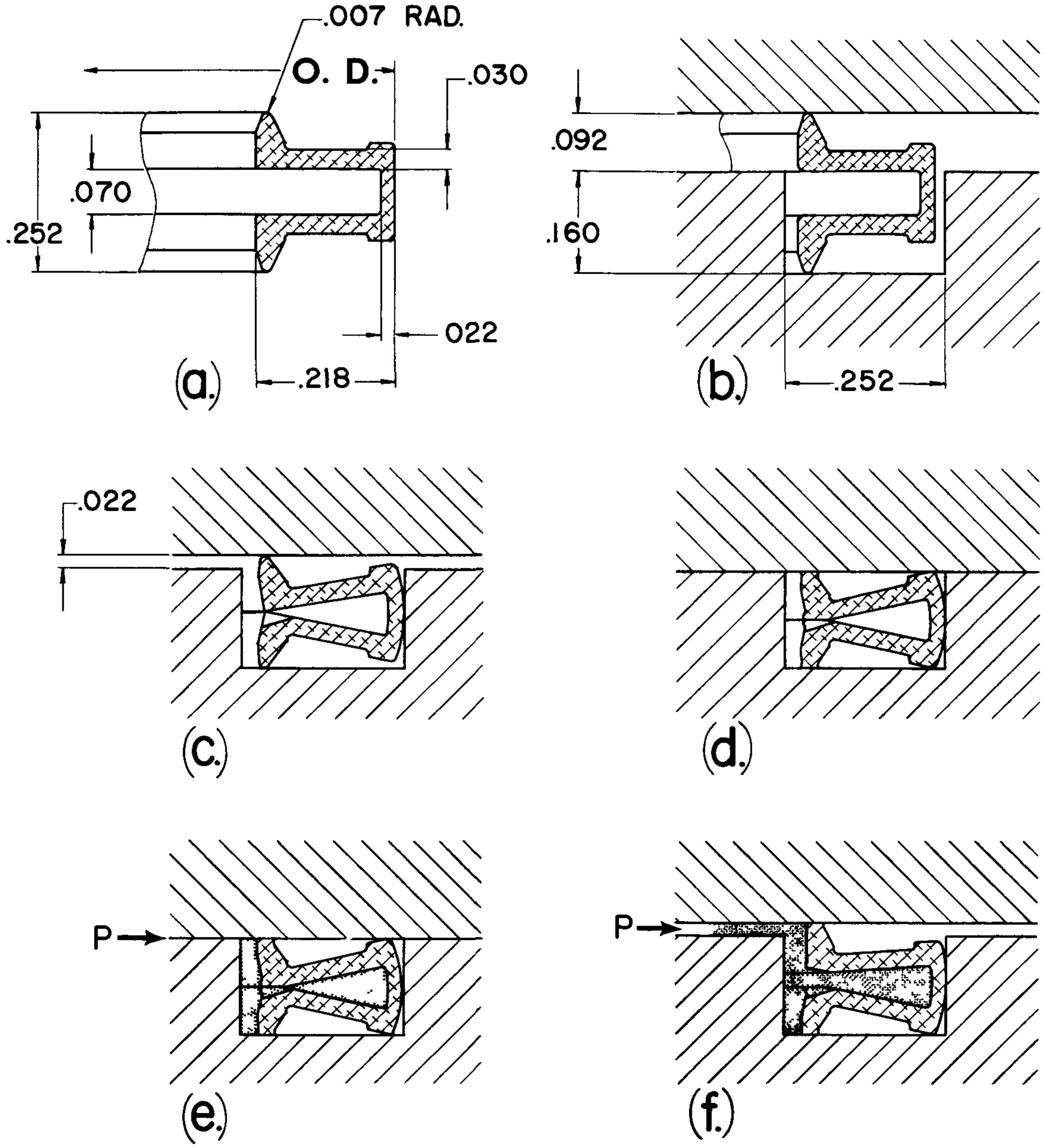

FIG. 4. OMEGA SEAL

(a.) Seal detail
(b.) Positioned
(c.) Seal legs in contact during assembly
(d.) Assembled
(e.) Low pressure
(f.) High pressure with flange separation

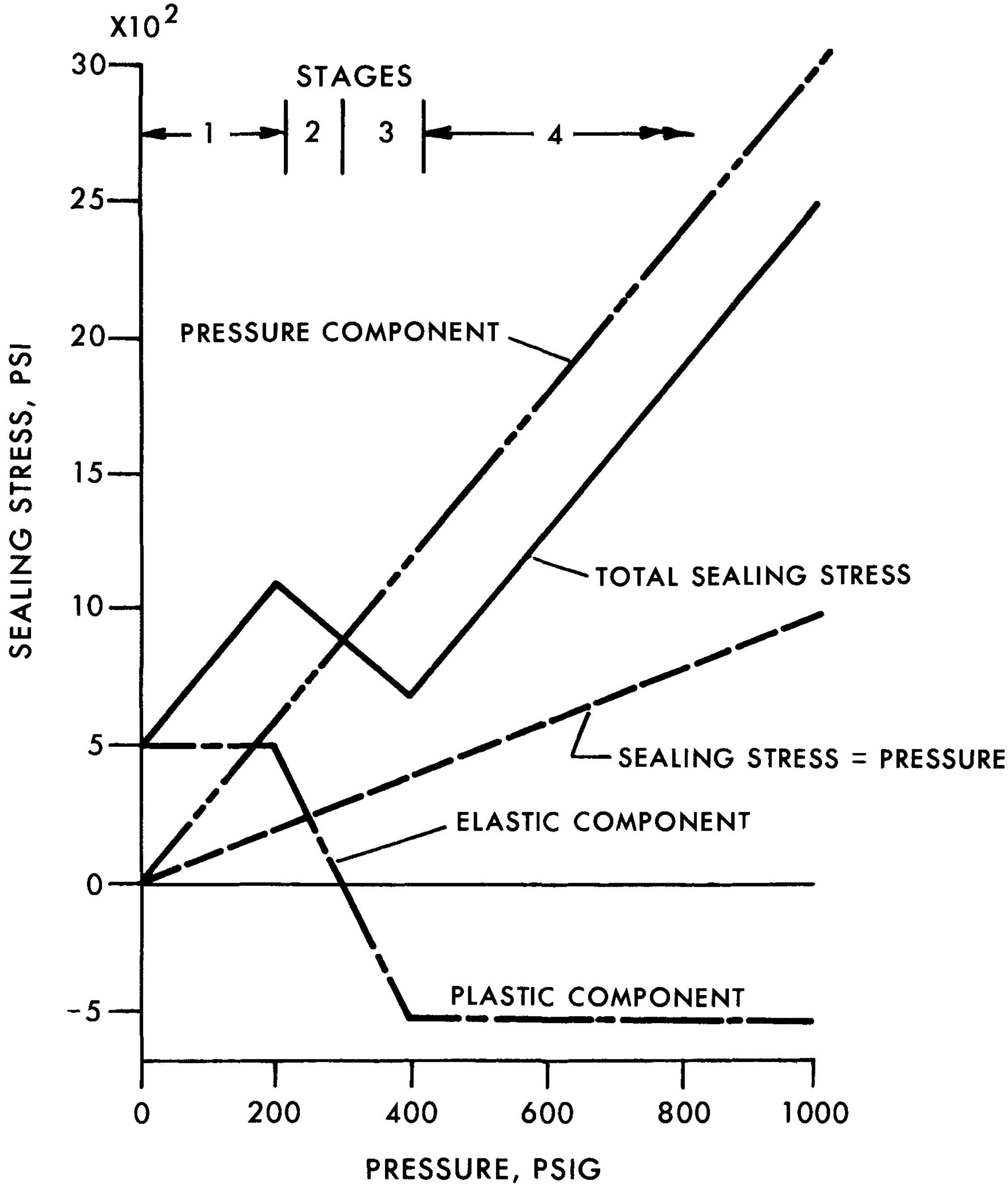

FIG. 5. IDEALIZED OMEGA SEAL OPERATION

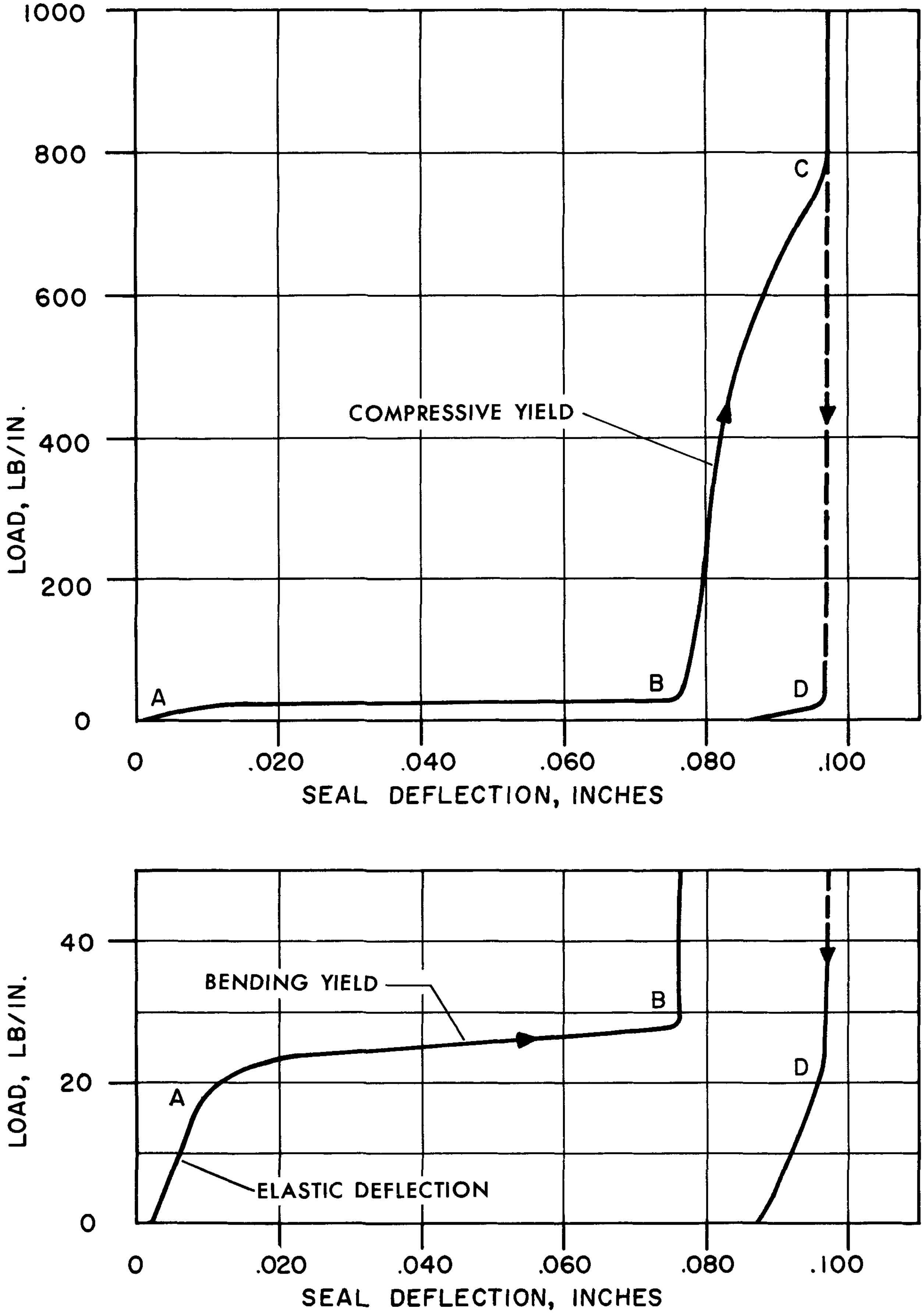

FIG. 6. OMEGA SEAL ASSEMBLY LOAD AND UNLOAD CYCLE

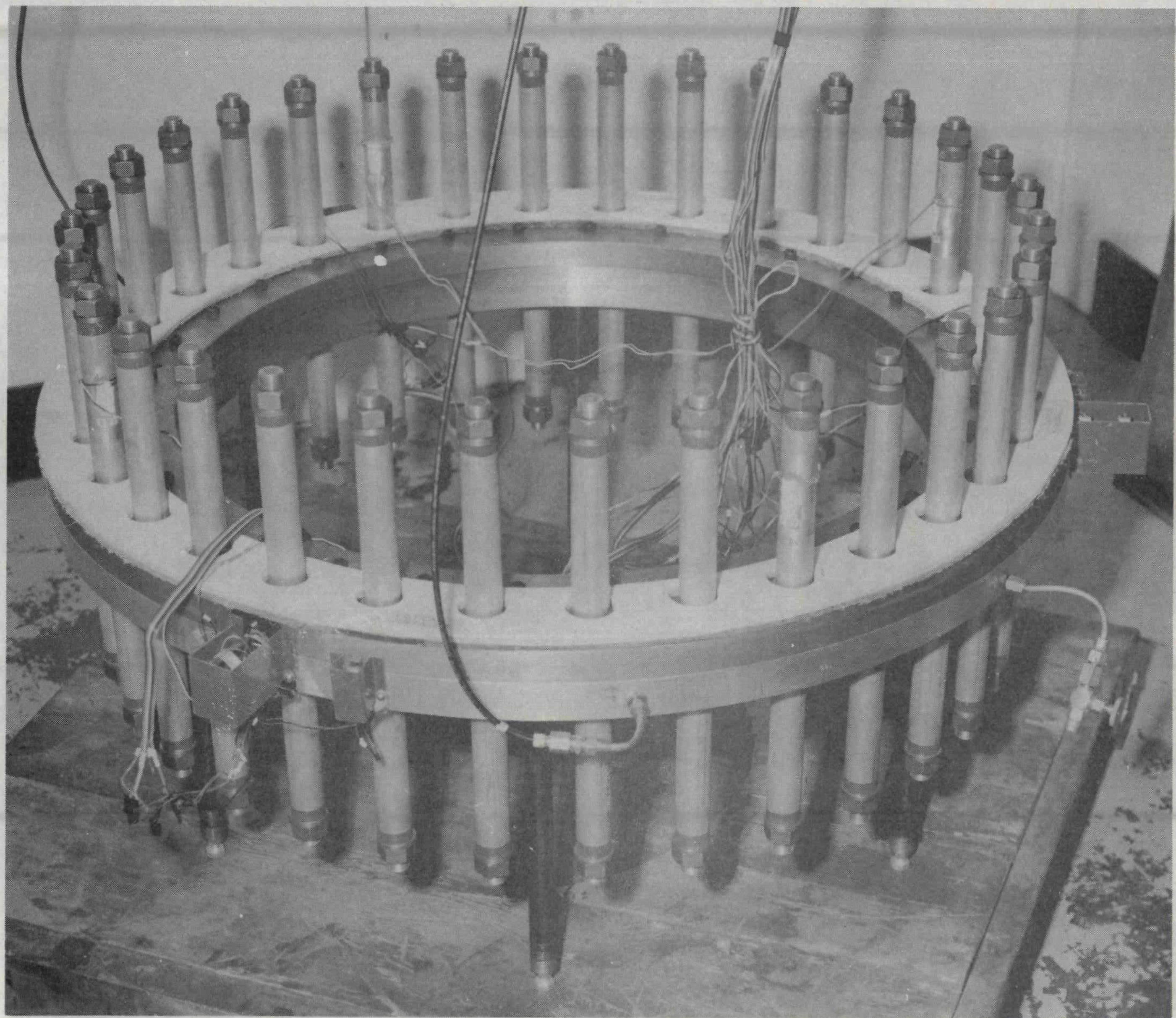

FIG. 7. METAL SEAL TEST ASSEMBLY

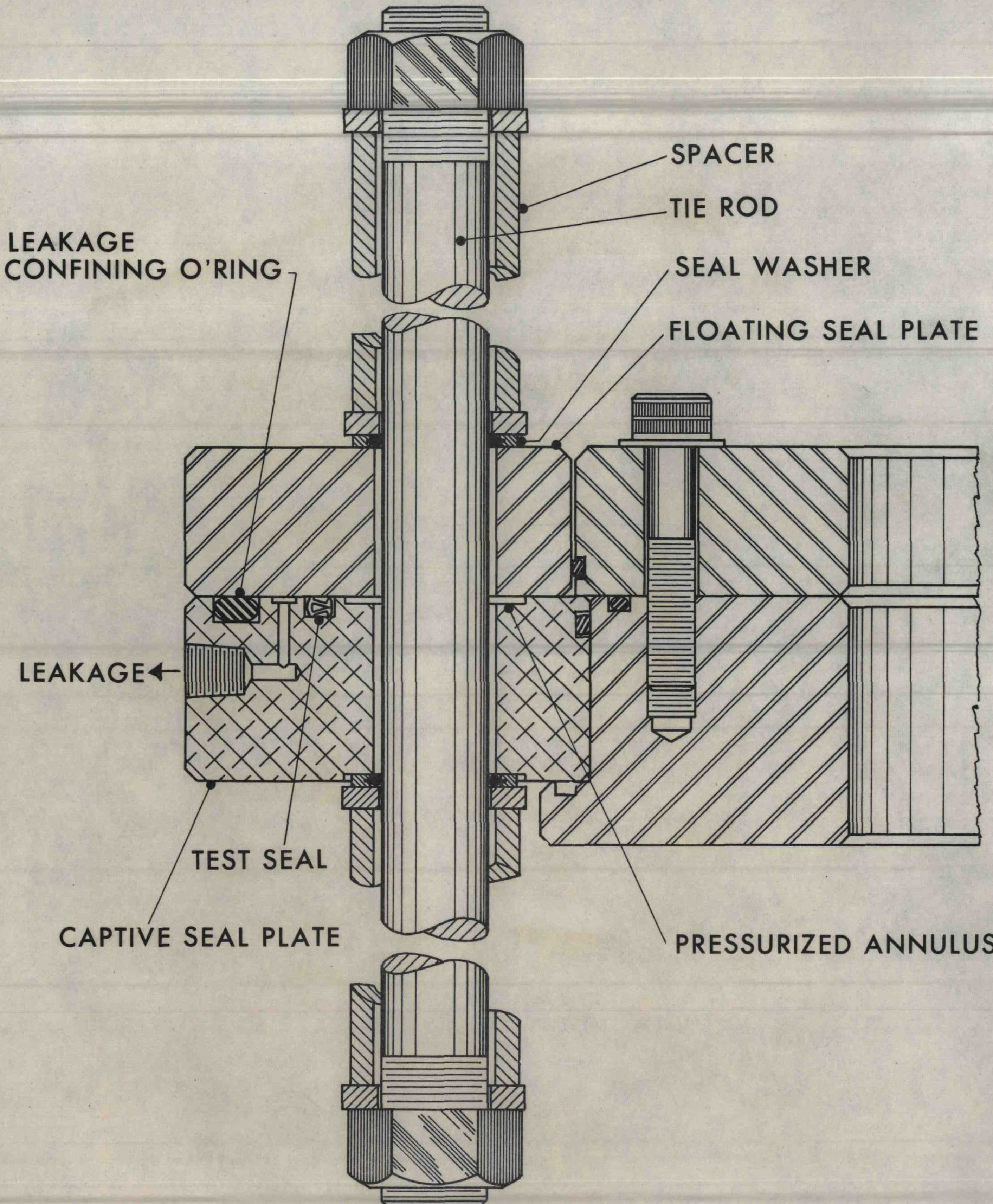

FIG. 8. METAL SEAL TEST ASSEMBLY