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Metallic Seal Rings for High Temperature Reciprocating Hydraulic Service

RATIONALE

This document has been determined to contain basic and stable technology which is not dynamic in nature.

STABILIZED NOTICE

This document has been declared "Stabilized" by the A-6 Technical Committee and will no longer be subjected to periodic reviews for currency. Users are responsible for verifying references and continued suitability of technical requirements. Newer technology may exist.

1. PURPOSE

This report is intended for use by those people involved in the design and application of high temperature hydraulic systems and components. It is, in essence, a first attempt at the standardization of seal ring, groove, and gland design for hydraulic piston heads. Specifically, the report covers some design, application, and test aspects of metallic seal rings for reciprocating hydraulic service at -65 F to +630 F fluid or ambient temperature and up to 4000 psi for cylinder diameters ranging from 1/2 inch to 6 inches.

2. SCOPE

An attempt has been made to consider all features of seal ring design including configuration, materials, hardness, dimensions, surface finishes, surface treatment, leak testing, and general quality. In addition to this, allowable cylinder breathing and general quality requirements of mating hardware are discussed. Also, at the end of this report, there is a brief paragraph on other types of seal rings.

3. GENERAL

The information presented herein is based upon current, most common usage factors and represents what is considered to be sound and proven engineering practice.

4. SEAL RING CONFIGURATION

Survey results indicate that the most commonly used configuration is a two piece bi-directional seal ring assembly, consisting of a step joint outer ring and a straight cut inner ring. In operation, the step joint outer ring blocks the axial leakage path, while the straight cut inner ring seals the radial leakage path and acts as an expander at zero or very low pressure differentials. (See Figure I.)

4.1 Seal Ring Materials:

4.1.1 Step Joint Outer Ring: High Temperature Alloyed Cast Iron with the following mechanical and physical properties:

4.1.1.1 Minimum Tensile Strength = 35,000 psi.

4.1.1.2 Modulus of Elasticity = 11×10^6 psi to 15×10^6 psi.

4.1.1.3 Minimum Izod Impact Strength (.140 x .260 unnotched bar) = 3.5 in. lbs.

4.1.1.4 Hardness (3000 Kg. 10 mm Ball) = 220 - 297 BHN, or Rockwell "G" 77 - 94.

4.1.1.5 Density = .26 lb/in^3 approximately.

4.1.1.6 Coefficient of Expansion = 7.5×10^{-6} in/in/deg F.

4.1.1.7 Maximum allowable operating temperature (of alloyed cast iron) = 850 F.

4.1.2 Straight Cut Inner Ring: 17 - 4 PH Stainless Steel (per AMS 5643 or AMS 5398) with a hardness of Rockwell "C" 30 - 40. AMS 5613 with the same hardness is also frequently used. These materials have a maximum allowable operating temperature of 900 F and exhibit excellent spring properties and corrosion resistance.

4.2 Recommended Seal Ring Dimensions: (See Table A.)

4.3 Surface Treatment:

4.3.1 Outer Ring: Parco Lubrite #2; this acts as a rust inhibitor and assures rapid seating of the cast iron ring.

4.3.2 Inner Ring: No surface treatment required.

4.4 Nomenclature: Standard SAE piston ring terminology is applicable to the various surfaces of the two piece seal ring assembly. (See Figure II.)

4.5 Sealing Principle: Figure III shows a cross-sectional view of a two piece seal ring assembly in its sealing position with primary sealing contact between the face of the outer ring and the cylinder bore, and with secondary sealing contact between the downstream side of the ring and the downstream side of the piston groove.

4.5.1 Primary Sealing: Initial sealing at the primary contact surface is achieved through: 1) the inherent tension manufactured into the outer ring, and 2) by the calculated spring tension manufactured into the inner ring.

4.5.2 Secondary Sealing: When system pressure is applied, primary sealing prevents leakage past the face of the outer ring. The fluid, then, is forced to follow a tortuous path down between the upstream sides of the ring and groove, past the back of the inner ring and up between the downstream sides of the ring and groove, in order to leak past the seal ring assembly. (See Figure IV.) Fluid flow through these very small side and back clearances creates a pressure drop across the seal ring assembly and, as a result, the axial hydraulic force is greater on the upstream side than on the downstream side of the seal assembly. The resultant unbalanced force seats the seal ring assembly against the downstream groove side, thus establishing secondary sealing contact. Once established, primary and secondary sealing are sustained by the steady unbalance of radial and axial hydraulic forces caused by continuous application of hydraulic system pressure. (See Figure V.)

4.6 General Quality Requirements:

4.6.1 Light Tightness: The seal ring assembly should be 100% light tight, when mounted in a ring gage of the specified gage diameter, and 100% light tight between inner and outer rings for a distance extending at least 20 deg either side of the outer ring step joint. Light which can be pressed out upon application of a radial load (not exceeding 5 lb. per in. of diameter) is acceptable.

4.6.2 Surface Finishes:

4.6.2.1 The face and back of the outer ring should be finished 20 RMS and the sides 16 RMS, or better, before surface treatment. Mating surfaces of the outer ring step joint should be finished 90 RMS, or better, prior to surface treatment.

4.6.2.2 The face of the inner ring should be finished 20 RMS and the sides 16 RMS, or better.

4.6.3 The O.D. edges of the outer ring may have .003 max. radius for gage diameters up to 4 inches and .005 max. radius for gage diameters above 4 inches. The I.D. edges of the inner ring may have a .015 max radius. All edges should be free of burrs.

4.7 Leakage Control: In addition to the seal ring assembly characteristics which are controlled by the seal ring manufacturer, there are two other major factors which affect leakakge; these are the system operating conditions defining the sealing requirement, and the quality of all mating parts, which is controlled by the actuator manufacturer.

4.7.1 Effects of System Operating Conditions:

4.7.1.1 Pressure: A properly designed and manufactured seal ring assembly exhibits decreasing leakage rates with increasing pressure, from 50 psi to 4500 psi. Increasing pressures tend to improve contact between the O. D. of the seal ring assembly and the actuator bore, the inner ring O. D. and the outer ring I. D., the downstream seal ring and groove sides, and the mating surfaces of the outer ring step joint. Poorly designed and manufactured seal ring assemblies (that is, rings having poor shape, poor flatness, or excessive step joint clearance or interference) have definite leakage orifices and exhibit increasing leakage rates with increasing pressures.

4.7.1.2 Temperature: Leakage increases at higher temperatures because of the associated decrease in hydraulic fluid viscosity. The viscosity of some hydraulic fluids decreases by a factor of 200 or greater in going from room temperature to 600 F, and leakage, in some cases, increases by nearly the same factor. Even the most precisely manufactured seal ring assembly exhibits this characteristic, since complete conformability with all mating parts is achieved only in theory. Experience and test results, however, have shown that sufficient leakage control can be achieved at high temperatures with the use of properly made, leak tested parts.

4.7.1.3 Diameter: Leakage, in general, increases with diameter due to the increased number of minute surface irregularities present on the increased contact areas and because of the larger seal ring cross-sections which are somewhat less conformable than those of smaller diameter seal ring assemblies. Within the diameter range of 1/2 in. to 6 in. into which most seal ring applications fall, it can be stated, generally, that two piece seal ring assemblies can be sufficiently controlled to assure reasonably predictable leakage results.

4.7.1.4 Motion: Perhaps the largest single variable in the prediction of dynamic leakage is the nature of piston motion. Frequency, velocity, and length of stroke; curvilinear rod and piston paths; and large pressure pulsations can unseat the seal ring assembly, temporarily, from the actuator bore or the downstream groove side. Also, at the point of reversal of piston direction or upon pressure reversal, the seal ring assembly can become unseated from the groove side, temporarily, with resulting leakage. These conditions, however, are transient only and occur over such short periods of time that the leakage increase is, at best, very difficult to detect.

4.7.1.5 System Cleanliness: Dirt or excessive wear particles in the hydraulic system can cause leakage by lodging between the sealing contact surfaces. For best results, filters of 5 to 10 micron ratings are recommended.

4.7.2 Mating Parts Requirements: Case history has shown that a seal ring assembly is only as good as the lowest quality part with which it mates. Seal ring manufacturers, since they cannot control the quality of mating parts, sometimes suggest that actuator parts be made to a quality level frequently unobtainable. Industry survey indicates, however, that there is an attainable and reasonable level of quality for actuator hardware, which allows a well-made seal ring assembly to function properly with an acceptable leakage rate. (See Table B.)

4.7.2.1 Actuator Bore: Seal ring assemblies that are made to be light tight at a given gage diameter, cannot be expected to seal perfectly in bores which have large diameter tolerances or excessive out-of-roundness. It is recommended, therefore, that bore out-of-roundness be limited to a maximum of .0005 in. per in. of diameter and that bore diameter tolerances per Table B, Column D, be considered maximum allowable, for optimum results. Bore taper, even with good control of circularity, can cause leakage and should not exceed .0005 in. per in. of cylinder length. Surface finish, although not as critical as other bore characteristics, should be 16 RMS or better for diameters up to and including 5 in., and 32 RMS or better for diameters greater than 5 inches. Bore lead-in angles should not exceed 30 deg (20 deg is most commonly used); the minimum length of lead-in should be equal to 75% of the ring width.

4.7.2.2 Cylinder Breathing: One of the most serious problems confronting the successful performance of a seal ring assembly is expansion and contraction of the actuator cylinder due to temperature and/or pressure fluctuations. This phenomenon, called cylinder breathing, is usually localized or non-uniform and tends to deteriorate, temporarily, the quality of bore taper, circularity, and piston alignment, causing an increase in leakage rate. It is recommended that, where possible, actuator cylinders be designed to limit cylinder breathing to .002 in. per in. of diameter.

4.7.2.3 Piston Groove: Groove sides should be square relative to the center line of the bore, if the seal ring assembly is to be expected to seal properly on both its primary and secondary sealing surfaces. Out-of-squareness should not exceed .0005 in./in. and all taper must be outward from the groove root; zero inward taper. Tests indicate that the surface finish of the groove sides has a greater effect on leakage than bore finish and should be 16 RMS or better. Groove side flatness is also important, since surface waviness can cause leakage even though side finish and groove width are properly controlled.

4.8 Leakage Testing: It can be seen from the number of variables involved that the accuracy of any prediction of dynamic leakage past a two piece seal ring assembly is dependent upon not only the seal ring, but the general condition of the hydraulic system itself. In order to eliminate some of these variables, it is recommended that two piece seal ring assemblies be statically leak tested by the seal ring manufacturer. Varying degrees of control of seal ring tolerances, finishes, and general quality can be established by the seal ring manufacturer in order to produce three major classes of seal ring assemblies.

4.8.1 Test Fluid: Initially, seal ring manufacturers and others involved in leak testing used JP-4 fuel as the test fluid, since its 0.8 centistoke viscosity at room temperature was similar to or less than the viscosity of most popular hydraulic fluids at their operating temperatures. Due to the obvious danger involved in working with a jet fuel, it became necessary to find a fluid of similar viscosity with a higher flash point. One such fluid, which has been found satisfactory and is used for viscosity calibration by many companies, is MIL-F-7024A, Type II; its room temperature viscosity is about 1.1 centistokes.

4.8.2 Test Pressures: It has been found that, for a hydraulic system such as that described herein, two test pressures can be used to determine the quality of seal ring assemblies: 750 psi and 3000 psi. Static, room temperature leakage tests at these two pressures give good indications of sealing quality for any type of seal assembly designed for use in aircraft-type hydraulic systems.

4.8.3 Seal Classification By Leak Test: The following three classes of two piece seal ring assemblies have been shown to provide adequate sealing quality in high temperature hydraulic systems, when statically leak tested at room temperature in MIL-F-7024A, Type II, at 750 psi and 3000 psi.

4.8.3.1 Class I: For diameters from 1/2 to 2-1/2: 10 cc/minute, or less.

4.8.3.2 Class 2: For diameters from 2-1/2 to 5: 25 cc/minute, or less.

4.8.3.3 Class 3: For diameters from 5 to 7: 50 cc/minute, or less.

4.8.4 Leak Test Cycle: The seal ring assembly should be installed in a heavy wall ring gage, or other suitable test fixture, with a 16 RMS finished bore of diameter D_o per Table A. A piston plug of suitable diameter per Table B, Column C, should be used to simulate the downstream groove side. Fluid per MIL-F-7024A, Type II, at room temperature should be applied to one side of the seal ring assembly at 3000 psi for no more than 2 minutes. Leakage should then be checked prior to the end of the third minute at pressure. After the leakage at 3000 psi has been measured, the pressure should be dropped to 750 psi and the leakage should be checked within one minute. This test should be repeated, then, on the opposite side of the seal ring assembly. Seal ring assemblies which exceed the specified maximum allowable leakage rate when tested per the above, should be considered unacceptable.

4.9 Frictional Drag Considerations: Seal ring assemblies, as discussed herein, have been designed for optimum conditions of leakage, strength, and frictional drag. At pressures above approximately 100/200 psi, where most hydraulic systems operate, the mechanical tension of the seal ring assembly is negligible in calculating drag forces. Therefore, in general hydraulic application, the axial drag force is directly proportional to the hydraulic system pressure and can be calculated using the formula:

$$H_A = \pi \quad D_o \quad b_o \quad \mu \quad \frac{\Delta\ p}{2}$$

where: H_A = Axial drag force due to hydraulic pressure (in pounds)

D_o = Gage diameter of seal ring assembly (in inches)

b_o = Outer ring width (in inches)

μ = Coefficient of friction of ring on cylinder

$\Delta\ p$ = Differential pressure across seal ring assembly (in psi)

It can be seen from the formula that drag force is also directly proportional to the seal ring width and that it can be reduced by decreasing the width of the seal ring. Narrower rings, however, create shorter flow lengths (thereby increasing leakage) and weaker cross-sections. Therefore, it is believed that the dimensions of Table A should be considered optimum.

4.10 General Application Considerations: Survey results indicate that the two piece seal ring assembly, as described herein, is compatible with all known hydraulic fluids, including the commercial non-flammable and the silicone base fluids. Adequate material compatibility and wear life are exhibited with all popular cylinder materials, except plain aluminum and soft 300 series stainless steels; excellent wear and compatibility were reported when the seal ring was used against chrome plate, hardened steel, nitride, and tool steel. Within the scope of this report, no problems relative to corrosion, temperature, pressure, friction, break-in, life, or failure were experienced by the industries surveyed.

4.11 Installation Procedures: Two piece seal ring assemblies should be installed individually, one ring at a time, starting with the inner ring. Opening (installation) stresses have been calculated for all parts tabulated herein, based upon the following procedure. The inner ring should be opened tangentially at the joint to a diameter just large enough to slide over the end of the piston and down into the ring groove. The same procedure should be followed for installation of the outer ring, taking care that the inner ring and outer ring joints are installed 180 deg apart.

5. OTHER TYPES OF SEAL RINGS

Isolated reports of other types of sealing devices, including three piece seal ring assemblies (two straight cut outer rings and one straight cut inner ring), two piece AMS 7310 Grey Cast Iron seal rings, and single piece stainless steel seal rings were received in answer to the survey, but these designs were used only in experimental systems of higher temperature and/or pressure or in lower class industrial hydraulic systems where higher leakage rates were tolerable.

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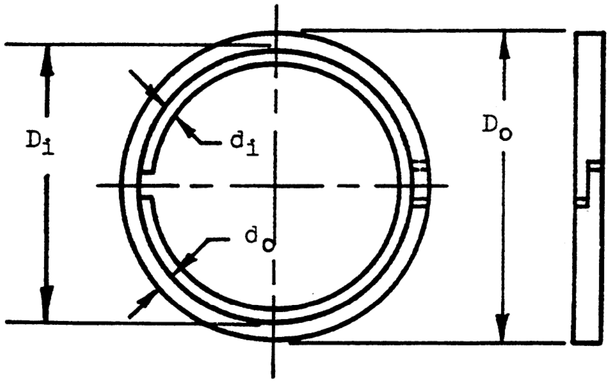

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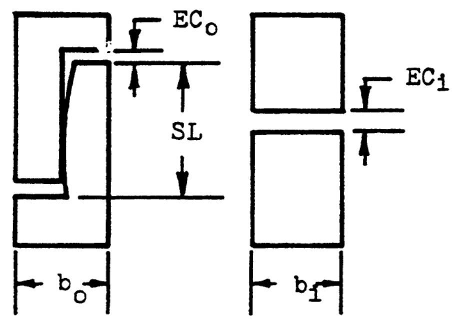


ENLARGED VIEW OF JOINTS

D_o = GAGE DIAMETER (OR OUTER RING GAGE DIAMETER); THIS DIMENSION EQUALS THE MINIMUM CYLINDER I.D.

D_i = INNER RING GAGE DIAMETER.

d_o = OUTER RING WALL.

d_i = INNER RING WALL.

b_o = OUTER RING WIDTH.

b_i = INNER RING WIDTH.

SL = STEP LENGTH

EC_o = OUTER RING END CLEARANCE.

EC_i = INNER RING END CLEARANCE.

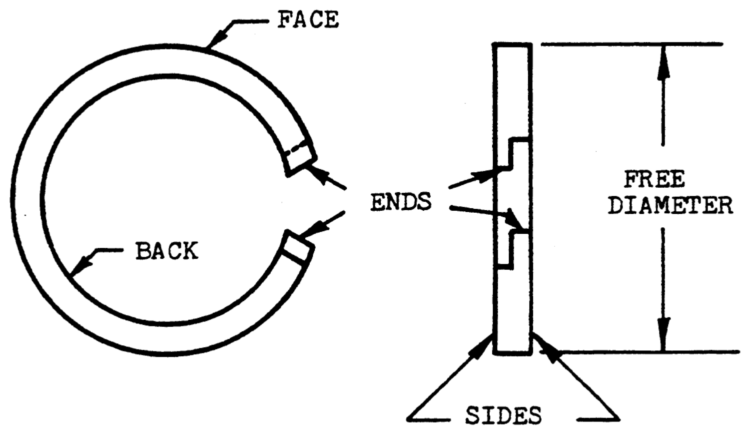
NOTES:

1. EC_o IS MEASURED WITH THE OUTER RING INSTALLED IN A RING GAGE OF D_o DIAMETER.
2. EC_i IS MEASURED WITH THE INNER RING INSTALLED IN A RING GAGE OF D_i DIAMETER.
3. WHEN THE INNER RING & THE OUTER RING ARE DISASSEMBLED AND ALLOWED TO REST SEPARATELY IN THEIR FREE POSITIONS, THE END CLEARANCES (EC_o & EC_i) ARE CALLED FREE GAPS (FG_o & FG_i, RESPECTIVELY).

FIGURE I

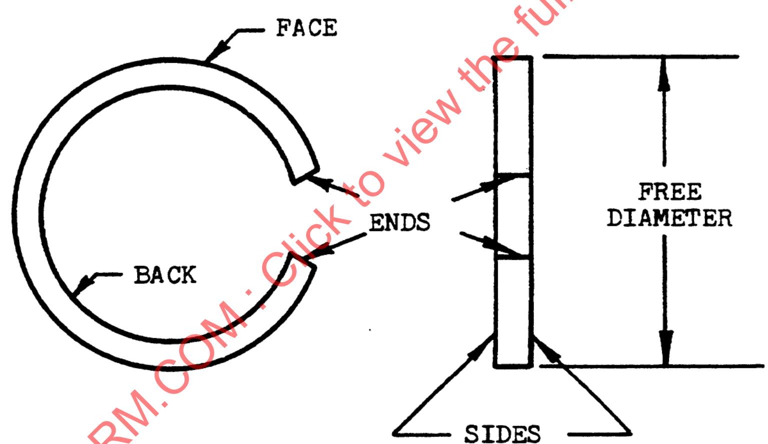
TWO PIECE BI-DIRECTIONAL SEAL RING ASSEMBLY

(SHOWN CLOSED TO ITS OPERATING DIAMETER)



STEP JOINT OUTER RING

(SHOWN IN FREE POSITION)

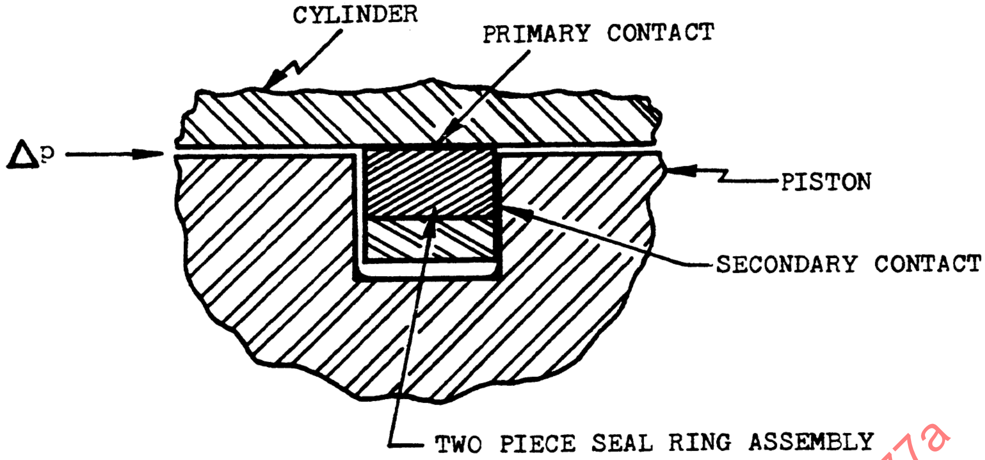


STRAIGHT CUT INNER RING

(SHOWN IN FREE POSITION)

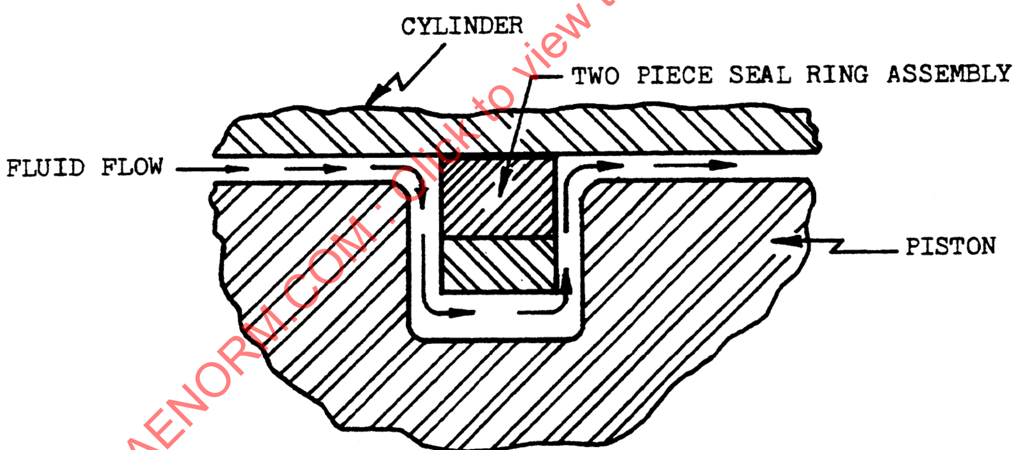
FIGURE II

DISASSEMBLED TWO PIECE SEAL RING ASSEMBLY



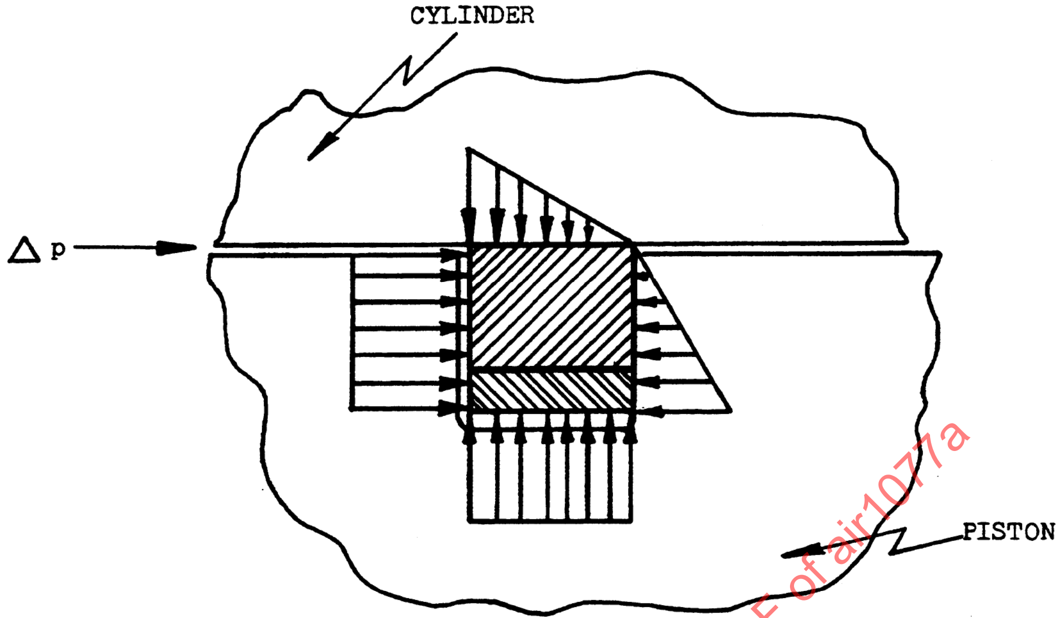
CROSS-SECTIONAL VIEW OF TWO PIECE SEAL RING ASSEMBLY
IN ITS SEALING POSITION

FIGURE III



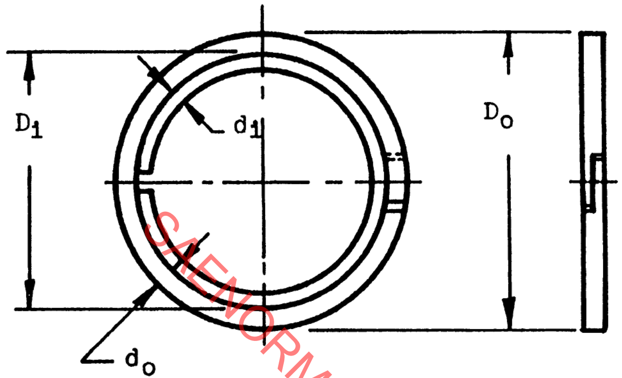
INITIAL FLOW PAST SEAL RING UPON APPLICATION
OF SYSTEM PRESSURE

FIGURE IV

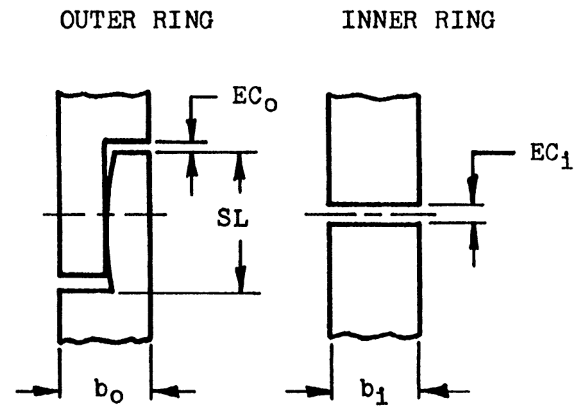


HYDRAULIC FORCES ON TWO PIECE SEAL RING ASSEMBLY
IN SEALING POSITION

FIGURE V



ASSEMBLE AS INDICATED ABOVE



ENLARGED VIEWS OF JOINTS

MATERIAL:
OUTER RING: HIGH TEMPERATURE ALLOYED CAST IRON; HARDNESS: 220-297 BHN
INNER RING: 17-4PH STAINLESS STEEL, AMS 5643 OR AMS 5398; HARDNESS: Rc 30-40

SURFACE TREATMENT:
OUTER RING: PARCO LUBRITE #2, OR EQUIVALENT
INNER RING: NONE

NOTES:

1. ASSEMBLY TO BE 100% LIGHT TIGHT IN GAGE OF SPECIFIED DIAMETER AND 100% LIGHT TIGHT BETWEEN INNER & OUTER RINGS FOR A DISTANCE EXTENDING 20 DEG EITHER SIDE OF OUTER RING STEP JOINT; LIGHT WHICH CAN BE PRESSED OUT WITH MODERATE FINGER PRESSURE (NOT EXCEEDING 5 LBS. PER INCH OF DIAMETER) IS ACCEPTABLE.
2. UNLESS OTHERWISE SPECIFIED SURFACES TO HAVE 125 RMS FINISH.
3. O.D. & I.D. OF OUTER TO BE FINISHED 20 RMS, SIDES TO BE FINISHED 16 RMS, AND MATING SURFACES OF STEP JOINT TO BE FINISHED 90 RMS BEFORE PARCO.
4. O.D. OF INNER TO BE FINISHED 20 RMS AND SIDES TO BE FINISHED 16 RMS.
5. O.D. EDGES OF OUTER MAY HAVE .003 MAX. RADIUS UP TO 4 INCHES; OVER 4 INCHES .005 MAX. RADIUS. I.D. EDGES OF INNER MAY HAVE .015 MAX. RADIUS. ALL EDGES SHALL BE FREE OF BURRS.
6. END CLEARANCE OF EACH RING TO BE MEASURED INDIVIDUALLY IN GAGE OF SPECIFIED DIAMETER.
7. TENSION CONTROLLED BY INNER RING FREE GAP.
8. STATIC LEAKAGE IN MIL-F-7024A, TYPE II @ ROOM TEMPERATURE @ 750 PSI & 3000 PSI MUST NOT EXCEED 10 CC/MIN. FOR GAGE DIAMETERS UP TO 2-1/2 INCHES; 25 CC/MIN. FOR GAGE DIAMETERS FROM 2-1/2 to 5 INCHES, AND 50 CC/MIN. FOR GAGE DIAMETERS FROM 5 to 7 INCHES.

TWO PIECE METALLIC SEAL RING ASSEMBLIES

TABLE A
(Page 1 of 2)