



AEROSPACE RECOMMENDED PRACTICE

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Superseding ARP1833B

Sealing Techniques for Missile Applications

RATIONALE

This document is being revised as part of the SAE Five-Year Review policy to add metric equivalents, add a note about temperature controlled inspection, and correct some typographical errors.

FOREWORD

Missile hydraulic and pneumatic component designers have been handicapped by the absence of concise design criteria for two difficult sealing conditions usually existing in missile applications as follows:

- Static pressure condition - Low pressure for long periods in a cyclic temperature environment (i.e., long-term storage requirements).
- Dynamic pressure condition - High pressures are suddenly applied in an extreme temperature environment (i.e., operational firing requirement).

Each of the two conditions listed above is frequently required to be satisfied by a single seal in a missile hydraulic or pneumatic component with sealing requirements changing abruptly from the first to the second set of conditions.

This design standard is intended to facilitate more nearly optimum designs by presenting specific recommendations in the areas of materials, finishes, configurations, and inspection criteria that past experiences show to be desirable and prudent.

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1. SCOPE

The purpose of this recommended practice is to provide the missile hydraulic and pneumatic component designer with information learned, tested, and substantiated in the correction of problems and failures experienced with seals that are subject to the unique requirements of missile static storage and subsequent dynamic operational conditions.

2. REFERENCES

The following publications form a part of this document to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of other publications shall be the issue in effect on the date of the purchase order. In the event of a conflict between the text of this document and the references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

2.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org.

AMS7379	Rubber: Fluorocarbon Elastomer (FKM), Low Temperature Sealing Tg -40 °F (-40 °C), 70 to 80 Type 'A' Hardness for Elastomeric Seals in Aircraft Engine Oil, Fuel and Hydraulics Systems
AIR786	Elastomer Compatibility Considerations Relative to Elastomer Selection
AIR1412	Designing for Long Life With Elastomers
ARP1231	General Gland Design Criteria for static and Dynamic O-Ring Seal Applications Specifically for Engines and Engine Control Systems
ARP1234	Static Axial O-Ring Seal Applications Without Anti-Extrusion Devices for Engine and Engine Control Systems
AS568	Aerospace Size Standard for O-Rings
AS4716	Gland Design, O-Ring and Other Seals
AS5857	Gland Design, O-Ring and Other Elastomeric Seals, Static Applications
AS6235	Face Seal Gland Design, Static, O-Ring and Other Seals for Aerospace Hydraulic and Pneumatic Applications

2.2 ASME Publications

Available from ASME, P.O. Box 2900, 22 Law Drive, Fairfield, NJ 07007-2900, Tel: 800-843-2763 (U.S./Canada), 001-800-843-2763 (Mexico), 973-882-1170 (outside North America), www.asme.org.

ASME B46.1-2019 Surface Texture (Surface Roughness, Waviness, And Lay)

2.3 ASQ Publications

Available from American Society for Quality, 600 North Plankinton Avenue, Milwaukee, WI 53203, Tel: 800-248-1946 (United States or Canada), 001-800-514-1564 (Mexico) or +1-414-272-8575 (all other locations), www.asq.org.

ANSI/ASQ Z1.4 Sampling Procedures and Tables for Inspection by Attributes

2.4 ISO Publications

Copies of these documents are available online at <https://webstore.ansi.org/>.

ISO 3601-2 Fluid power systems - O-rings - Part 2: Housing dimensions for general applications

ISO 3601-3 Fluid Power Systems - O-rings - Part 3: Quality acceptance criteria

3. ELASTOMERIC O-RING SEAL GUIDELINES

3.1 General Introduction

The O-ring is widely used as a seal element in a broad range of missile hydraulic and pneumatic component designs. It has been variably successful as attested by a continuing family of problems and failures associated with seal designs, especially in low-temperature and low-pressure, long-term storage applications.

In some cases, the problem of failure has been directly attributed to inadequacies of the O-ring as a seal element choice (i.e., a simple misapplication). In other cases, the problem or failure cause was traced to improper choice of O-ring material properties and dimensions; inadequate design features interfacing with the O-ring; damage and abuse; quality controls; and insufficient testing of the design to detect and correct design deficiencies before committing the design to production.

It is essential that the missile hydraulic and pneumatic component designer has a working understanding of the O-ring as a candidate element for seal design, is aware of its functional limitations, and has some guidance for approaching design requirements which are not clearly defined in the existing published design guides to general O-ring usage.

This section of the standard is intended to supply information learned, tested, and substantiated in the correction of problems and failures experienced with O-rings used in missile hydraulic and pneumatic components.

3.2 Seal Principle Discussion

The O-ring is made of highly elastic materials, performing its function in the deformed condition through its ability to reliably store elastic energy and use this energy to maintain sealing pressures against mating interfaces.

Mechanical deformation of the O-ring by the surrounding structure or by the pressure of confined fluids, individually or in combination, is used to force the material into potential leak paths to provide a positive block to the flow of the confined fluid.

When the O-ring is deformed solely by the preloads applied by the mating structure, and the resulting interface pressure is sufficient to provide the effective seal over the full range of functional and storage conditions without being assisted by the confined fluid pressure, it can be considered an ordinary gasket.

When the O-ring is primarily deformed by the pressure forces of the confined fluid, and secondarily by the structural and environmental effects, it must not be treated as a gasket.

The designer using the O-ring element must understand that the difference in deformation mode can be critical to successful sealing.

In low-pressure (below approximately 300 psig [2068 kPa]), low-temperature (dependent on material) design, the seal interface pressure must be achieved by maintaining adequate structurally applied preload compression in the O-ring, since fluid pressures in this range will usually not adequately deform O-rings made of known materials in the available hardness (durometer) ranges to form a seal.

In pressures ranging above 300 psig (2068 kPa), and up to the extrusion pressure of the particular O-ring material choice, the fluid pressure is generally sufficient to deform the O-ring to provide effective interface pressures.

There are some special environmental conditions, such as low temperatures approaching the glass-transition of the material, the effect of which may best be determined by development testing as described herein.

In a robust static seal design, the sealing is obtained principally by the mechanical deformation of the O-ring as described above. If lubricant is used in a static seal, its primary purpose should be to help in installation (usually for static radial seals; face seals should generally not require lubrication). Lubricants generally provide minimal (if any) sealing for static seals. Hence, for static seals, use lubricant sparingly (see 3.5), especially since excess lubricant can migrate to undesirable areas, which can have negative consequences. For dynamic seals, if lubricant is used, then it should be sufficient (but not excessive) to reduce friction in the relative movement of the seal against the dynamic surface.

3.3 Common Failure Causes

Most failures of missile hydraulic and pneumatic component O-ring seals occur at low-pressure and low-temperature extremes and can be attributed to any of the following causes:

- a. Insufficient elastomeric force caused by a variety of factors such as:
 - Inadequate design squeeze (squeeze means a change in shape due to surrounding structures, whereas deformation may be caused by confined fluid pressures as well)
 - Excessive compressive set of seal materials
 - Inadequate durometer rating (i.e., too low)
 - Inadequate elastic compliance
 - Inadequate elastic temperature range
 - Reduction in volume due to migration of material additives
- b. Spiraling of seals during operation in a reciprocating application or during assembly. This condition is generally associated with any of the following:
 - Inadequate stiffness of the O-ring cross-section
 - Excessive eccentricity
 - Inadequate lubrication
 - Inadequate assembly procedures or aids
 - Discontinuities and/or protrusions on the flash line
 - Incompatible surface finishes or conditions
 - Long stroke cylinders
- c. Contamination lying across the seal contact surfaces.
- d. Handling or assembly damage.
- e. Molding or processing flaws in the seal.
- f. Extrusion of the seal during prior operation or environmental exposure due to either excessive design clearance or structural deformation under pressure or due to the gland being undersized and/or the O-ring being oversized.

- g. Excessive wear of dynamic seals due to microscopic pits and discontinuities in contacting surfaces such as anodized aluminum.
- h. Porosity or other defects in contacting surfaces.
- i. Physical properties or dimensions that are not adequately controlled.
- j. Improper selection of seal materials and/or lubricants.
- k. Stretch too high; causing premature aging (see 3.7).
- l. Ignoring the effect of eccentricity in radial seals, which can cause a leak at a specific location of the 360-degree circumference even if the seal holds well at all other locations of the circumference (especially relevant for high-pressure or vacuum seals).
- m. Surface roughness too smooth for dynamic seals causing excessive wiping off of lubricant (similar to a squeegee) leading to an increase in friction and possible abrasion and/or wear out of the elastomeric seal. Surface roughness for dynamic seal glands sealing surface should range between 4 to 16 microinches (0.1 to 0.4 μm) Ra. Excessively rough gland surface can lead to abrasion and/or wear out of the elastomeric seal for dynamic seals. It can also lead to leakage in static seals even if the squeeze is within the desired range.

3.4 Development and Qualification Test Recommendations

Due to the time and cost constraints, development and qualification tests normally do not evaluate the effects of:

- Long-term storage and field service
- All possible variations in the material properties
- All possible dimensional and finish variations

Because of the factors above, development tests should selectively include conditions that exceed the maximum specified environmental conditions. Marginal seal designs are prone to be revealed by exposure to temperatures that exceed specified extremes followed by leakage tests at:

- Low temperature and low pressure
- High temperature and low pressure
- High temperature and high pressure applied suddenly
- Low temperature and high pressure applied suddenly

The above tests are intended to encompass all extreme conditions. Usually, the most adverse combination of conditions is exposure to high temperature followed by leakage tests at low temperature (with high pressure applied suddenly).

Development test hardware should exceed design tolerances (i.e., margin excess involves consideration of the critical nature of the seal, machining capabilities, cost, and time constraints) in each of the following areas:

- High diametral clearances
- High gland eccentricities
- Rough finishes
- High and low preload squeeze conditions

Each vendor elastomeric compound for use should be qualified for specific seal applications. Specific material (i.e., physical characteristics) verification tests should be conducted on each vendor compound along with functional seal qualification tests.

Qualification by similarity should not be used to justify the omission of functional qualification tests for critical seal applications.

The use of plastic models and leak path isolation fixtures as developmental test beds are important in evaluating sealing effectiveness, problems, and failures.

The use of specialized leak detection and measurement equipment (i.e., excellent commercial detectors are available) is invaluable in understanding the margins of seal efficiency.

3.5 Design Guide for High Reliability

Efforts to achieve high reliability in an O-ring seal design will involve several of the following factors:

- Use of specific vendor elastomeric compounds that have demonstrated overstress margin of sealing qualities during and after exposure to fluids and specified environmental conditions. AIR786 provides initial guidance on establishing elastomer/fluid compatibility.
- Tightened inspection of critical O-rings for specific types of defects and mechanical properties, and of mating surfaces for dimensions and surface finish. Where possible measurements should be taken at a stabilized (for at least 1 hour) temperature of $73^{\circ}\text{F} \pm 2^{\circ}\text{F}$ ($23^{\circ}\text{C} \pm 1^{\circ}\text{C}$).
- Selective deviation from conventional gland dimensions, usually in the direction of increased squeeze and reduced gland volume. This provides increased interface pressure and is especially important in low-pressure (or vacuum), low-temperature applications. The volume of the gland must be at least equal to the volume of the seal under the worst-case condition of seal swell and dimensional tolerance.
- Use of the largest standard O-ring cross-section and the highest standard durometer permitted by available space and by assembly damage considerations. In static low-pressure applications, these features facilitate the achievement of higher interface pressures, in conjunction with nonstandard gland dimensions. In reciprocating seal applications, these features minimize the probability of spiraling.

NOTE: The statement regarding the highest durometer rating is based upon extensive testing in 1976 on low-pressure seals for the Raytheon AIM-54 Phoenix missile. The statement may seem controversial but has been found to be consistent with O-ring seal theory.

- The selective use of cap strips for reciprocating seal applications. Cap strips offer the advantage of improved wear resistance, reduced extrusion, and reduced friction. Cap strips made of appropriate materials such as Virgin Polytetrafluoroethylene (PTFE) or UHMWPE (Ultra High Molecular Weight Polyethylene) can be particularly effective in reducing break-out friction after long-term storage. If a cap strip with a profile to approximately match the shape of the installed O-ring is used, the increased squeeze on the O-ring produces an increased load and may increase friction for low-pressure applications. Care should therefore be taken to ensure the appropriate performance of the application.
- However, cap strips have the disadvantage of reduced sealing reliability in certain low-pressure, low-temperature applications.
- The use of backup rings to prevent extrusion in high-pressure, high-clearance, and high-temperature applications. However, the use of backup rings may cause excessive break-out friction in certain pneumatic applications.
- The selective and controlled use of lubrication. Lubrication is generally to be used only as an installation aid (see 3.2).
- Use of spring-energized PTFE or UHMWPE U-cups to overcome environments hostile to O-rings especially if dynamic break-out friction is a problem.
- AIR1412 contains helpful suggestions on designing for long life with elastomers.

3.6 Design Tips - Positive Considerations

The following should be considered when designing hydraulic missile components:

- Consider creating source control drawings to define O-ring requirements as well as the necessary quality controls. Drawings should include the following:
 - a. Where practical, the specification of standard sizes and tolerances per AS568, and where necessary, the use of non-standard sizes and/or tolerances.
 - b. Reference to the appropriate material specification. Additional requirements should be stated where necessary.
 - c. Approved vendor compound numbers. The approved compounds should be strictly limited to those that have demonstrated suitability during qualification test programs.
 - d. Inspection for flaws per ISO 3601-3, Grade S or CS. The minimum inspection should be a 1.0 AQL, Level II, per ANSI/ASQ Z1.4. Consider 100% inspection for critical seal applications.
- Specify O-rings with the largest standard cross-section and highest durometer rating permitted by space, material, and assembly considerations (for reasons given in 3.5).
- Control the gland's total eccentricity to a practical minimum for dynamic radial seals. Lowering gland eccentricity may be limited by how tight the dimensional tolerances can be. Lowering dimensional tolerances will lower total gland eccentricity but tightening dimensional tolerances beyond a certain level may not be financially and/or technically viable.
- Control cylindrical sealing bore taper to a practical minimum for reciprocating dynamic seals.
- Consider designing gland widths as narrow as practical when not covered by existing standards. However, the gland should accommodate the volume of the seal under maximum material and swell conditions.
- Face seal designs should provide sufficient structural rigidity to minimize the opening of extrusion gaps during high-pressure exposures. When weight is important, high-pressure face seals may be a bad design choice unless parts are inherently stiff for other reasons.
- Where pressure dictates, use backup rings designed on a case-by-case evaluation of compatible material, assembly, and environmental factors, especially in high-temperature high diametrical clearance applications. Select continuous, rather than scarf-cut, backup rings if assembly, dynamic friction, and environmental factors allow.
- Consider the incorporation of design features and processes which give a margin against assembly damage.
- Provide the most practical edge relief possible in holes, slots, and other discontinuities which cannot be avoided, and which O-rings must pass over during assembly or functional operation. Putting a blend radius on a sharp corner is preferred over an edge break.
- Minimize the number of seals and the total linear length of seals in all component designs.

3.7 Design Tips - Negative Considerations

The following should be avoided when designing hydraulic missile components:

- Avoid specifying O-rings by only standard specifications (i.e., MS, AMS, AS, etc.) in critical seal applications.
- Avoid the use of acrylonitrile butadiene (NBR) material for external static seals for long-term storage or service especially if the application includes either of the following conditions:
 - a. Sealing pressures from zero to approximately 300 psi (2068 kPa) while being subject to extreme cyclic changes in temperature (i.e., from below -25 °F [-32 °C] to above +160 °F [71 °C]).

- b. Functional operation at temperatures below -25 °F (-32 °C) especially after long-term cyclic temperature changes noted in condition (a) above.
- Avoid gland surface finishes greater than 32 microinches (0.8 µm) Ra or less than 4 microinches (0.1 µm) Ra. In critical dynamic applications, avoid surface finishes greater than 16 microinches (0.4 µm) Ra. High-pressure or vacuum applications may require a surface finish better than 16 microinches (0.4 µm) Ra.
 - Avoid designs that require excessive O-ring stretch (i.e., greater than 50%) during assembly.
 - Avoid installed stretch exceeding 5% based on nominal O-ring dimensions.
 - Avoid, if possible, entrance angles greater than 15 degrees half-angle measurement and entrance cone diameters less than the free outside diameter of the O-ring.
 - Avoid gland designs in which the O-ring is required to seal more than one circumferential gap.
 - Avoid, if possible, designs that require the O-ring to slide over holes, slots, or other discontinuities with sharp edges. If not possible to avoid edges, then put a blend radius on all such edges; if not possible, then at least break the edges. (Use the numerical values for edge breaks listed in Tables 2 through 15 for guidance.)
 - Avoid, if possible, non-circular face seal gland designs.
 - Multiple O-rings in the same groove may seem an attractive option when physical space is limited, as is often the case in missiles. For example, sometimes both an EMI (Electromagnetic Interference) and an environmental seal are necessary, and the same seal may not be able to fulfill both functions adequately. However, using multiple separate O-rings in the same groove is not recommended. If it is absolutely necessary to have dual seals in the same groove, then use co-extruded or co-molded seals.

3.8 Detail O-Ring Gland Design

3.8.1 General Description

Figure 1 and Tables 1 through 16 outline configuration and O-ring gland design information for various conditions encountered in missile hydraulic and pneumatic component designs. The information represents experience gained in the development of several missile systems; however, in certain areas, the information represents extrapolated data. It is anticipated that the information will facilitate more nearly optimum designs; however, design configurations should be proven by qualification testing as recommended in 3.4.

3.8.2 Gland Design Criteria

Inch units should be used to carry out the detail gland design to ensure tolerances and sizes are specified with the correct precision recommended by this ARP. Gland dimensions may be converted to SI units if required after the sizes have been calculated in inches. Metric equivalent sizes in Table 1B through Table 15B are presented for information only.

The information contained in the tables presented in this section is similar to that in AS4716, with the primary differences as follows:

- a. The gland volumes are smaller. Some of the glands are as little as 7% greater than the O-ring volumes under maximum material conditions. If a volumetric swell greater than 7% is anticipated, the groove volume should be increased accordingly by increasing the width.
- b. The gland edge breaks are sharper in high-pressure applications.
- c. The squeeze is higher except in the dynamic pneumatic applications.
- d. The diametral clearance is tighter in high-pressure applications without backup rings.
- e. Cylinder bore sizes below 1 inch (25.40 mm) generally conform to standard reamer sizes (see Table 16).

- f. Rod gland bore sizes below 1 inch (25.40 mm) generally conform to standard reamer sizes (see Table 16) from which the rod and groove diameters can be calculated using clearance D per Figure 1.
- g. The gland depth tolerances are generally tighter through size 020.
- h. The allowable eccentricities are smaller in high-pressure and/or dynamic applications.
- i. The sides of the grooves are perpendicular to the axis of the gland rather than being a maximum of 5 degrees from perpendicular.

NOTES:

1. Squeeze computations for the tables do not consider the reduction in O-ring cross-section due to installed stretch. If the stretch is kept below the recommended 5% per 3.7 then the error due to this is small but it will not be negligible if the stretch is substantially greater than 5%.
2. There are other standards available for review and comparison. These include: AS5857, ARP1234, ARP1231, AS6235, and ISO 3601-2.

3.8.3 Index of Figures and Tables

Figure 1 - Conventional glands design applications

Table 1A - Recommended bore and shaft dimensions in inches

Table 1B - Recommended bore and shaft dimensions in millimeters for information only

Tables 2A through 15A - Recommended gland dimensions in inches, surface finishes, inspection levels, and material hardnesses for various applications

Tables 2B through 15B - Recommended gland dimensions in millimeters, surface finishes, inspection levels, and material hardnesses for various applications for information only

Tables 2A and 2B - 0 to 300 psi (0 to 2068 kPa) - Dynamic hydraulic applications

Tables 3A and 3B - 300 to 1500 psi (2068 to 10340 kPa) - Dynamic hydraulic applications

Tables 4A and 4B - 1500 to 3500 psi (10340 to 24130 kPa) - Dynamic hydraulic applications

Tables 5A and 5B - 0 to 300 psi (0 to 2068 kPa) - Static hydraulic applications

Tables 6A and 6B - 300 to 1500 psi (2068 to 10340 kPa) - Static hydraulic applications

Tables 7A and 7B - 1500 to 3500 psi (10340 to 24130 kPa) - Static hydraulic applications

Tables 8A and 8B - 0 to 500 psi (0 to 3447 kPa) - Dynamic pneumatic applications

Tables 9A and 9B - 500 to 3000 psi (3447 to 20680 kPa) - Dynamic pneumatic applications

Tables 10A and 10B - 3000 to 10000 psi (20680 to 68950 kPa) - Dynamic pneumatic applications

Tables 11A and 11B - 0 to 500 psi (0 to 3447 kPa) - Static pneumatic applications

Tables 12A and 12B - 500 to 3000 psi (3447 to 20680 kPa) - Static pneumatic applications

Tables 13A and 13B - 3000 to 10000 psi (20680 to 68950 kPa) - Static pneumatic applications

Tables 14A and 14B - Face seals for hydraulic applications

Tables 15A and 15B - Face seals for pneumatic applications

Table 16 - Recommended bore sizes for using standard-inch reamers

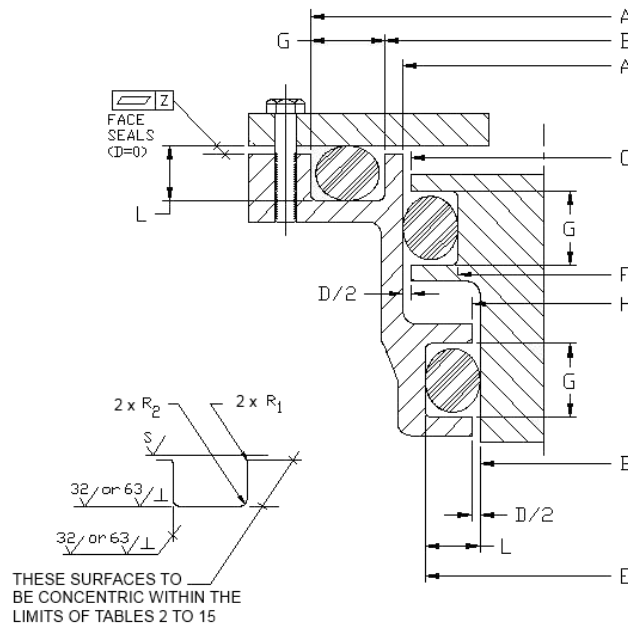
Table 17 - Summary of elastomeric materials

A decision as to which section of Tables 2 through 15 is appropriate will at times require a choice if an application encompasses:

- a. More than one set of conditions. In this case, the designer should choose the smaller clearance and eccentricity, sharper corner break, smoother finish, and tighter inspection requirements.
- b. Both the static and dynamic conditions. In this case, the better choice will probably be the lesser squeeze shown for dynamic applications to minimize the spiraling and friction potential, especially in reciprocating applications.

3.8.4 Drawing Diameter Calculation

The gland depths and diametral clearances shown in Figure 1 will not appear on engineering drawings, but are used to calculate certain diameters that will appear on the drawings as follows:



NOTES AND DEFINITIONS:

A – Cylinder Bore ID per Table 1A

B – Rod OD per Table 1A

C – Piston OD (= A-D)

D – Diametral Clearance per Tables 2A to 13A

E – Internal Gland ID (= B+2L)

F – Piston Gland OD (= A-2L)

G – Gland Width per Tables 2A to 15A

H – Rod Bore ID (= B+D)

L – Gland Depth per Tables 2A to 15A

R₁ – Corner Break Radius per Tables 2A to 15A

R₂ – Fillet Radius per Tables 2A to 15A

$\sqrt{S}$ – Finish of Adjacent Surface per Tables 2A to 15A, as defined in ASME B46.1-2019

Z – Surface Flatness per Tables 14A and 15A

Figure 1 - Gland design

a. For external seals:

- The rod/piston diameter C is calculated from dimension A per Table 1A and D per Tables 2A through 15A:

$$C = A + D \quad (\text{Eq. 1})$$

The tolerance on C is the tolerance on D minus the tolerance on A. (See further for special cases.) As a check:

$$D = A - C \quad (\text{Eq. 2})$$

where D must equal the value in Tables 2A through 15A, including the specified tolerances.

- The cylinder/bore diameter A is per Table 1A.
- The gland root diameter F is calculated from dimension A per Table 1A and L per Tables 2A through 15A:

$$F = A - 2L \quad (\text{Eq. 3})$$

The tolerance on F is twice the tolerance on L, minus the tolerance on A. As a check:

$$L = (A - F)/2 \quad (\text{Eq. 4})$$

where L must equal the value in Tables 2A through 15A, including the specified tolerances.

- The groove width and other parameters are per Tables 2A through 15A.

b. For internal seals:

- The rod/piston diameter B is per Table 1.
- The cylinder/bore diameter H is calculated from dimension B per Table 1A and D per Tables 2A through 15A:

$$H = B + D \quad (\text{Eq. 5})$$

The tolerance on H is the tolerance on D minus the tolerance on B. (See further for special cases.) As a check:

$$D = H - B \quad (\text{Eq. 6})$$

where D must equal the value in Tables 2A through 15A, including the specified tolerances.

- The gland diameter E is calculated from dimension B per Table 1A and L from Tables 2A through 15A:

$$E = B + 2L \quad (\text{Eq. 7})$$

- The tolerance on E is twice the tolerance on L minus the tolerance on B. As a check:

$$L = (E - B)/2 \quad (\text{Eq. 8})$$

where, L must equal the value in Tables 2A through 15A, including the specified tolerances.

- The groove width and other parameters are per Tables 2A through 15A.

c. For face seals, internally pressurized:

- The outside diameter is dimension A per Table 1A:

$$OD = A \quad (\text{Eq. 9})$$

- The inside diameter is calculated from dimension A per Table 1A and G from Table 14A or 15A:

$$ID = A - 2G \quad (\text{Eq. 10})$$

- The tolerance on the ID is twice the tolerance on G minus the tolerance on A. As a check:

$$G = (A - ID)/2 \quad (\text{Eq. 11})$$

where, G must equal the value in Tables 2A through 15A, including the specified tolerance.

- The groove depth and other parameters are per Table 14A or 15A.

d. For face seals, externally pressurized:

- The inside diameter is dimension B per Table 1A:

$$ID = B \quad (\text{Eq. 12})$$

- The outside diameter is calculated from dimension B per Table 1A and G from Table 14A or 15A:

$$OD = B + 2G \quad (\text{Eq. 13})$$

- The tolerance on the OD is twice the tolerance on G minus the tolerance on B. As a check:

$$G = (OD - B)/2 \quad (\text{Eq. 14})$$

where, G must equal the value in Tables 2A through 15A, including the specified tolerance.

- The groove depth and other parameters are per Table 14A or 15A.

In some cases, the tolerances that are specified for the diametral clearances D in Tables 2A through 15A (i.e., for high-pressure pneumatic applications, without backup rings) are equal to or tighter than the tolerances specified for the diameters A and B in Table 1A. In these cases, the tolerances in Tables 2A through 15A take precedence, and, therefore, the tolerances for A or B must be reduced appropriately.

EXAMPLE: Designing an internal seal of nominal size 013, for a low-pressure dynamic hydraulic application would result in the following steps:

a. The applicable dimensions, in inches, would be:

B = 0.435 ± 0.0005 per Table 1A
L = 0.055 ± 0.0005 per Table 2A
D = 0.004 ± 0.003 per Table 2A

b. The internal gland diameter is calculated using Equation 7:

$$\begin{aligned} E &= B + 2L \\ &= (0.435 \pm 0.0005) + 2(0.055 \pm 0.0005) \\ &= (0.435 \pm 0.0005) + (0.110 \pm 0.001) \end{aligned}$$

Adding the basic dimensions and subtracting the B bore tolerance from the gland depth tolerance:

$$E = 0.545 \pm 0.0005 \text{ (ANSWER)}$$

- c. The gland depth check using Equation 8 is:

$$\begin{aligned} L &= (E - B)/2 \\ &= [(0.545 \pm 0.0005) - (0.435 \pm 0.0005)]/2 \\ &= (0.110 \pm 0.001)/2 \\ &= 0.055 \pm 0.0005 \text{ (CHECK)} \end{aligned}$$

- d. The rod bore using Equation 5 is:

$$H = (0.435 \pm 0.0005) + (0.004 \pm 0.003)$$

Adding the basic dimensions and subtracting the B bore tolerance from the diametral clearance tolerance:

$$= 0.439 \pm 0.0025 \text{ (ANSWER)}$$

- e. The diametral clearance check using Equation 6 is:

$$\begin{aligned} D &= H - B \\ &= (0.439 \pm 0.0025) - (0.435 \pm 0.0005) \\ &= 0.004 \pm 0.003 \text{ (CHECK)} \end{aligned}$$

3.8.5 Size and Stretch Considerations

Designs that require shaft or bore sizes different from those in the tables should consider the following factors:

- a. Shaft Size - Excessive O-ring stretch in the installed state should be avoided. Specifically, the shaft diameter (diameter B or F per Figure 1) should be within the following limits:

Not more than 5% larger than the O-ring nominal ID.

Not smaller than the O-ring maximum ID.

NOTES:

1. These limits do not accommodate an infinite range of shaft sizes. Therefore, special O-rings and backup rings may be required.
2. Little information is available regarding the effects of O-ring stretch beyond 5%, except that it contributes to the aging of nitriles, particularly at elevated temperatures.
3. If the designer employs stretch above 5%, the following formula can be used to predict the resulting flattening of the O-ring:

$$y = 54 [1 - (6/\sqrt{36 + x})] \quad (\text{Eq. 15})$$

where:

y = percent loss of compression diameter due to stretch (i.e., if y = 4, the reduction in the O-ring cross-section is 4%)

x = percent stretch on the inside diameter (i.e., for 5% stretch, x = 5)

4. The formula is empirical and represents published data with a maximum error of 0.3% for x = 0 to x = 26.

- b. Bore Size - To comply with good design practices, an effort should be made to use bore sizes that coincide with standard reamer sizes. This is sometimes awkward when designing internal seals where the bore diameter (H per Figure 1) is determined by the rod diameter plus the clearance (B + D per Figure 1). The resulting dimension should be adjusted to coincide with a standard reamer size in the following cases:

The bore size is less than 1 inch.

The resulting O-ring stretch is compatible with the previously stated stretch limitations.

The resulting dimensions are compatible with other design requirements.

NOTE: Standard reamer sizes are shown in Table 16.

EXAMPLE: Consider the following rationale:

1. The bore diameter H of 0.439 ± 0.0025 derived in the previous example (i.e., 3.8.4) does not encompass one of the standard reamer sizes of Table 16.
2. Using the previously stated limits (i.e., 3.8.5), the rod diameter B can vary between 0.431 (the O-ring maximum ID) and 0.447 (5% greater than the nominal ID) or:

$$\begin{aligned} B &= 0.431/0.447 \\ &= 0.439 \pm 0.008 \end{aligned}$$

3. The bore diameter H, in this case, is 0.004 ± 0.003 larger than B or:

$$\begin{aligned} H &= (0.439 \pm 0.008) + (0.004 \pm 0.003) \\ &= 0.443 \pm 0.011 \end{aligned}$$

4. The calculated bore diameter H thus encompasses either the 7/16 (0.4375) or the 29/64 (0.4531) reamer size. Since either size is satisfactory, assume that the more common size 7/16 (0.4375) is selected.
5. The new size for the H bore diameter is 0.0015 smaller than the originally calculated diameter of 0.439. Therefore, each of the other two corresponding diameters is adjusted by the same amount and becomes:

$$\begin{aligned} B &= 0.4335 \pm 0.0005 \text{ (ANSWERS)} \\ E &= 0.5434 \pm 0.0005 \end{aligned}$$

Table 1A - Bore and rod dimensions (inches) for O-ring gland seal design

AS568 Uniform Dash No.	Standard O-Ring Size				Bore Dia External Seals		Rod Dia Internal Seals		Notes
	ID	+ TOL	W	+ TOL	A	+ TOL	B	+ TOL	
004	0.070	0.005	0.070	0.003	0.185 (0.189)	0.0005	0.074	0.0005	(a)(b)(c)(d)
005	0.101	0.005	0.070	0.003	0.213 (0.221)		0.105		(a)(b)(c)(d)
006	0.114	0.005	0.070	0.003	0.228 (0.2344)		0.123		(b)(c)(e)
007	0.145	0.005	0.070	0.003	0.261 (0.2656)		0.152		(b)(c)(e)
008	0.176	0.005	0.070	0.003	0.295 (0.2969)		0.183		(b)(c)(e)
009	0.208	0.005	0.070	0.003	0.323 (0.3281)		0.217		(b)(c)(e)
010	0.239	0.005	0.070	0.003	0.358 (0.3594)		0.248		(b)(c)(e)
011	0.301	0.005	0.070	0.003	0.4219		0.311		(b)(e)
012	0.364	0.005	0.070	0.003	0.4844		0.373		(b)(e)
013	0.426	0.005	0.070	0.003	0.552		0.435		(a)
014	0.489	0.005	0.070	0.003	0.617		0.498		(a)
015	0.551	0.007	0.070	0.003	0.6875		0.561		(a)
016	0.614	0.009	0.070	0.003	0.750		0.623		(a)
017	0.676	0.009	0.070	0.003	0.8125		0.686		(a)
018	0.739	0.009	0.070	0.003	0.875		0.748		(a)
019	0.801	0.009	0.070	0.003	0.9375		0.811		(a)
020	0.864	0.009	0.070	0.003	1.000	0.0005	0.873	0.0005	(a)
021	0.926	0.009	0.070	0.003	1.0625	0.001	0.935	0.001	(a)
022	0.989	0.010	0.070	0.003	1.125		0.998		(a)
023	1.051	0.010	0.070	0.003	1.1875		1.061		(a)
024	1.114	0.010	0.070	0.003	1.250		1.123		(a)
025	1.176	0.010	0.070	0.003	1.3125		1.186		(a)
026	1.239	0.010	0.070	0.003	1.375		1.248		(a)
027	1.301	0.010	0.070	0.003	1.4375		1.311		(a)
028	1.364	0.013	0.070	0.003	1.500		1.373		(a)
029	1.489	0.013	0.070	0.003	1.625		1.498		(a)
030	1.614	0.013	0.070	0.003	1.750	0.001	1.623	0.001	(a)

NOTES:

- (a) Not recommended for reciprocating applications.
- (b) Sizes noted require considerable stretch when installed in standard external grooves and may require compounds with superior elongation, or two-piece pistons.
- (c) Use dimensions in parentheses for dynamic pneumatic applications.
- (d) Not for use in face seal applications.
- (e) Not for use in internal pressure face seal applications.

Table 1A - Bore and rod dimensions (inches) for O-ring gland seal design (continued)

AS568 Uniform Dash No.	Standard O-Ring Size				Bore Dia External Seals		Rod Dia Internal Seals		Notes
	ID	+ TOL	W	+ TOL	A	+ TOL	B	+ TOL	
031	1.739	0.015	0.070	0.003	1.875	0.001	1.748	0.001	(a)
032	1.864	0.015	0.070	0.003	2.000		1.873		(a)
033	1.989	0.018	0.070	0.003	2.125		2.000		(a)
034	2.114	0.018	0.070	0.003	2.250		2.125		(a)
035	2.239	0.018	0.070	0.003	2.375		2.250		(a)
036	2.364	0.018	0.070	0.003	2.500		2.375		(a)
037	2.489	0.018	0.070	0.003	2.625		2.500		(a)
038	2.614	0.020	0.070	0.003	2.750		2.625		(a)
039	2.739	0.020	0.070	0.003	2.875		2.750		(a)
040	2.864	0.024	0.070	0.003	3.000		2.875		(a)
041	2.989	0.024	0.070	0.003	3.125		3.000		(a)
042	3.239	0.024	0.070	0.003	3.375		3.250		(a)
043	3.489	0.024	0.070	0.003	3.625		3.500		(a)
044	3.739	0.027	0.070	0.003	3.875		3.750		(a)
045	3.989	0.027	0.070	0.003	4.125		4.000		(a)
046	4.239	0.030	0.070	0.003	4.375		4.250		(a)
047	4.489	0.030	0.070	0.003	4.625		4.500		(a)
048	4.739	0.030	0.070	0.003	4.875		4.750		(a)
049	4.989	0.037	0.070	0.003	5.125		5.000		(a)
050	5.239	0.037	0.070	0.003	5.375	0.001	5.250	0.001	(a)
102	0.049	0.005	0.103	0.003	0.228	0.001	0.061	0.001	(b)(c)
103	0.081	0.005	0.103	0.003	0.257		0.092		(b)(c)
104	0.112	0.005	0.103	0.003	0.290		0.123		(b)(c)
105	0.143	0.005	0.103	0.003	0.323		0.155		(b)(c)
106	0.174	0.005	0.103	0.003	0.358		0.186		(b)(c)
107	0.206	0.005	0.103	0.003	0.3906		0.217		(b)(c)
108	0.237	0.005	0.103	0.003	0.4219		0.248		(b)(c)
109	0.299	0.005	0.103	0.003	0.4844		0.311		(b)(c)
110	0.362	0.005	0.103	0.003	0.5625	0.001	0.373	0.001	(b)(c)

NOTES:

- (a) Not recommended for reciprocating applications.
- (b) Sizes noted require considerable stretch when installed in standard external grooves and may require compounds with superior elongation, or two-piece pistons.
- (c) Not for use in internal pressure face seal applications.

Table 1A - Bore and rod dimensions (inches) for O-ring gland seal design (continued)

AS568 Uniform Dash No.	Standard O-Ring Size				Bore Dia External Seals		Rod Dia Internal Seals		Notes
	ID	+ TOL	W	+ TOL	A	+ TOL	B	+ TOL	
111	0.424	0.005	0.103	0.003	0.625	0.001	0.436	0.001	(b)(c)
112	0.487	0.005	0.103	0.003	0.6875		0.498		(b)(c)
113	0.549	0.007	0.103	0.003	0.750		0.561		(b)(c)
114	0.612	0.009	0.103	0.003	0.8125		0.623		(c)
115	0.674	0.009	0.103	0.003	0.875		0.686		(c)
116	0.737	0.009	0.103	0.003	0.9375		0.748		
117	0.799	0.010	0.103	0.003	1.000		0.811		(a)
118	0.862	0.010	0.103	0.003	1.0625		0.873		(a)
119	0.924	0.010	0.103	0.003	1.125		0.936		(a)
120	0.987	0.010	0.103	0.003	1.1875		0.998		(a)
121	1.049	0.010	0.103	0.003	1.250		1.061		(a)
122	1.112	0.010	0.103	0.003	1.3125		1.123		(a)
123	1.174	0.012	0.103	0.003	1.375		1.186		(a)
124	1.237	0.012	0.103	0.003	1.4375		1.248		(a)
125	1.299	0.012	0.103	0.003	1.500		1.311		(a)
126	1.362	0.012	0.103	0.003	1.5625		1.373		(a)
127	1.424	0.012	0.103	0.003	1.625		1.436		(a)
128	1.487	0.012	0.103	0.003	1.6875		1.498		(a)
129	1.549	0.015	0.103	0.003	1.750		1.561		(a)
130	1.612	0.015	0.103	0.003	1.8125		1.623		(a)
131	1.674	0.015	0.103	0.003	1.875		1.686		(a)
132	1.737	0.015	0.103	0.003	1.9375		1.748		(a)
133	1.799	0.015	0.103	0.003	2.000		1.811		(a)
134	1.862	0.015	0.103	0.003	2.0625		1.873		(a)
135	1.925	0.017	0.103	0.003	2.125		1.936		(a)
136	1.987	0.017	0.103	0.003	2.1875		2.000		(a)
137	2.050	0.017	0.103	0.003	2.250		2.0625		(a)
138	2.112	0.017	0.103	0.003	2.3125		2.125		(a)
139	2.175	0.017	0.103	0.003	2.375		2.1875		(a)
140	2.237	0.017	0.103	0.003	2.4375	0.001	2.250	0.001	(a)

NOTES:

- (a) Not recommended for reciprocating applications.
- (b) Sizes noted require considerable stretch when installed in standard external grooves and may require compounds with superior elongation, or two-piece pistons.
- (c) Not for use in internal pressure face seal applications.

Table 1A - Bore and rod dimensions (inches) for O-ring gland seal design (continued)

AS568 Uniform Dash No.	Standard O-Ring Size				Bore Dia External Seals		Rod Dia Internal Seals		Notes
	ID	+ TOL	W	+ TOL	A	+ TOL	B	+ TOL	
141	2.300	0.020	0.103	0.003	2.500	0.001	2.3125	0.001	(a)
142	2.362	0.020	0.103	0.003	2.5625		2.375		(a)
143	2.425	0.020	0.103	0.003	2.625		2.4375		(a)
144	2.487	0.020	0.103	0.003	2.6875		2.500		(a)
145	2.550	0.020	0.103	0.003	2.750		2.5625		(a)
146	2.612	0.020	0.103	0.003	2.8125		2.625		(a)
147	2.675	0.022	0.103	0.003	2.875		2.6875		(a)
148	2.737	0.022	0.103	0.003	2.9375		2.750		(a)
149	2.800	0.022	0.103	0.003	3.000		2.8125		(a)
150	2.862	0.022	0.103	0.003	3.0625		2.875		(a)
151	2.987	0.024	0.103	0.003	3.250		3.000		(a)
152	3.237	0.024	0.103	0.003	3.500		3.250		(a)
153	3.487	0.024	0.103	0.003	3.750		3.500		(a)
154	3.737	0.028	0.103	0.003	4.000		3.750		(a)
155	3.987	0.028	0.103	0.003	4.250		4.000		(a)
156	4.237	0.030	0.103	0.003	4.500		4.250		(a)
157	4.487	0.030	0.103	0.003	4.750		4.500		(a)
158	4.737	0.030	0.103	0.003	5.000		4.750		(a)
159	4.987	0.035	0.103	0.003	5.250		5.000		(a)
160	5.237	0.035	0.103	0.003	5.500		5.250		(a)
161	5.487	0.035	0.103	0.003	5.750		5.500		(a)
162	5.737	0.035	0.103	0.003	6.000		5.750		(a)
163	5.987	0.035	0.103	0.003	6.250		6.000		(a)
164	6.237	0.040	0.103	0.003	6.500		6.250		(a)
165	6.487	0.040	0.103	0.003	6.750		6.500		(a)
166	6.737	0.040	0.103	0.003	7.000		6.750		(a)
167	6.987	0.040	0.103	0.003	7.250		7.000		(a)
168	7.237	0.045	0.103	0.003	7.500		7.250		(a)
169	7.487	0.045	0.103	0.003	7.750		7.500		(a)
170	7.737	0.045	0.103	0.003	8.000	0.001	7.750	0.001	(a)

NOTE:

(a) Not recommended for reciprocating applications.