



AEROSPACE RECOMMENDED PRACTICE

ARP7204™

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Polymeric Bearings for Linear Actuators

RATIONALE

Guidelines for the application of polymeric bearings in linear actuators do not exist in the form of an SAE document, and therefore this document is needed to establish consistency and good practice within the aerospace industry.

FOREWORD

Traditional engineering practice for controlling dynamic effects between two sliding surfaces is to use dissimilar materials to prevent or minimize excessive wear and galling. This may not be ideal in service and, in this case, an alternate method should be used, which may be the use of metallic bushings or polymeric bearings. Polymeric bearings are often selected over metallic bushings where weight reduction, contamination reduction, corrosion resistance, low friction, and cost savings are desired. In contrast, metallic bushings are often selected over polymeric bearings where unit loading exceeds the capabilities of polymeric materials.

This Aerospace Recommended Practice suggests methodologies for the practical use of polymeric bearings between sliding surfaces in aerospace hydraulic actuators, namely between pistons and cylinder bores or between rods and rod housing bores. These are designed to be a near net fit, i.e., not pressed in or shrunk on, with minimal clearances to enable suitable lateral location, with a scarf cut that permits installation into a closed groove and circumferential freedom to account for thermal expansion. These scarf-cut bearings are not intended for use in rotary motion applications as butt-cut or solid bearings are typically recommended for those circumstances.

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1. SCOPE

This SAE Aerospace Recommended Practice (ARP) provides guidelines for the application of polymeric bearings for linear actuation systems. Design considerations are included for recommended fit and function in conjunction with material selection and load-bearing capability.

2. REFERENCES

2.1 Applicable Documents

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 the 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.

2.1.1 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.

AIR1243	Anti Blow-By Design Practice for Cap Seals
ARP4386	Terminology and Definitions for Aerospace Fluid Power, Actuation and Control Technologies
AS4716	Gland Design, O-Ring and Other Seals

2.1.2 ASTM Publications

Available from ASTM International, 100 Barr Harbor Drive, P.O. Box C700, West Conshohocken, PA 19428-2959, Tel: 610-832-9585, www.astm.org.

ASTM D621	Standard Test Methods for Deformation of Plastics Under Load
ASTM D638	Standard Test Method for Tensile Properties of Plastics
ASTM D695	Standard Test Method for Compressive Properties of Rigid Plastics
ASTM D2990	Standard Test Methods for Tensile, Compressive, and Flexural Creep and Creep-Rupture of Plastics
ASTM D3702	Standard Test Method for Wear Rate and Coefficient of Friction of Materials in Self-Lubricating Rubbing Contact Using a Thrust Washer Testing Machine

2.1.3 ISO Publications

Available from International Organization for Standardization, ISO Central Secretariat, 1, ch. de la Voie-Creuse, CP 56, CH1211 Geneva 20, Switzerland, Tel: +41 22 749 01 11, www.iso.org.

ISO 527-1	Plastics - Determination of tensile properties - Part 1: General principles
ISO 604	Plastics - Determination of compressive properties
ISO 899	Plastics - Determination of creep behaviour - Part 1: Tensile creep

2.2 Definitions

The definitions for some of the words and phrases used within this document can be found below. For additional terminology and definitions, please refer to ARP4386.

BEARING: This is a sacrificial element used between two moving surfaces to react loads, provide low wear characteristics, and prevent scoring and galling of metallic surfaces by eliminating the possibility of undesirable metal-to-metal contact.

POLYMER: This is a nonmetallic material, such as a thermoplastic or thermoset.

ADJACENT SEAL: This is a seal element adjacent to, or in series with, the subject bearing that is used to prevent the escape of operating fluid.

GROOVE: This is the machined recess in which a seal or bearing is housed.

LAYDOWN: This is the radial displacement of the rod or piston within a housing in relation to the centerline, leading to a larger radial clearance on one side versus the other.

2.3 Abbreviations

CTE Coefficient of thermal expansion

CX Cross-Section

3. FUNCTIONAL INTENT

3.1 Rod Bearings

Rod bearings (see Figure 1) are designed to fit into a suitable groove in a rod gland bore with sufficient axial clearance and a sufficient gap between the ends of the scarf cut to accommodate circumferential thermal expansion. The radial dimensions are designed to provide minimal diametral clearance to provide a suitable concentric location. The selected material, radial CX, and width are designed to provide a sufficient load-bearing capability, with minimal radial deformation under load, as that can negate the intended function of ensuring the rod does not contact the mating gland bore.

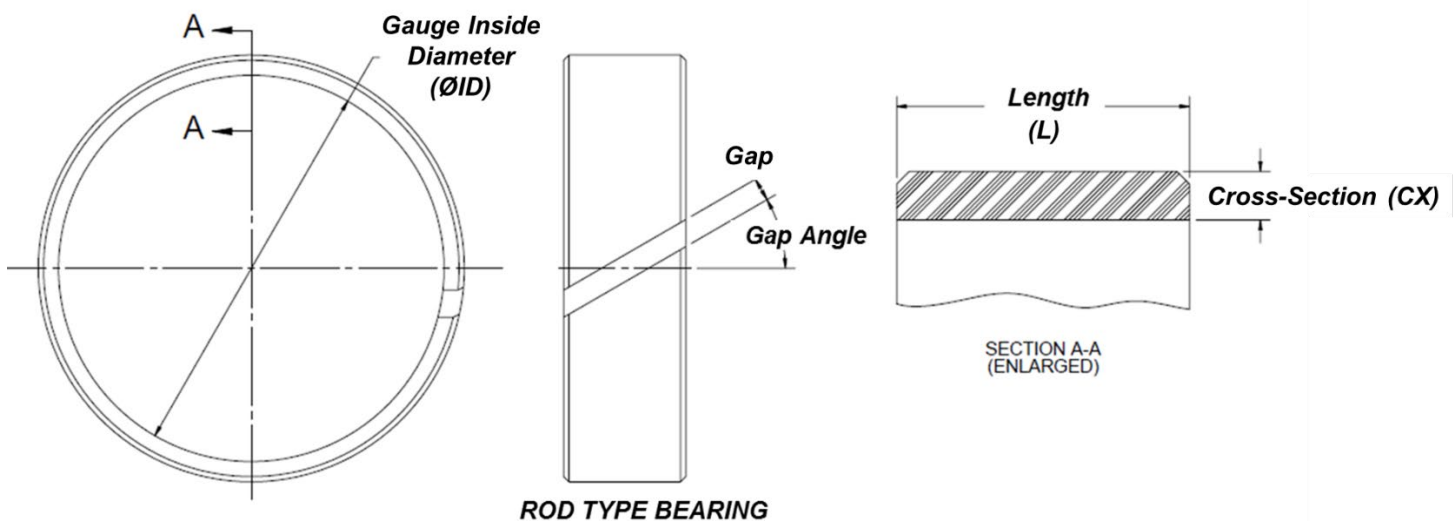


Figure 1 - Rod-type bearing

Eccentric positioning is also detrimental to the adjacent seal(s) which are generally per AS4716.

3.2 Piston Bearings

Piston bearings (see Figure 2) are designed to fit into a suitable groove on a piston outside diameter with sufficient axial clearance and a sufficient gap between the ends of the scarf cut to accommodate circumferential thermal expansion. The radial dimensions are designed to provide minimal diametral clearance to provide a suitable concentric location. The selected material, radial CX, and width are designed to provide a sufficient load-bearing capability with minimal radial deformation under load, as that can negate the intended function of ensuring the piston does not contact the mating cylinder bore.

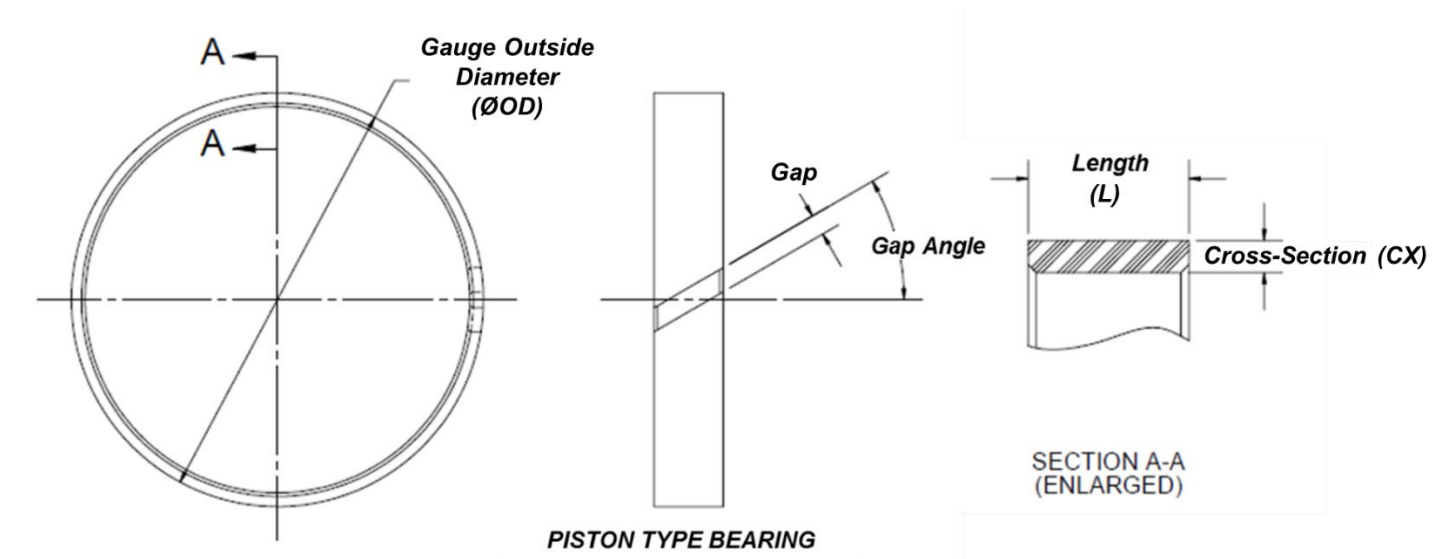


Figure 2 - Piston-type bearing

Eccentric positioning is also detrimental to the adjacent seal(s) which are generally per AS4716.

4. HARDWARE (GLAND) DESIGN

The hardware dimensions that should be considered when designing a polymeric bearing are as follows (see Figures 3 and 4 for visual representations of the typical hardware layouts):

- Dynamic Diameter (rod diameter for rod-type configurations – $\varnothing B$ in Figure 3, cylinder bore diameter for piston-type configurations – $\varnothing A$ in Figure 4)
- Groove Diameter ($\varnothing E$ in Figure 3, $\varnothing F$ in Figure 4)
- Hardware Clearance Diameter (rod bore diameter for rod-type configurations – $\varnothing H$ in Figure 3, piston diameter for piston-type configurations – $\varnothing C$ in Figure 4)
- Diametral Clearance (K; shown as radial clearance K/2 in Figures 3 and 4)
- Groove Width (G in Figures 3 and 4); +0.010/-0.000 inch (+0.25/-0.00 mm) tolerance recommended
- Groove Corner Radii (R in Figures 3 and 4; smallest possible radii recommended to reduce bearing corner interference; refer to 5.6 for alternate groove and bearing corner solutions)
- Groove Break Edges (BRK in Figures 3 and 4; 0.005 inch [0.13 mm] to 0.010 inch [0.25 mm] recommended)
- Groove Wall Angles (not pictured, refer to AS4716 for visual examples; 0 to 5 degrees recommended)

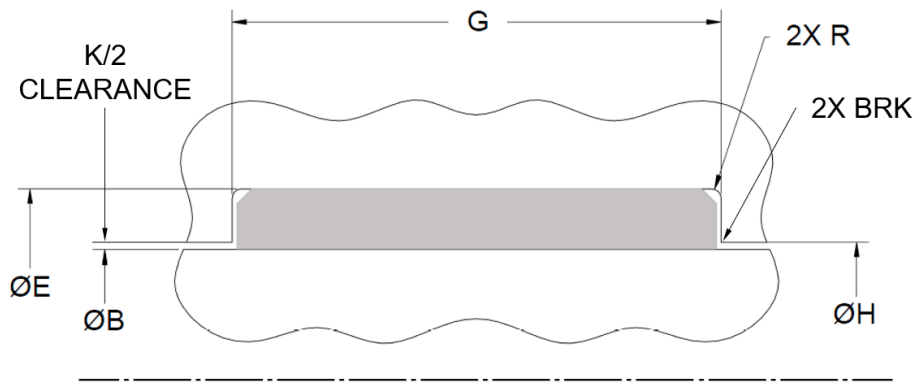


Figure 3 - Typical hardware layout for rod bearings

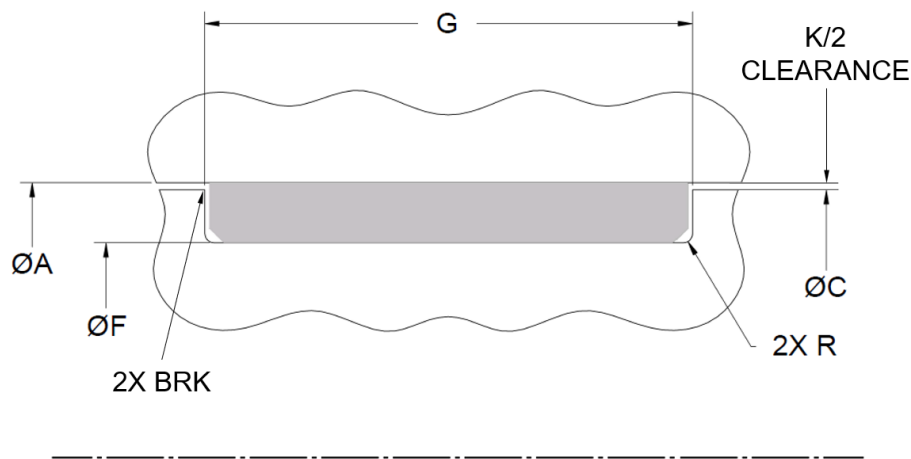


Figure 4 - Typical hardware layout for piston bearings

The majority of the hardware datum dimensions that affect a polymeric bearing are typically governed by the actuator design and adjacent seal requirements. If an actuator contains seal glands that are adjacent to the intended bearing gland, assuming the seal glands are sized per the AS4716 gland standard, then the resulting rod diameter (for a rod-type configuration) or cylinder bore diameter (for a piston-type configuration) would be defined by AS4716.

The actuator design often influences the available axial and radial envelope space for a bearing system. Just as with optimum seal size selection, there is a suitable size bearing proportion, in terms of the selected radial CX for a given diameter, and, likewise, a suitable axial width for the given radial CX and/or projected bearing area criteria. In some cases, adjacent seals are not present, which would then allow for more flexibility in the bearing hardware design.

Note that when polymeric bearings are used, the diametral hardware clearance (K) should be increased to ensure that the bearing will function under worst-case tolerance situations (i.e., no metal-to-metal contact). This increase in the working clearance should be applied through the entire axial plane of the actuator, including associated seals that are in series along this same working plane, to avoid the running hardware surface from contacting one or more metallic land areas before the bearing surface. Additional safety factors are often added to the clearance as well to account for compression of the bearing material, particularly under side loading.

Figure 5 depicts the correct application of hardware clearances when a bearing is in series with a seal, where the clearance is a continuous diameter across both glands.

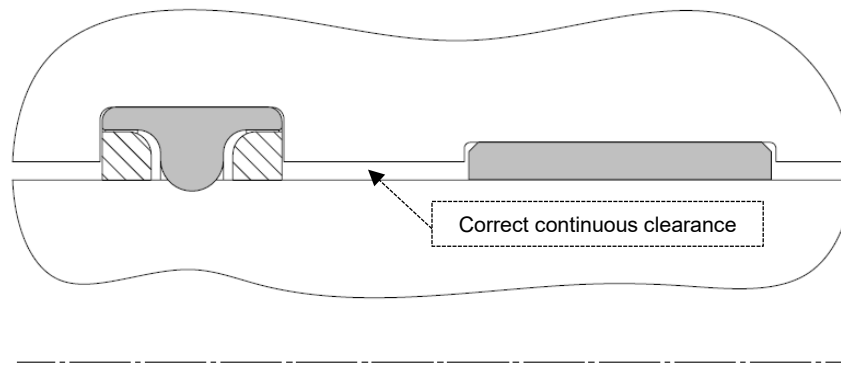


Figure 5 - Correct application of hardware clearances for bearings when adjacent seals are present

Figure 6 shows the incorrect application of those clearances by adding a “stepped” area near the seal groove. The “stepped” area is often added with the intent to reduce the extrusion gap for the seal; however, that can then render the bearing ineffective by potentially allowing metal-to-metal contact when side loading or off-center conditions are present.

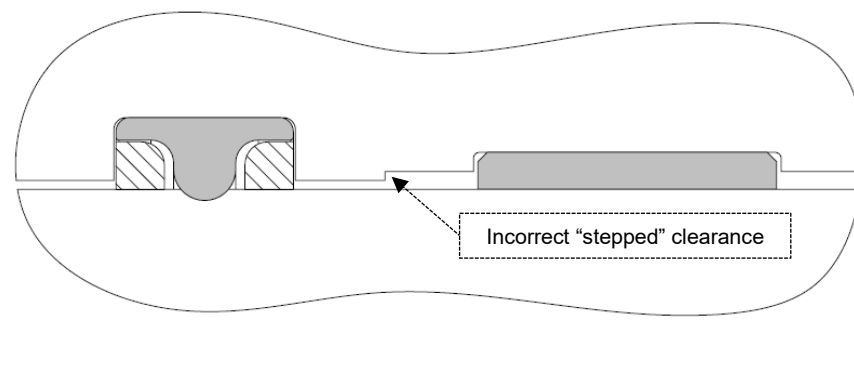


Figure 6 - Incorrect application of hardware clearances for bearings when adjacent seals are present

The equations for calculating the groove diameter ($\varnothing E$ for rod type or $\varnothing F$ for piston type), hardware clearance diameter ($\varnothing H$ for rod type or $\varnothing C$ for piston type), and groove width (G), based on the given application datum diameter (rod diameter $\varnothing B$ or cylinder bore diameter $\varnothing A$) and desired radial CX and axial length (L) of the polymeric bearing, can be found below. These equations apply when the bearing radial CX and axial length dimensions are predefined. For the equations needed to size a bearing based on predefined groove dimensions, see Section 5.

4.1 Rod-Type Configuration Hardware Calculations

For a rod-type configuration:

$$\text{Min } \varnothing E = (\text{Max } \varnothing B + 0.001 \text{ inch } (0.03 \text{ mm})) + (2 \times \text{Max } CX) \quad (\text{Eq. 1})$$

where:

$\varnothing E$ = rod-type configuration groove diameter

$\varnothing B$ = rod diameter

CX = bearing radial cross-section

$\varnothing B$ and CX dimensions and tolerances are known/given

Tolerance for Min $\varnothing E$ = reverse of tolerance for Max $\varnothing B$, (i.e., if $\varnothing B \text{ Tol} = +0.000/-0.002 \text{ inch}$, $\varnothing E \text{ Tol} = +0.002/-0.000 \text{ inch}$)



$$\text{Min } \varnothing H = \text{Max } \varnothing E - (2 \times \text{Min } CX) + CSF \quad (\text{Eq. 2})$$

where:

$\varnothing H$ = rod bore (clearance) diameter

Tolerance for Min $\varnothing H$ = reverse of tolerance for Max $\varnothing B$, (i.e., if $\varnothing B$ Tol = +0.000/-0.002 inch, $\varnothing H$ Tol = +0.002/-0.000 inch)

CSF = clearance safety factor

NOTE: The clearance safety factor (CSF) is added to (for rod-type configurations) or subtracted from (for piston-type configurations) the hardware clearance diameter to allow for the proper functionality of the bearing in the event of material compression or deformation, particularly under side loading. This value can be determined by the bearing manufacturer based on specific materials and side loads applied, or in the absence of manufacturer direction, a minimum CSF of 0.010 inch (0.25 mm) can be applied.

$$\text{Min } G = \text{Max } L + 0.010 \text{ inch (0.25 mm)} \quad (\text{Eq. 3})$$

where:

G = groove width

L = bearing axial length

L dimension and tolerance are known/given

Tolerance for Min G = +0.010/-0.000 inch (+0.25/-0.00 mm)

NOTE: Equation 3 includes the addition of 0.010 inch (0.25 mm), which is a built-in clearance to allow for axial space next to the bearing once installed. Most manufacturer catalogs include this typical axial clearance value.

Example calculations to determine the groove diameter, rod bore diameter, and groove width (for a rod-type configuration) are as follows:

$$\varnothing B = 1.500 \text{ inches} + 0.000/-0.002 \text{ inch (38.10 mm} + 0.00/-0.05 \text{ mm)}$$

$$CX = 0.031 \text{ inch} \pm 0.001 \text{ inch (0.79 mm} \pm 0.03 \text{ mm)}$$

$$L = 0.313 \text{ inch} + 0.000/-0.010 \text{ inch (7.95 mm} + 0.00/-0.25 \text{ mm)}$$

$$CSF = 0.010 \text{ inch (0.25 mm)}$$

Therefore,

$$\text{Min } \varnothing E = (1.500 \text{ inches (38.10 mm)} + 0.001 \text{ inch (0.03 mm)}) + (2 \times 0.032 \text{ inch (0.82 mm)}) = 1.565 \text{ inches (39.75 mm)}$$

$$\varnothing E = 1.565 \text{ inches} + 0.002/-0.000 \text{ inch (39.75 mm} + 0.05/-0.00 \text{ mm)}$$

$$\text{Min } \varnothing H = 1.567 \text{ inches (39.80 mm)} - (2 \times 0.030 \text{ inch (0.76 mm)}) + 0.010 \text{ inch (0.25 mm)} = 1.517 \text{ inches (38.53 mm)}$$

$$\varnothing H = 1.517 \text{ inches} + 0.002/-0.000 \text{ inch (38.53 mm} + 0.05/-0.00 \text{ mm)}$$

$$\text{Min } G = 0.313 \text{ inch (7.95 mm)} + 0.010 \text{ inch (0.25 mm)} = 0.323 \text{ inch (8.20 mm)}$$

$$G = 0.323 \text{ inch} + 0.010/-0.000 \text{ inch (8.20 mm} + 0.25/-0.00 \text{ mm)}$$