O-ring Seal Design Best Practices

12-15-12
Rev. 1

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1.0 O-RING SEALS – THEORY AND DESIGN PRACTICES

Theory:

An o-ring seal consists of an o-ring and a properly designed gland which applies a predictable deformation to the o-ring. The gland is basically a groove dimensioned to a certain height "H" and width "W" (Figure 1) to allow a fixed compression of the o-ring when the gland flanges make metal to metal contact. It is also oversized volumetrically such to allow accommodation of the o-ring as it flows under compression. Unlike gaskets which seal just by the resiliency of the material under mechanical compression of the joint, an o-ring can provide a seal both through the resiliency of the pre-compressed material and the pressure activation of the seal. The pre-compression of the o-ring applies a calculated mechanical contact stress or pressure at the o-ring contacting surfaces in the gland. As the o-ring seal is pressurized or "activated" the pressure on the o-ring further increases the contact stress on the o-ring contacting surfaces of the gland as the o-ring moves or "flows" toward the low pressure side. This means the pressure of the contained fluid transfers through the essentially incompressible o-ring material, and the contact stress rises with increasing pressure. As long as the pressure of the fluid does not exceed the contact stress of the o-ring, leakage should not occur.

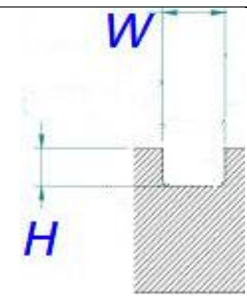


Figure 1 - O-Ring Groove Dimensions

At zero gauge pressure, only the pre-compressed resiliency of the o-ring provides the seal (see Figure 2). If the system pressure is in the range of 0 - 100 or even to 400 pounds per square inch (psi) it can be considered as low pressure (in this paper 400 psi or less is considered low pressure), and the seal is maintained predominantly by the pre-compression or "squeeze" on the o-ring and its resulting contact stress. As the pressure increases, the o-ring is forced to the low pressure side of the gland. This provides an additional increase in contact stress as the o-ring deforms to a "D" shape (see Figure 3) and the contact area of sealing under pressure increases to almost twice the original zero-pressure area. For this reason, an o-ring can easily seal a high pressure as long as it does not mechanically fail.

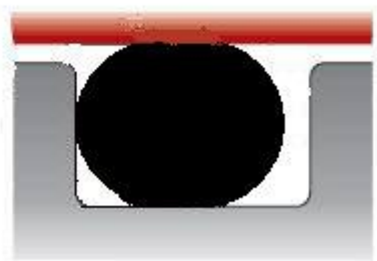


Figure 2 - O-ring un-pressurized

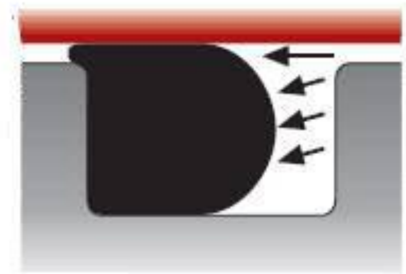


Figure 3 - O-ring with high pressure

<u>Best Design Practices:</u>

a. The flexible nature of o-ring materials accommodates imperfections and/or waviness in the gland parts. But it is still important to maintain a good surface finish of those mating parts. The following best practices are suggested: 32 micro-inch finish on the contact surfaces (top of gland and bottom or groove); 63 micro-inch finish on the sides of the groove; machined radii in bottom of groove of 1/32" (reference 2); holding waviness of groove bottom to less than 2% of o-ring thickness per 12" length of groove.

Figure 4 shows a poor surface finish and affect the tool mark direction. Such a finish can cause leaks.

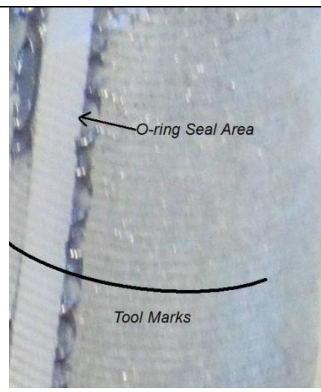


Figure 4 - Surface Finish Can Cause Leaks

b. As for o-ring compression or "squeeze" it is a result of three factors: the force to compress the o-ring, durometer, and cross section thickness. A 15-20% compression for dynamic (moving) applications (to mitigate wear) and 35-40% for static applications (reference 3) is generally suggested. Whereas, reference 2 (Parker) recommends 16% for dynamic applications and 30% for static. However, analysis and testing of the application will determine the ultimate compression. Compression % is defined as the deflection of the seal divided by the cross-section thickness (cord diameter) and the results times 100.

c. The rigidity of the gland closure and closure bolting spacing must be adequate to compress the o-ring without deflecting. Any deflection will reduce the design compression of the seal.

d. The area of the cross-section of the gland should be in the range of 15 to 40% greater than the area of the cross-section of the o-ring. A 75% fill is suggested which leaves 25% empty space in the gland groove (reference 2). However, it is very important that the installed o-ring contact the low pressure side of the gland groove (see Figure 1 in Appendix) such that the o-ring only has to move very little when pressurized or "activated".

e. For o-ring gland grooves that are non-circular, the groove turn radii in rectangular and square layout (i.e. the corners), must be large enough so the o-ring will not kink and such that the o-ring will fully contact the low pressure side of the groove (Figure 5 shows an example of too sharp of a corner in the o-ring groove). Otherwise, the small radius turn may impede any "activation" of the seal. For example, the radius in the corners of an o-ring groove layout is suggested to be at least 2 inches for a ¼" (0.275") nominal diameter o-ring stock, or 7 to 8 times the o-ring diameter. Fabricate the o-ring to snugly fit the low pressure side of the gland groove all the way around including the corner radii.

Figure 5 - Too Sharp of Corner

f. It is important not to stretch the o-ring since stretch affects seal compression by reducing cross section, which reduces the sealing potential of the o-ring. A stretch greater than 5% on the o-ring I.D. (equivalent ID in non-circular case) is not recommended because it can lead to a loss of seal compression (reference 3).

g. When sizing an o-ring, choose the largest cross-section thickness as practical. The larger the cross-section, the more effective the sealing and longer the life of the seal. However, with a dynamic application in which friction is a factor, a compromise will be required.

h. A 70 durometer (shore A) hardness should be used in the design whenever possible since it usually has the best combination of properties for most applications. It provides good conformability versus a mid-range contact stress capability (see Graph 1). It is also considered the standard o-ring hardness and is readily available from suppliers.

i. A lubricant compatible with the o-ring material should be applied to the o-ring as it is installed to decrease friction and assist in the "activation" of the seal under pressure.

j. O-ring re-use: reusing an o-ring in an assembly after some time in service is generally not recommended. Is the o-ring deformed, cracked, harder than when new, discolored, or less than clean? When in doubt, change out.

2.0 ANALYSIS OF O-RING SEAL DESIGNS

The maximum sealing capability of high pressure o-ring seal designs is dependent on seal "activation" as discussed earlier and is not solely dependent on the initial contact pressure as determined by the compression of the seal in its housing (groove). However for low pressure designs, an analysis of the contact stress makes it possible to better predict the success of the seal while assuming no "activation" of seal. Two methods are presented below:

Parker Method:
As an example referring to the "Parker O-ring Handbook, Figures 2-8" (reference 2) , below is an analysis of a ¼" o-ring design, Shore A durometer, with 20% compression:

For a 20% compression on a ¼ inch nominal o-ring, , the compression load per linear inch of the seal is at around 35 pounds from the Parker Figures. Referring to the paper "O-rings for Low Pressure Service", (reference 1), the contact area "b" per linear inch of seal is estimated by b=2.4x, where x is the deflection of the o-ring cross-section. Therefore, the contact area per inch is (2.4) (.20)(.275) = .132 square inch. The contact stress "S_{max}" per linear inch just from pre-compressed resiliency of the seal is 35 pounds divided by .132 square inches yielding 265 psi. This means, in theory with all things being perfect, the seal should not leak until the pressure exceeds 265 psig, as a minimum.

Lindly Method:
As an example using Lindly's analysis as referred to in reference 4, below is an analysis of a ¼" o-ring design, Shore A durometer, with 20% compression:

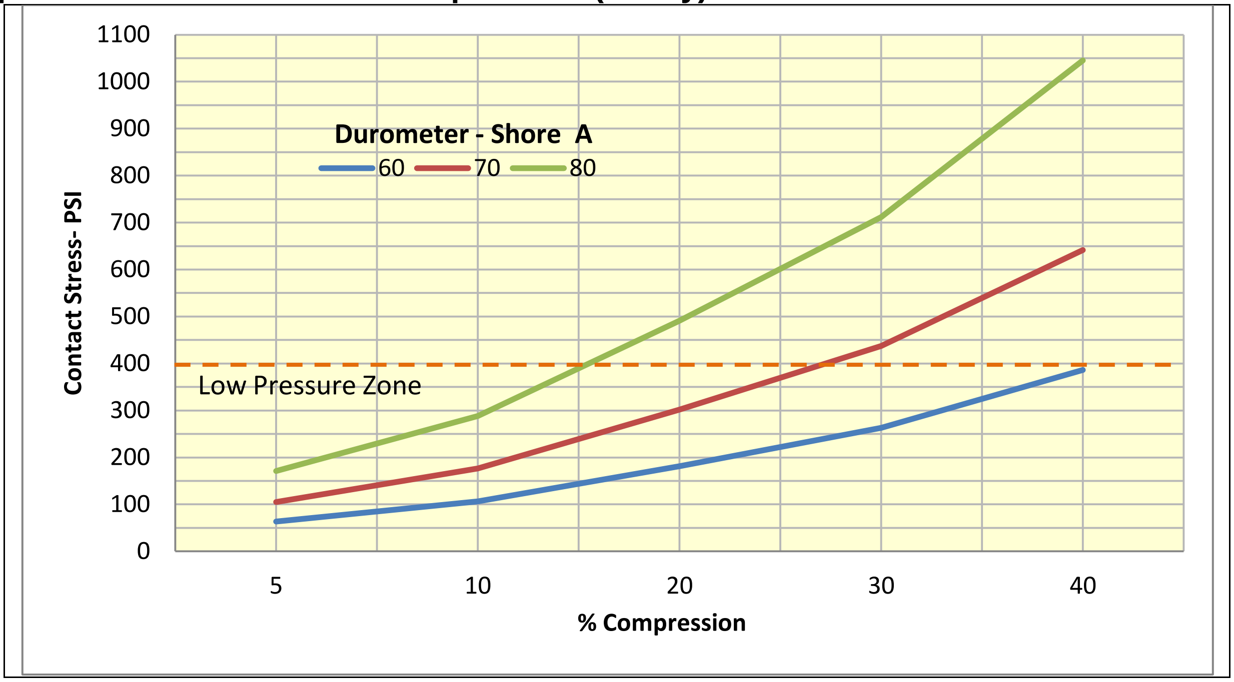
Lindly derived the equation below for predicting the contact stress "S_{max}" with respect to the modulus of elasticity "E" (E can be derived from Shore A durometer):

S_{max}=E $(0.849(1.25\delta^{1.5} + 50\delta^{6}))^{0.5}$
where: S_{max} = contact stress psi, δ = fractional compression, (20% = 0.2), E= modulus of elasticity (which can be found in reference 1, of E versus Shore A durometer; 60 = 630 psi, 70= 1,040 psi, 80 = 1,705 psi)

S_{max} is plotted in Graph 1 for Shore A durometer 60, 70, and 80. The Lindly Method gives a higher contact stress than the Parker, however, due to the simplicity of use, the Lindly Method is the preferred method.

Calculating contact stress is only a starting point in the seal design for determining the minimum required compression, however, with variables in the "less than perfect" sealing system, a margin above that is prudent, so it is suggested to follow the best practices in Section 1.0 in order to achieve a successful seal.

Graph 1 - Contact Stress vs. Compression (Lindly)



3.0 APPENDIX

References:

1. Hertz,D.L.,1979, "O-Rings for Low Pressure Service", Machine Design, 4/12/79, pp.94-98 (note, paper applies mainly to dynamic applications)
2. Parker Hannifin Corporation, O-Ring Handbook, Catalog ORD 5700A/US
3. Apple Rubber Products Inc.,www.applerubber.com
4. Green, Itzhak and English, Capel, “Stresses and Deformation of Compressed Elastomeric O-Ring Seals”