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MACHINE DESIGN

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O-RINGS FOR LOW-PRESSURE SERVICE

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O-RINGS FOR LOW-PRESSURE SERVICE

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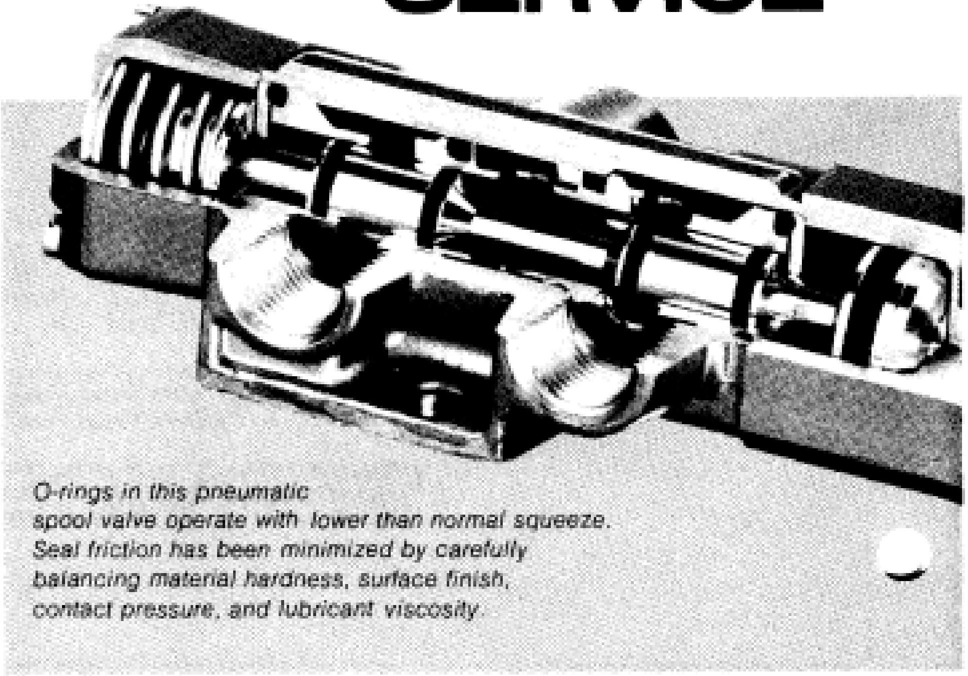


O-rings in this pneumatic spool valve operate with lower than normal squeeze. Seal friction has been minimized by carefully balancing material hardness, surface finish, contact pressure, and lubricant viscosity.

O-RINGS normally operate with about 15% squeeze to ensure a tight seal. But at system pressures below 400 psi, this amount of squeeze can cause high friction and excessively high actuating forces.

Reducing the amount of squeeze lowers friction to acceptable levels; however, lower squeeze also means lower sealing pressure and greater potential for leakage. This problem is aggravated by the stress relaxation characteristics of the seal material. Thus, an O-ring that seals well initially may lose resilience with time and fail suddenly.

Designing O-ring seals for low pressures, therefore, is not simply a matter of reducing the amount of squeeze: it involves a delicate balancing of material hardness, dimensional tolerances, stress relaxation, and friction characteristics.

Table 1—Recommended Squeeze for O-rings

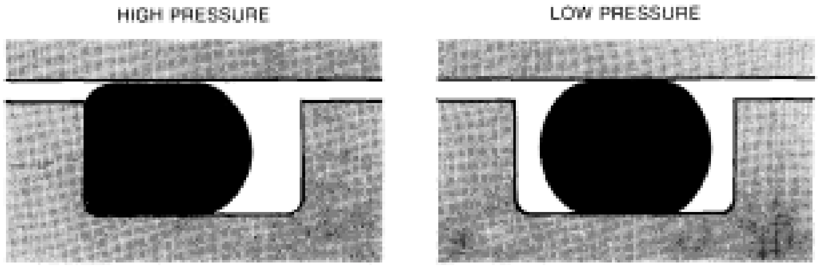
Seal Thickness (in.)	Squeeze (%)	Max % Squeeze Taken by Thickness Tolerance
0.070 ± 0.003	14.9-24.7	8.6
0.103 ± 0.003	8.1-15.0	5.8
0.139 ± 0.004	8.5-15.0	5.8
0.210 ± 0.005	8.3-13.5	4.8
0.275 ± 0.006	10.2-15.1	4.4

Material Hardness

The initial phase of designing a low-pressure O-ring seal is the same as that for a conventional O-ring. Size and fluid compatibility requirements are evaluated and O-ring dimensions selected from a catalog. The catalogs usually list a recommended range of squeeze values, as shown in Table 1.

Squeeze is defined as the ratio

How Trouble Starts



High system pressures deform an O-ring into a D-shape, increasing contact area and sealing force. However, pressures below 400 psi are not strong enough to deflect the seal, and sealing force comes only from the compressive stress developed in the O-ring.

Most O-rings operate with enough "squeeze" to provide a reliable seal under almost any conditions. But low-pressure systems generate less squeeze, increasing the potential for leakage. Only a careful balancing of O-ring material properties ensures leak-free operation at low pressures.

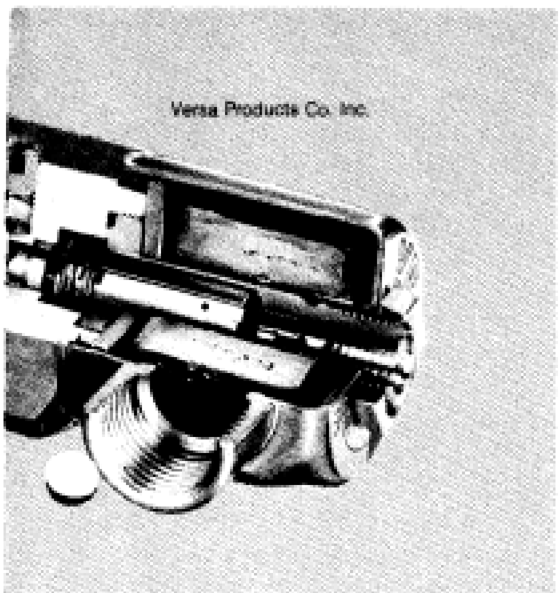


Table 2—Young's Modulus for O-ring Materials

Shore A Hardness (±2)	Young's Modulus (psi)
40	213
45	256
50	310
55	460
60	630
65	830
70	1,040
75	1340

of seal deflection to seal thickness, x/d. Generally, the seal is designed to operate at the high end of the squeeze range to ensure a tight seal. But at low system pressures, squeeze must be specified at the low end of the range.

The squeeze values listed in Table 1 are based on nominal seal thickness. One problem with specifying squeeze at the low end of the range is that dimensional tolerances can reduce the amount of squeeze actually placed on the O-ring. For instance, tolerance on the 0.070-in. thick O-ring is ±0.003 in. In the worst case (0.067-in. thickness), this tolerance can account for 8.6% of the squeeze allowance, leaving only 6.3% to be supplied by the fit between parts. In other words, an undersize O-ring has less material to compress and cannot be squeezed as tightly against the sealing surfaces. This problem can be minimized by specifying O-rings with one-half the normal dimensional tolerances. Such seals are available from most manufacturers at a premium price.

The next step in the design procedure is to calculate the compression force developed in the O-ring. This force is directly related to the sealing ability of the ring and is calculated from

$$F = \pi d D_m E\left[1.25\left(\frac{x}{d}\right)^{1.5} + 50\left(\frac{x}{d}\right)^{6}\right]$$

To use this equation, Young's Modulus must be determined first. This value depends on material hardness, and typical values are listed in Table 2. For most applications, a Shore A hardness of 70 is sufficient; therefore, the initial calculation of F is based on this hardness.

From the specified squeeze and seal thickness, contact area can be calculated from

$$b = 2.4x$$

Then, the peak contact stress can be found from

$$f' = \frac{4F}{\pi^2 b D_m}$$

If f' is greater than the system pressure, the O-ring will seal the joint. If f' is less than system pressure, the ring will leak and a material with a higher Young's Modulus must be specified, thereby increasing compressive force and contact stress.

Seal Friction

In low-pressure systems, seal friction can raise the required actuating pressure to many times that available in the system. Therefore, seal friction must be minimized for the system to operate properly. Generally, seal friction force should be maintained below 20 lb to keep actuating force within reasonable limits.

The friction force for an O-ring seal can be estimated from

$$F_f = \mu F$$

where coefficient of friction, μ, can change from 0.001 to over 10, depending on the operating conditions.

When more than one O-ring is used in the system, the friction forces from all the seals must be combined to determine the total friction force. If the calculated force is greater than

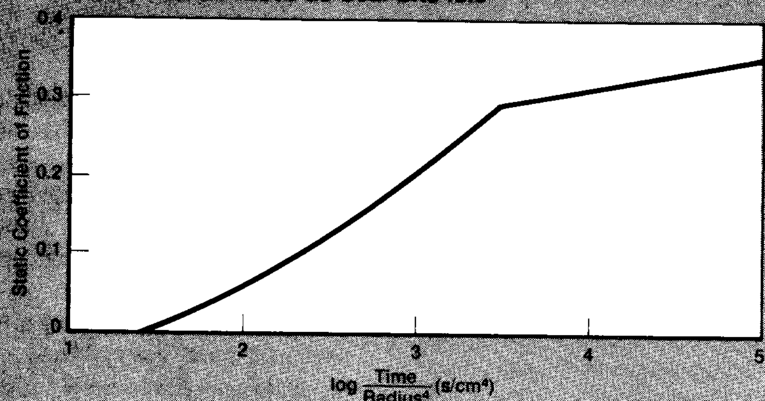
20 lb, a softer seal material should be used; this lowers Young's Modulus and compressive force. However, the change to a softer seal material must be made with care because a lower compressive force also means a lower contact stress. Thus, the change could lower peak contact stress below system pressure, resulting in a leaky seal.

If a softer material lowers contact stress too much, friction force must be lowered by reducing the coefficient of friction. This factor is a complex function of lubricant film thickness, time, contact stress, sliding speed, and surface finish.

Tests have shown that the longer a lubricated seal sits idle, the higher its static, or breakaway, coefficient of friction. Eventually, the friction coefficient reaches a maximum value almost as high as that for an unlubricated seal.

This increase with time is caused by the atomic interaction between the O-ring and its sealing surface, which causes the two surfaces to adhere tightly. The adhesive force can be quite high and eventually squeezes most of the lubricant from under the contact area. On start-up, the adhered O-ring peels away in progressive waves that break away and reform

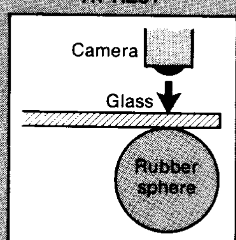
Static Friction Rises as Seal Sits Idle

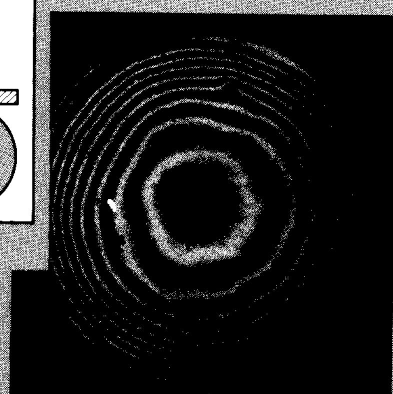


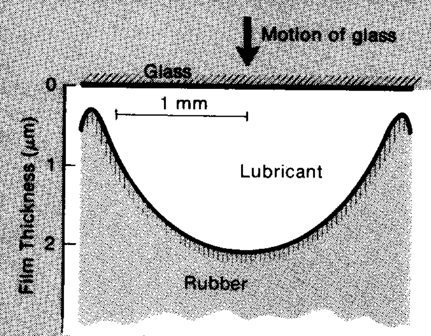
Tests of lubricated rubber sliding against a steel surface show how static friction increases the longer an O-ring sits idle. The increase is caused by the interatomic attraction between the rubber and steel, which slowly squeezes lubricant from under the contact surface.

Seal Adhesion Increases Friction

AT REST

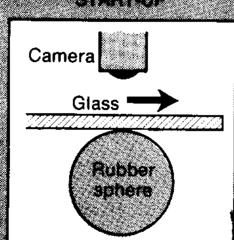


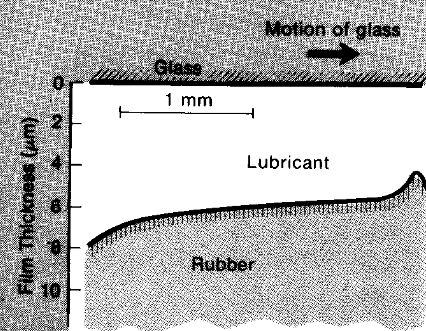




Optical interferograms of a glass plate pressing against a rubber sphere show how elastomers adhere to their sealing surfaces. (In the photos, dark areas are adhered rubber and light areas are trapped lubricant.) When motion begins, the adhered areas move forward in waves prior to breaking away. This phenomenon can raise the required actuation pressure above that available in the system.

START-UP





on the moving surface, This action shears what little lubricant is present and traps it in the rubber folds.

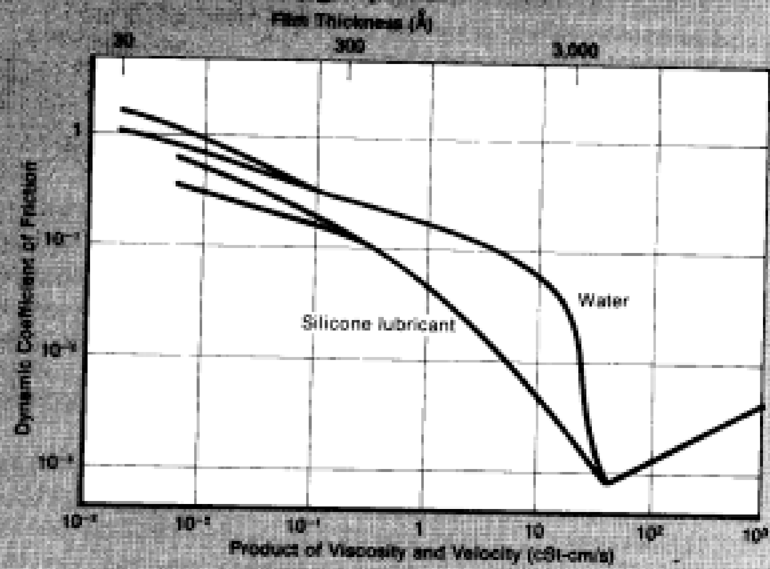
Seal adhesion can be minimized by optimizing surface finish and lubricant viscosity. Experience has shown that the optimum surface finish is 0.4μm. This finish leaves tiny pockets that collect lubricant, making it available at startup. Too smooth a finish leaves no pockets for the lubricant, while too rough a finish causes high wear.

A reciprocating seal should be lubricated with high viscosity lubricants because they produce a strong hydrodynamic film. This film resists displacement by the adhesive forces when the seal is stationary. A rotary seal, on the other hand, can be lubricated with low-viscosity lubricants because rotary motion aids development of a hydrodynamic film.

Thickness of the hydrodynamic film between asperities on the O-ring and sealing surface has been calculated as 6 x 10 in. Shear of this film is the prime cause of dynamic or running friction. In general, the dynamic coefficient of friction is a function of lubricant viscosity and sliding velocity. The coefficient generally starts high, decreases to a minimum value, then increases again. Thus, running friction can be minimized by optimizing viscosity and velocity.

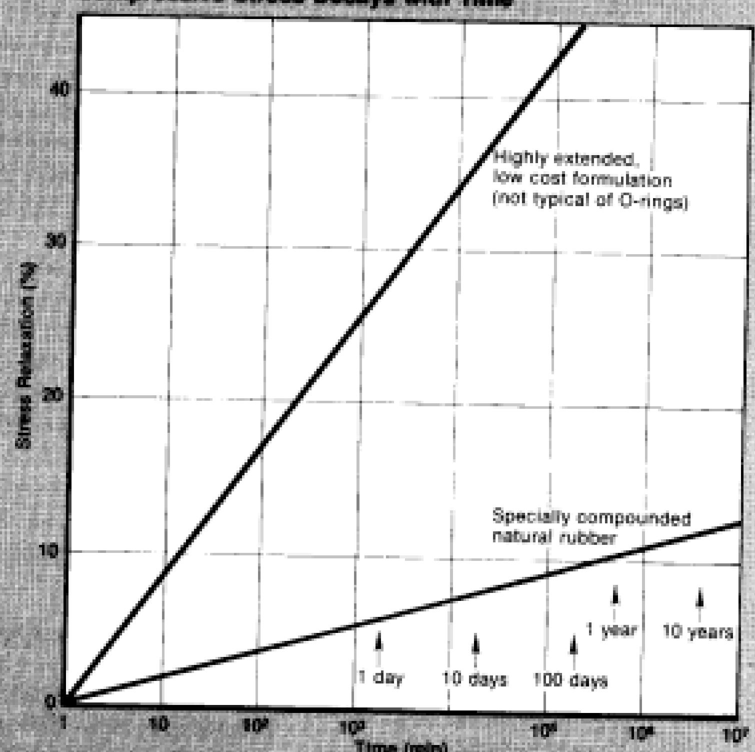
Several tests have been run to determine the effect of material-formula modifications on seal friction. The addition of materials such as graphite, molybdenum disulphide, and PTFE sometimes reduce friction, but the reduction is more likely a result of lowering Young's Modulus than a lubricating effect. Also, the incorporation in the elastomer of high-molecular-weight waxes and oils that migrate to the surface has proved unsuccessful in

Friction Drops at High Speeds



Dynamic (or running) friction is caused by viscous shear of the lubricant film. Tests of a steel ball sliding on a rubber surface show that at high speeds, friction is proportional to the square root of the viscosity-velocity product. This condition corresponds to hydrodynamic lubrication. Friction rises at lower speeds as the contact approaches boundary conditions.

Compressive Stress Decays with Time



Relaxometer tests provide data on the stress relaxation rates of O-ring materials. The relaxation rate for natural rubber generally is lower than that for rubber with additives. Knowing the relaxation rate allows prediction of a seal's useful life.

lowering friction.

Surface treatments have been more successful. Halogenation with chlorine or bromine reduces friction by lowering the surface free energy (and, therefore, attraction force) and by creating lubricant pockets. Fluorination, although far less common, has similar effects.

Surface treatments of PTFE-resin binder coatings and tumble treatment in molybdenum disulphide or graphite have been used along with silicone oil dips with limited success. Also, polymerization of monomers on the O-ring surface with plasma techniques offers some improvement; however, the techniques are costly and slow. Finally, the grafting onto the seal surface of high-molecular-weight oils having reactive end groups shows promise for the future.

Stress Relaxation

The useful sealing life of an O-ring depends on two viscoelastic material properties: compression set, the residual deformation of a material after the load is removed; and stress relaxation, the decrease in stress after a given time at a constant strain. These properties reduce the resiliency of the seal material and must be taken into account when specifying material hardness.

When a seal is under constant compression, the initial stress decays at a rate proportional to the logarithm of time. The stress relaxation rate varies with material composition, temperature, and fluid reactions. Typical values range from 0.5% to 10% per time decade. (The time from 1 to 10 min is designated as one decade, as is the much longer time from 1 to 10 weeks.)

The result of stress relaxation is that peak compressive stress eventually drops below system pressure, and the seal leaks. Thus, stress relaxation effects must be factored into the determination of material hardness and compressive stress.

Stress relaxation rates are available from O-ring manufacturers; however, the ratings may be for a temperature or fluid condition different from that required. If the correct data are not available, the stress relaxation rate can be determined from a simple relaxometer test, such as that described in ASTM D-1395, or with a Lucas relaxometer.

Once the stress relaxation rate is known, the time for peak contact stress to equal system pressure can be calculated easily. In general, if the calculated time period produces a seal life lower than 20 x 10^6 cycles, then a harder seal material (higher Young's Modulus and higher compressive stress) must be used.

Nomenclature

b = Seal contact area, in.^{2}
D_i = Seal inside diam, in.
D_m = Seal mean diam, in.
D = Seal outside diam, in.
d = Seal thickness, in.
E = Young's Modulus, psi
F = Compressive load, lb
F_f = Friction force, lb
f = Peak contact stress, psi
x = Seal deflection, in.
μ= Coefficient of friction

Temperature Effects

The effects of operating temperature are more pronounced for low-squeeze O-rings because the seal has a lower tolerance for change. The volumetric expansion rate for rubber is about 15 times higher than that for steel. Therefore, at high temperatures, insufficient groove volume can produce expansion forces that extrude the seal into the clearances. This problem can be minimized by increasing groove dimensions to provide sufficient room for expansion.

Lowering operating temperature results in a continuous decrease in the physical volume of the seal. Eventually, the seal reaches its so-called glass-transition temperature, where it seals only along two thin lines. Further reduction of temperature shrinks the seal even more, resulting in leakage.

The glass-transition temperature corresponds to 100% compression set. At this temperature, the seal can shatter like glass if subjected to a shock or impact load. Values of glass transition temperature for O-ring materials are listed in Table 3. To avoid low temperature problems, O-rings should operate at temperatures 10° to 15°F higher than those listed in the table.

Table 3-Glass-Transition Temperatures for O-ring Materials

Material	Transition Temperature (°F)
Nitrile (NBR) 34% ACN	-21
38% ACN	-13
Fluoro Rubber (FKM)	-4
Silicone (VMQ)	-85
Chloroprene (CR)	-40
Ethylene Propylene (EPDM)	-85

References

1. C.J. Derham, "Elastomeric Sealing," *Engineering,* May 1977.
2. P. B. Lindley, "Engineering Design with Natural Rubber," Malaysian Rubber Producers' Research Association, London, 1974.
3. P. B. Lindley, "Compression Characteristics of Laterally Unrestrained Rubber O-Rings," *Journal of the Institution, of the Rubber Industry,* July/August 1967.
4. A. D. Roberts, "Optical Rubber," *Rubber Deuelopments, Vol. 29,* No. 1, *1976.*
5. A. D. Roberts, "Looking at Rubber Friction," *Rubber Deuelopments, Vol. 29,* No. *4, 1976.*