Frictional performance of an O-ring type seal at the commencement of linear motion

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Results of investigation into the relationship between friction in the O-ring type seal and gauge pressure at the commencement of linear motion of a shaft are presented and discussed. A number of different O-ring materials were studied and the lowest friction under dry conditions was found to be produced by a PTFE-encapsulated silicone seal. The effect of a number of lubricating fluids on friction in the seal was also studied and the results obtained are included.

Keywords: O-ring seal; dry friction; lubricated friction; friction coefficient

1. Introduction

For applications such as rotary process pumps in chemical and petrochemical industries or compressors in the offshore oil exploration industry, a seal is a vital element for reliable and safe operation.

A seal is basically a device for closing a gap or making a joint fluid-tight. When sealing takes place between surfaces that have relative movement, e.g. rotation of a shaft relative to a housing, the seal is of a dynamic type. The majority of dynamic seal types in general use can be classified as contact seals. The other category of dynamic seals is usually referred to as clearance seals because they operate with positive clearance. They are usually called mechanical face seals or rotary shaft seals. Successful operation depends on achieving the right conditions at the interface, i.e. the faces themselves must be separated by a thin film of the fluid contained.

The rotary shaft seal is mainly used in pumps and compressors. A reliable operation of the seal is especially important for compressors used to circulate high-pressure natural gas extracted from the North Sea [1–3]. The principle of operation of the so-called dry gas seal is the balancing of aerostatic and aerodynamic forces to provide a stable, minimal running clearance. The Department of Mechanical Engineering, Brunel University has been involved in the development of a new type of dry gas seal [4]. A schematic diagram of the dry gas seal is shown in fig. 1. It consists of two faces. The dynamic face is held rigidly to the drive shaft by the dynamic carrier and locking sleeve. The static face is free to move axially with retaining three springs and O-ring arrangement. The axial freedom is required to allow for the gap between seal faces to be established under given operating conditions.

The O-ring seal prevents pressurised gas escaping behind the static face. The retaining springs force the static and dynamic faces to be in contact until the correct operating pressure across the face is reached and separation of the faces occurs. Typically, on start up of a compressor the seal faces are in contact with one another due to the action of the retaining springs. Any clearance between the two faces at this point of compressor operation would lead to an unacceptable leakage flow. However, once the compressor is up to speed the faces separate leaving a very small gap. The size of the gap is very critical as it controls the leakage.

The main problem associated with designing a dry gas seal is calculating the forces on the seal faces during operation. The forces have to be known so that the seal faces can be designed to open at the correct operation conditions and the clearance between the faces maintained at the required value. Under normal operating conditions there is a force balance between the separating forces and the closing forces. Friction within the O-ring type seal, which is incorporated into dry gas seal design, plays an important role in the balance of forces, especially during the start up when the gap between faces is being created. It is essential that the separating forces can overcome not only the closing forces but also the frictional force in the

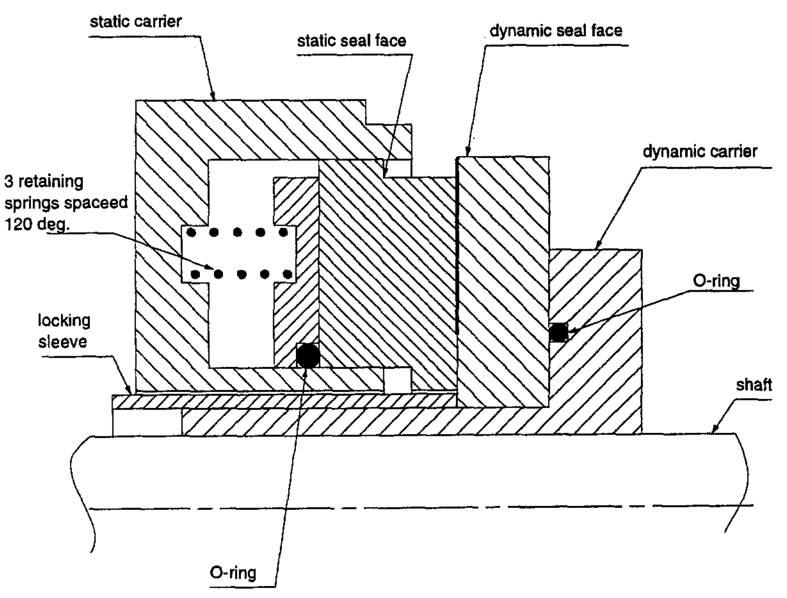

Fig. 1. Schematic representation of the dry gas seal.

O-ring seal located behind the static face. Until this force is overcome the seal faces will not come apart.

The present paper presents results of experimental studies [5] where the primary objective was to measure the frictional force in an O-ring seal at the commencement of linear motion as a function of pressure using a configuration close to that of a dry gas seal. There have been a number of studies, mainly of theoretical nature, on the O-ring type seal. Green [6] used the finite element method to analyse the behaviour of an elastomeric O-ring seal in compression. The state of deformation of an unpressurised elastomeric O-ring seal inserted into a rectangular groove was studied by Dragoni [7,8]. Stress fields in a compressed unconstrained elastomeric O-ring seal were investigated by George [9]. He compared his computer model predictions with experimental results. All these studies have concentrated on the static performance of an O-ring seal. Narumiga [10] studied an elliptically deformed sealing ring for a shaft seal. Although he considered dynamic conditions, the emphasis was on the state of deformation within the O-ring seal.

It can be safely concluded that the public domain information about the level of friction in an O-ring seal at the moment when the shaft commences its linear motion is rather scarce. As the object of our study was a real, full-scale O-ring seal, interpretation of the results in terms of precedents, such as dry friction of rubber or the lubrication of elastomers, is practically impossible and will therefore not be attempted in this paper.

2. Experimental apparatus and procedures

2.1. TEST APPARATUS

In order to test the frictional performance of the O-ring type seal under conditions similar to those encountered during operation of the dry gas seal, a suitable apparatus was required. The main conditions to be simulated during testing are as follows:

(i) unlubricated contact between seal and contacting material,
(ii) variable pressure environment with maximum gauge pressure of 10 MPa,
(iii) minimal leakage through the O-ring seal,
(iv) linear motion of the shaft,
(v) measurement of friction at the commencement of motion of the shaft.

The general layout of the apparatus used is shown in fig. 2. The apparatus itself basically consists of a shaft running inside two O-ring seals located within a container able to withstand a pressure of up to 10 MPa. The force required to draw the shaft through the pair of seals would then be measured at different pressures. This idea is schematically shown in fig. 3. However, the problem with this particular design is that the friction force measured is not for one but for two seals. By considering the frictional force to be split equally between the two O-rings, a degree of

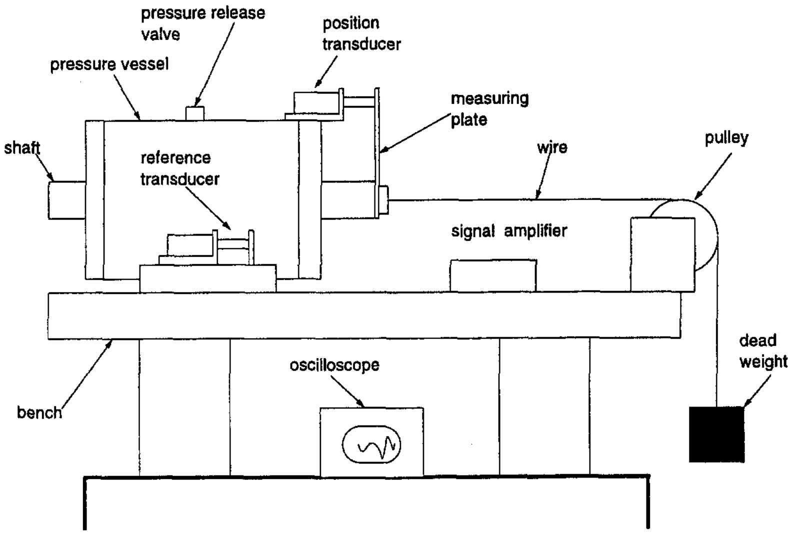

Fig. 2. The general layout of the test apparatus.

approximation was brought into the results. Because the O-ring is made from an elastomeric material it tends to extrude between the shaft and the housing containing it. Obviously, moving the shaft against the direction of extrusion would require a higher force than moving the shaft in the same direction as the extrusion. This simply means that in reality the frictional forces on the two seals would not be the same.

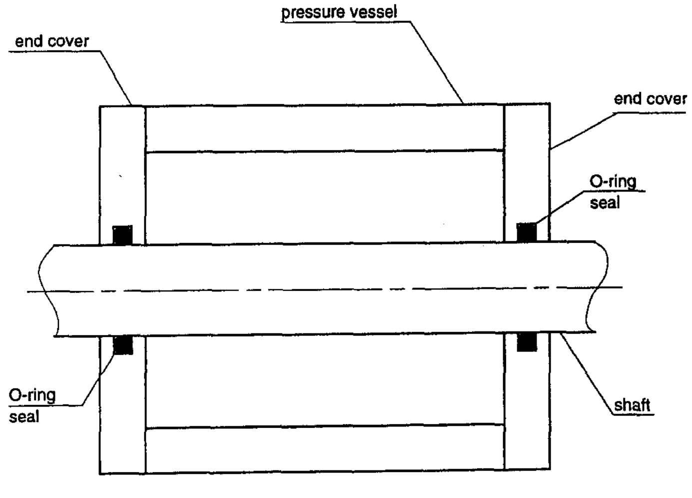

Fig. 3. Diagram showing the idea of friction force measurement in the seal under pressure.

The shaft was made of a mild steel and had surface finish of 0.2 μm. Its diameter was 25 mm and the length 250 mm. The shaft, for safety reasons, had a 50 mm flange located 60 mm from one end. Both ends of the shaft were chamfered to allow for easy assembly into the O-ring seals.

The apparatus was equipped with two inductive transducers acting as a sensor, to measure the axial displacement of the shaft.

2.2. GLAND DESIGN

The O-ring type seal has to be housed in a rectangular gland, schematically shown in fig. 4. The dimensions of the gland are dependent on the application for which the seal is to be used. The primary decision to be made is the degree of squeeze required. Squeeze is defined as the amount by which the seal cross section is larger than the space available between the shaft and the gland as a proportion of the shaft. Thus,

$$\text{squeeze} = [(d - b)/d] \times 100 \quad (\%),$$

where d is the seal cross-sectional diameter and b denotes the gap between the shaft and gland. The degree of squeeze required is governed by the nature of the application of the seal. High-pressure applications require a considerable amount of squeeze in order to maximise the sealing action. This, however, results in the increase in the frictional force which, in turn, produces other undesirable effects such as power loss, heat generation and wear of the elastomer.

The width of the groove, A, should be larger than the cross section of the seal to allow for the seal expansion under operating conditions. If the groove is not sufficiently wide the elastomer can become too large for the gland and begins to extrude between the sealed surfaces. Radial clearance should be kept to a minimum to prevent extrusion of the elastomer.

The tests were carried out at pressures of up to 10 MPa, which represents the

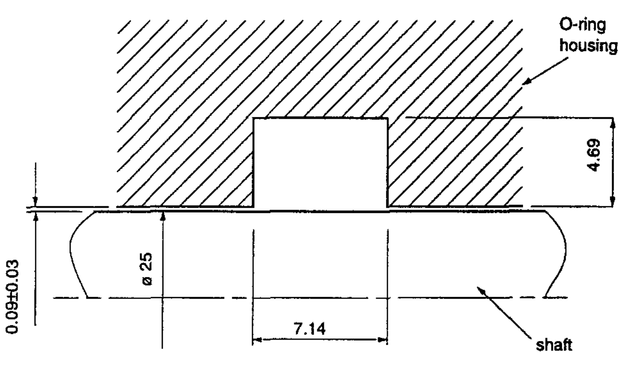

Fig. 4. Important dimensions of the seal's housing.

limit for the O-ring to operate without extrusion rings. For this reason, the radial clearance was kept to a minimum on the low-pressure side of the seal.

2.3. O-RING SEALS TESTED

The tests reported here are primarily concerned with O-rings of special design. All O-ring seals tested had the same dimensions, i.e. inside diameter 24.77 mm (to be used with 25 mm shaft diameter) and cross section diameter 5.33 mm. However, a number of other seal designs have also been tested for comparison of performance. A special design of seal is a PTFE-encapsulated O-ring. The central core is a solid O-ring moulded in silicone rubber. This is surrounded by a tube of PTFE. The seal design aims to combine the sealing performance of an elastomer with the chemical resistance and low friction of PTFE. The core provides the sealing action and the elasticity required for continuous reset of the seal after compression, although not quite as much as a standard elastomer O-ring. Over long periods of time the PTFE tube protects the seal from hardening and thus the elasticity is maintained.

Despite the apparent simplicity of the seal, a number of precautions have to be taken to ensure correct operation of the O-ring seal. The dimensions of the seal and its housing are critical. Poor surface finish of the mating element can also lead to excessive wear of the seal. In general, it is important that the seal comes into contact with no sharp machined edges.

The O-ring materials tested are given in table 1.

2.4. INSTRUMENTATION

The studies required accurate measurement of axial displacement of the shaft (not greater than a fraction of a millimetre) as the rubber materials used had a tendency to creep before large movement of the shaft occurred. The axial displacement of the shaft was measured using two inductive transducers; one was active and the other one passive. The transducers were able to detect any change in distance between the transducer and a ferromagnetic material through a variation in the inductance in the coils within the probe. The output from the two probes was sent

Table 1

Seal material	Shore A hardness
teflon/silicone	90
polychloroprene	70
natural rubber	60
viton	70
silicone	70
fluorocarbon	75

to a conditioning unit that amplified the voltage output. The output was read from an oscilloscope. The reference signal produced by the passive probe can be adjusted by changing the gap between the probe and the steel bracket that is attached to the vessel base. The active probe measured the gap between itself and a strip of steel plate attached to the end of the shaft running through the vessel. As the shaft was drawn out of the vessel the plate moved and the gap between the plate and the probe changed giving a change in signal.

2.5. TEST PROCEDURE

The primary goal of testing was to measure the friction in a number of different seal types. Each seal type was tested at a range of pressures and also with different values for the amount of squeeze on the seal. The following relationships were studied,

(1) effect of O-ring material and construction on friction,
(2) effect of gas pressure on friction,
(3) effect of seal squeeze on friction,
(4) effect of lubrication on friction.

Before testing, the rig was assembled and sealed. The first series of tests was carried out with two teflon-encapsulated O-rings. The squeeze was set at 16%. Then, the shaft was set in position by moving it until the output from the position transducer was at the zero level. Very fine adjustment of the output signal could be achieved using the zero adjustment screw on the signal amplifier unit.

The vessel was then pressurised up to a maximum value. The friction on the shaft was measured by loading the weights hanger progressively, noting any change in the output from the position transducers on the oscilloscope. The loading was continued until the shaft moved. The shaft was then moved back to its original position and some of the weights were removed from the hanger. The pressure valve was opened slowly reducing the pressure until the shaft began to move. A pressure reading was taken at the point shaft movement began under the new loading.

For subsequent series of tests, the endplate inserts were both removed and the seals were replaced with a pair of seals made from a different material. The inserts were then replaced and resealed. The new pair of O-rings was tested in exactly the same way as that described above. This procedure was observed for all the different types of seal tested.

Initially, testing was carried out at 16% squeeze. This value is normally used in static applications. By remachining the endcap inserts, the squeeze was decreased to 9%. This value is normally recommended for dynamic applications. The readings were taken at two different values of squeeze to enable the effect of different squeeze to be compared.

Finally a number of tests were carried out to evaluate the effect of lubrication on friction. Two lubricants were used: a mineral oil, ISO VG 32 and a lithium-based grease, MIL-L-7711A.

3. Discussion of results

3.1. PTFE-ENCAPSULATED SEAL

The results of testing are shown in fig. 5 and refer to the seal tested over the entire 10 MPa operating pressure range at 9% squeeze. As expected, increasing the pressure differential across the seal increased the friction force considerably. Indeed, the frictional force at 10 MPa was over five times that at atmospheric pressure. This increase in friction force is undoubtedly related to the increase in contact area between the seal and the shaft, resulting from inevitable seal extrusion due to the pressure differential.

The curve representing the relationship between friction and pressure is not linear. It shows an initial period of increase in friction followed by a second period where the gradient is less and finally a third period where there is again a rapid rise in friction. A possible reason for this behaviour could be asigned to the fact that the seal is made of two materials with entirely different mechanical properties.

A series of measurements were taken for the PTFE-encapsulated O-ring at a value of 16% squeeze. They are shown in fig. 6 together with those found at 9% squeeze. The graph shows that at 16% squeeze the frictional force values are consistently higher than at 9%.

Repeated tests were carried out but is was found that the scatter of results is considerable and therefore it is rather difficult to suggest any governing law. This is believed to be linked to the lack of dimensional accuracy of the O-rings tested. Analysis of experimental error is given in a later section.

3.2. EFFECT OF O-RING MATERIAL

The performance of a PTFE-encapsulated O-ring was compared to that of an

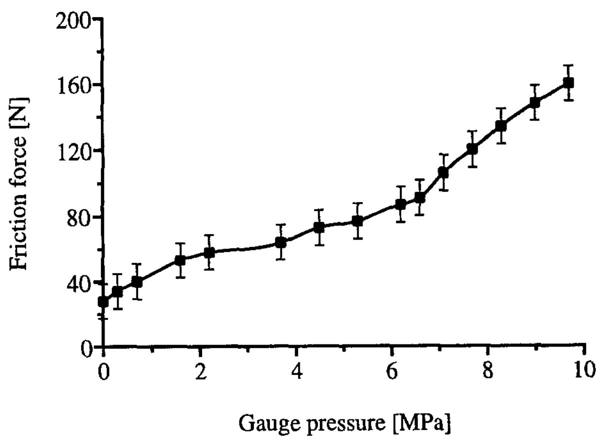

Fig. 5. Friction force against gauge pressure for teflon-encapsulated seal at 9% squeeze.

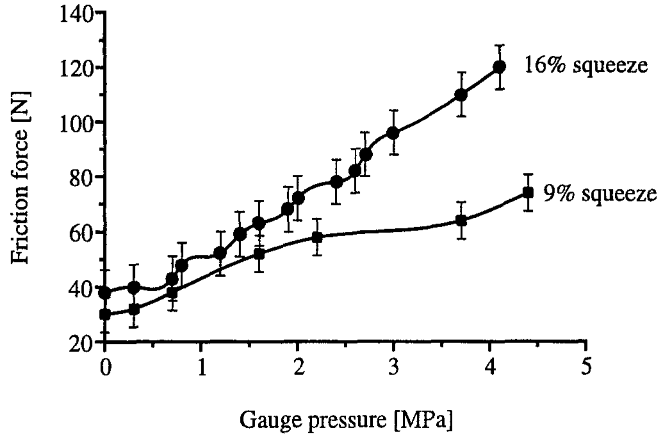

Fig. 6. Friction force against gauge pressure for teflon-encapsulated seal at 9% and 16% squeeze.

ordinary rubber O-ring as shown in fig. 7. It can be seen that the PTFE-encapsulated O-ring shows significantly lower friction than the conventional single-material O-ring type seal. The seals of ordinary design could only be tested over a limited pressure range. The force required to overcome the friction exceeded the limits of the test rig. However, it can be seen from fig. 7 that the rubber-only O-ring seals perform differently from the teflon-encapsulated O-ring seal. The gradient of the curves representing frictional performance of the rubber seals is greater than that of the PTFE-encapsulated seal with the force increasing far more quickly with increasing pressure. This could well reflect the difference in hardness of materials

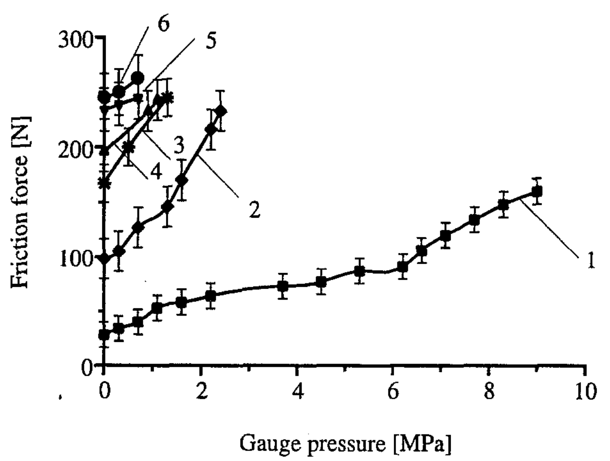

Fig. 7. Friction force against gauge pressure for different materials of the seal at 9% squeeze. (1) Teflon-encapsulated silicone; (2) fluorocarbon; (3) silicone; (4) natural rubber; (5) viton; (6) polychloroprene.

tested. The PTFE-encapsulated silicone O-ring had a hardness of 90 as measured on Shore A scale, while the hardness of the natural rubber O-ring was 60.

This friction behaviour can be explained in the following way. The exclusively rubber seals are all much more compliant than the teflon-encapsulated seals. This means that, when pressurised, the exclusively rubber seals have more tendency to be deformed and extruded between the shaft and the seal gland. As a result of such extrusion there is a greatly increased resistance to motion and hence frictional force.

As mentioned previously, a shaft with its surface ground to 0.2 μm was used throughout the studies and, therefore, the results presented are pertinent to this specific roughness. However, the effect of roughness on the performance of O-ring type seal considered in this paper could be significant.

3.3. EFFECT OF LUBRICATION

A series of tests were carried out on teflon-encapsulated O-rings by adding different lubricating fluids to the shaft/seal interface. The results of these tests are shown in fig. 8. The friction forces measured under lubrication conditions were, as anticipated, significantly less than those recorded under dry conditions. The differences in the magnitude of frictional force between oil and grease used are not very significant. The high-melting-point grease gave the lowest values. The results demonstrate that if any lubricant was to enter the shaft/seal interface the frictional force would vary greatly and in an unpredictable manner. If consistent values are to be found it is clearly important to ensure that lubricating liquids cannot enter the shaft/seal interface during operation.

Since all the friction measurements were carried out at the commencement of

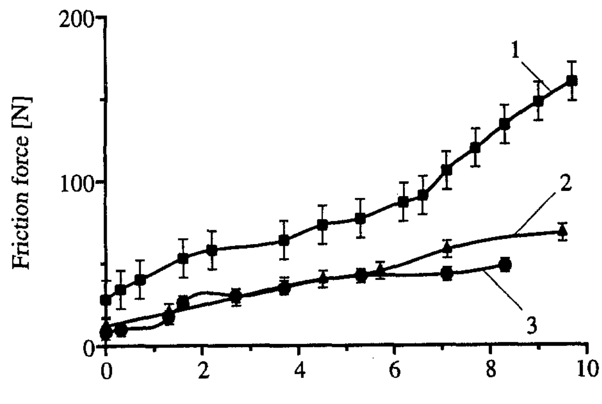

Fig. 8. Friction force against gauge pressure for teflon-encapsulated seal and different lubricating fluids. (1) Dry; (2) ISO VG 32 oil; (3) lithium-based grease MIL-L-7711A.

shaft motion, boundary lubrication was probably the prevailing mode of lubrication.

3.4. ESTIMATION OF EXPERIMENTAL ERROR

In the experiments described a number of areas in which error could occur can be distinguished.

(1) Loading measurement. The loading on the shaft was applied through adding weights to a pulley system. Although it is possible that there are some frictional losses in the pulley these can be considered to be minimal and not merit further consideration.

(2) Pressure reading. A simple, in-line pressure gauge was used, therefore an error in pressure reading is set at 5%.

(3) Seal squeeze. It was said that the degree of squeeze on the seal is critical to the level of friction. The degree of squeeze is affected by the dimensions of the gland and the diameter of the O-ring seal. Some error occurred in measuring the gland dimensions but this remained constant throughout the experiment. The diameter of the O-rings tested was probably not constant across the batch of O-rings. There are many problems associated with manufacturing a rubber seal to an exact tolerance, mainly because of the varying amount of "shrink" that occurs in the rubber after it has been moulded. The tolerance given by the manufacturers used in the tests was ± 0.13 mm on the cross-sectional diameter of the ring. This could affect the degree of squeeze considerably. Taking into account the definition of squeeze presented earlier, an overall error on squeeze can be estimated to be $\pm 23\%$.

The possible error is a large one and it is a very important point for practical applications of O-rings where frictional losses play an important role. The O-ring must be manufactured to very close tolerances if consistent values for the frictional forces are to be achieved.

4. Concluding remarks

The results of testing of O-ring type seals over a range of pressures up to 10 MPa against hard steel surfaces at a constant roughness of 0.2 μm show that the frictional force depends on the design of the seal, contact conditions and pressure differential across the seal. It was found that the frictional force attains quite high values at high pressures. Moreover, consistent values for the frictional force are difficult to obtain because of a large error on seal squeeze which, in turn, results from variations in cross-sectional diameter of O-rings. This lack of frictional force consistency from seal to seal could create problems in practical applications where a precise level of friction is required. The effect of lubrication of the O-ring/shaft interface is the reduction in friction. However, the difference between an oil and grease in the effectiveness of reducing friction is not very significant.

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