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SR-E - Practical Mechanical Sealing with Rotary, Pneumatic and Hydraulic Seal Types plus Gaskets

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Short Description

Whether you consider yourself as amateur or knowledgeable, practical or theoretical, you will find this mechanical seals manual is jam-packed with useful, easy to apply information.

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Whether you consider yourself as amateur or knowledgeable, practical or theoretical, you will find this mechanical seals manual is jam-packed with useful, easy to apply information.

Faced with the bewildering task of selecting the correct seal type and materials of construction for a given application, it's no wonder many end users leave the job to others.

This manual will give you the knowledge and confidence to select correct seal types, analyse failed seals, determine the cause/s of failure and propose practical, remedial action. Learn how, with simple modifications, you can extend seal life and reduce or eliminate causes of premature seal failure.

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Chapter 1: Fundamental Principles

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Fundamentals and Principles

1.1 Zero leakage

Though, the concept of zero leakage is often referred to in fabricator literature, it is quite misleading simply because there exists no accepted definition of the term. However in general practice, a zero leakage specification refers to the use of polymeric seats or seals. While metal-to metal seals fail to meet this requirement, an exception is the gasket used under static conditions where the metal is plastically deformed in order to obtain a leakage of less than 10^{-8}atmospheric cm^3/sec helium. Zero leakage concept can be achieved when the seals are new and assembled correctly.

Zero leakage according to a study conducted by Advanced Technology Laboratories affiliated to the general Electric Company is defined as a leakage of less than 10^{-8} atmospheric cm^3/sec of helium. NASA's manned spacecraft center defines zero leakage as leakage no more than 1.4×10^{-3} standard cm^3/sec GN_2 at 300 psig and at ambient temperature. Other specifications and requirements for valves obtained from standard manufacturers are found to vary between 1.15 x 10-5 standard cm^3/sec to 0 for N_2O_4 and from 8.3×10^{-3} standard cm^3/sec to 1.4×10^{-4}standard cm^3/sec for other gases.

Leakage rate is a relative concept and is generally defined by the amount of leakage that can be tolerated. It must be ensured that the leakage of expensive, corrosive, toxic and explosive fluids is kept at a minimum. There is as such no general rule regarding tolerable leakage rates and these vary according to the application. One fact that must be borne in mind is that an average drop of fluid equals approximately 0.05 cm^3 in volume and this serves as a good guideline for estimating the leakage rate of a fluid that can be tolerated.

1.2 Mechanics of Sealing

Static seals are zero-leakage seals in that they provide a complete physical barrier in a potential leakage path to which they are applied. Towards achieving this objective, the seal must be resilient enough to flow into and fill any irregularities in the surfaces and also maintain sufficient rigidity in order to resist any extrusion into the clearance gap between the surfaces under the full system pressure. The seal material in all elastomeric ring seals is required to have an interference fit with one of the mating parts on assembly. In the case of a solid rubber ring such as an O-ring, the material can be in direct compression or tension or part compression and part tension. A flexible lip seal can be similarly in tension or compression depending on where the section is sealing, the inner or outer diameter.

Irrespective of the type of seal used, a load is thus generated between the seal contact point and the mating area, which again depends on the amount of interference produced during assembly along with the modulus of elasticity of the material. The load distribution depends on the geometry of the section. In the case of dynamic seals, the load must be multiplied by the frictional coefficient, for obtaining the dynamic load to move the seal assembly.

Dynamic seal requirements are quite diverse and conflicting as well. Contact seals call for a high contact pressure on the surface to be sealed, while at the same time requiring the seal to have as less friction and rubbing wear as possible. A compression type seal has high preload and consequently a high amount of friction. Friction can be minimized significantly by choosing a proper seal material. In the case of packed glands, friction can be adjusted by tightening the gland to achieve minimum compression consistent with the degree of sealing. Also periodic readjustment may become necessary, to account for the wear that may occur on the face of the packing. For heavy duty rotary motions, packed glands offer greater durability against high temperatures and severe service conditions. On the other hand, pressure-energized seals are particularly useful for reciprocating duties and offer significantly reduced friction and wear.

Compression seals have relatively higher friction, regardless of the actual pressure. When operating against lower pressures, this leads to more compression and friction than is necessary. While, static friction and thus breakout force will also be higher than running friction, this difference may be negligible with PTFE impregnated or coated packings. Another disadvantage with using compression seals for dynamic applications is the fact that they are usually bulkier when compared with a pressure-energized seal. Compression packings may be designed to operate on dry surfaces, in which case the lubricant is contained within the packing material itself. Compression seal rings and pressure-energized seals or seal sets are normally designed to operate in a fully lubricated condition. Another critical aspect is the oil film thickness which if small can be bridged by surface irregularities, leading to high friction and rapid seal wear. If the film is too thick, the meniscus will break down, resulting in high leakage.

1.3 Leakage

By increasing the preload pressure, most compression seals may virtually be turned into static seals with rubbing contact, by virtue of a reduction in leakage to near zero levels. Equally in the case of pressure-energized or face seals which tend to leak under dynamic working, the leakage levels can be reduced by an increase in the contact pressure, although at the expense of higher friction and shorter seal life. A successful seal design provides the required sealing at acceptable levels of friction and wear and at acceptable leakage rates.

In the case of reciprocating seals, fluid transport results in further leakage. Thus in a reciprocating rod seal, an oil film is carried by the emergent rod. This indicates lubrication of the seal face and is not a necessary case of leakage. The film will normally be withdrawn past the seal during the in-stroke of the rod. In such cases, the true leakage is the amount of fluid actually falling off the rod or the amount that is accumulated on the dry side of the seal. In compression seals, there may be need for a continuous positive leakage in order to ensure sufficient seal face lubrication, unless the packing is of the self-lubricating type in which case increase in compression can be resorted to for stopping the leakage. In flexible seals, true leakage which may well be negligible is the difference between the emergent film and the re-entry film as actually deposited. Any excessive leakage may be traced to a leaking seal or excessive wiping action on the re-entry stroke.

From the above discussion, the following can be established:

- Any presence of oil on the dry side of a seal is normal and desirable.
- There is a distinction between true leakage and apparent leakage,

particularly with lip seals.
- Perfectly dry operation may well indicate lack of lubrication and the possibility of seal failure.

One reason why wipers are required to be provided in a contaminated environment is to prevent dust or dirt particles from clinging to the oil film on the re-entry stroke and further prevent them from being carried back into the seal with the re-entry film. Fluid loss by oil transport is insignificant in typical hydraulic systems. It should be noted that fluid transport loss is much higher for pressure-energized seals operating in a unpressurized state than in a pressurized state. Additionally, double acting seals such as piston seals can produce higher fluid transport losses than the single acting type.

1.4 Purpose of Sealing

The leakage of a liquid between two moving parts in the presence of friction and consequent wear is a standard problem common to any system, be it hydraulic, pneumatic or mechanical. This problem tends to get accentuated under conditions of high pressure and temperature and especially if the fluid is viscous, corrosive or explosive in nature. Prevention of this problem of leakage in a system is accomplished by the process of sealing. Sealing involves the use of seals for closing (sealing) a gap or making a joint fluid-tight. Seals also help prevent entry of foreign particles such as dust, dirt and other harmful contaminants into the system.

Modern machines demand more power, greater speed, longer running life and higher pressures which in turn requires efficient sealing methods along with the need for newer materials and improved design. Towards this purpose, seals are available in a variety of design configurations, utilizing a wide range of diversified sealing principles. For a specific application, consideration must be made with regard to factors such as pressure, temperature, corrosive environment, materials and shaft speed.

1.5 Speed and Pressure

The pressure-velocity or Pv factor is used by seal manufacturers to characterize the ability of a material to resist wear. The Pv factor is the algebraic product of the face contact pressure and the rubbing velocity. This product of pressure and velocity is equal to the rate of heat energy generated, assuming that the value of the frictional coefficient is known. Pv curves are characteristic for a given seal material and are furnished by manufacturers in connection with seal balance relationships.

Thus, the Pv value largely determines the suitability of material combinations for the seal faces and specifically, the amount of heat generated at the face. It also follows that at a given rotational speed; an increase in shaft diameter implies an increase in the Pv value since

Pv = pressure x ? x seal diameter x rotational speed

The Pv value is normally expressed in terms of bar m/sec. Some typical Pv values for different face material combinations are given below.

Table 1.1

Typical Pv values

Face Material Combination	Pv in bar m/sec
Carbon vs ceramic	190
Carbon vs hard faced stainless steel	190
Carbon vs leaded bronze	350
Carbon vs nickel iron	400
Carbon vs tungsten carbide	900

One way of reducing the working Pv value for a seal of given size is by reducing the effective value of P using a balanced seal.

The approximate PV Limits (Mpa-m/sec) for general seals with various combinations of seal face materials and fluids are given in the following table.

Table 1.2

Pv values for various combinations of seal face materials and fluids

1.6 Temperature Considerations and DT Limit

For a mechanical seal to function reliably, the fluid film needs to be maintained between the seal faces. Frictional heat generation at the sealing interface lowers the fluid viscosity and the load carrying capacity of the liquid film. The load bearing capacity can decrease sufficiently and result in heavy contact between the seal face, causing severe wear or face damage. The frictional heat can also raise the temperature of the liquid film at the sealing interface to such an extent that fluid instantaneously changes its phase from liquid to gaseous under the pressure that is present on the low-pressure side of the seal. This phase change often causes an intermittent banging or popping sound and results in severe face damage and excessive leakage. The graphic below is a depiction of the pressure drop and vaporization process during the phase change.

Figure 1.1: *Pressure drop and vaporization*

1.7 Basic Seal requirements

The ever increasing severity of modern day requirements in the form of more power, greater speed, higher pressures and longer running life have led to the introduction of various sealing devices with improved design, newer materials and better sealing methods. In addition to operational demands, sealing devices are also required to conform to environmental, safety and health requirements. Listed below are some important considerations related to seal requirements and applicable to all forms and types of sealing devices in general.

- High strength, resilience and optimum hardness.
- Adequate torsional strength to prevent or withstand spiraling.
- Sufficient flexibility to compensate for wear and also to counteract any vibration or deflection caused by shaft whip, run out, wobble or any other related disturbances in shaft rotation.
- High modulus of elasticity.
- Wide range of flexibility and compatibility to temperature, pressure, corrosion and shaft speed.
- Sufficient balancing to ensure proper operation.
- Good resistance to compression set.
- Ability to be molded to close tolerances.
- Adequate compensation for wear.

- Adequate tear and abrasion resistance.
- Thermal stability over the operating temperature range.
- Sufficient elongation.
- Chemical compatibility with service fluid and capacity to handle a wide variety of fluids.
- Low volume swell in contact with the service fluid.
- Adequate ozone resistance.
- Sufficiently inert and resistant to a wide range of corrosive chemicals.
- Good wearability and good adhesive characteristics.
- Low frictional coefficient and adequate tolerance to friction.
- Low coefficient of thermal expansion.
- Prevent leakage over various pressure ranges.
- Prevent contamination of the fluid.
- Easy to install and remove.
- Good thermal conductivity.
- High resistance to thermal shock.
- Adequate resistance to ageing and weathering.
- Resistance to emission of alpha, beta and gamma rays.
- Dimensional stability under various conditions of temperature, pressure and oxidation and corrosion related phenomena.

1.8 Seal Friction

Seal friction depends on a number of factors such as seal design, material, type of fluid, pressure, temperature, rubbing speed and surface finish. Although the frictional load as such may not be that significant in many applications, the frictional heat that is generated can cause degradation of the material and lubricating film and also result in an increase in leakage, by lowering the viscosity of the fluid. The seal friction in the absence of wedging is given by

$$\mathbf{F_s = ?.\ P_e \times a.b}$$

Where, F_s is the seal friction

? is the coefficient of friction

P_e is the effective contact pressure

a is the seal face contact width

b is the seal face contact equal to ? x d for a circular seal, where d is the contacting diameter

The coefficient of friction is a measure of the materials involved and also the lubrication present. But it is also a function of speed and therefore a more complete equation would be

F_s = ?. V. P_e x a.b

Where V is the rubbing speed

The above formulas find limited use especially so because assembly or interference pressure is normally unknown and thus P_e is largely indeterminate (this being equal to the sum of the interference pressure and fluid pressure in the case of pressure-energized seals). There is another formula which could possibly be more useful although factors such as rubbing speed and surface finish are ignored.

F_s = K ? P_e x D

Where, K is the empirical factor specific to the design of the installed seal and working under design conditions

D is the seal diameter

A true compression seal has a rigid section under compression with wedging unlikely to occur and friction being largely independent of internal pressure. The actual compression pressure may in such cases be considered as the effective pressure although it is difficult or sometimes impossible to determine. The coefficient of friction may also be unknown and variable although it may be possible to estimate this with reasonable accuracy.

As seen from the discussion above, unless the data is evaluated empirically or on a comparative basis, specific solutions are difficult to obtain. The above equations can be utilized only to determine differences if any in performance and also friction on similar compression seals (same type and material) belonging to different sizes.

Frictional Coefficients

Generally speaking, harder the material, higher is the friction and vice versa. This is more applicable to elastomers where the hardness can vary, but holds good only under conditions of low pressure. The dry friction coefficient for typical seal materials rubbing on smooth and dry surfaces may have a value between 0.4 and 1.0. This range becomes much lower (0.02 – 0.10) for lubricated surfaces and hold particularly true for elastomers. Fabric materials and impregnated fabrics have similar values of ? but with less variation (0.04 – 0.08) for lubricated conditions. While lubricated leather normally tends to have low friction, the values of ? for leather seals are often higher than elastomers. This is because leather rings are often used in conjunction with rougher rubbing surfaces.

The frictional coefficient is also a function of pressure, although the actual relationship is not quite clearly established. Friction is basically highest at low values of pressure and can vary with different seal sections. This is depicted in figure 1.1 below.

Figure 1.2*: Frictional coefficient vs. pressure*

The table contains the coefficient of friction values for various face combinations

Table 1.3

Coefficient of friction values for various face combinations

The variation of friction with pressure also depends on the surface finish. Let us consider the example of three different cylinder finishes as shown in the figure 1.2. It is seen that there is a rapid increase in friction with a rise in the working pressure in the case of a rougher surface finish and texture of a cold hammered tube, compared with honed or burnished tubes.

Figure 1.3*: Frictional differences with surface finish*

Friction and Speed

The variation in friction with rubbing speed is more clearly defined and generally follows three different stages. While static friction is normally high, the coefficient of friction falls to a low value at low speeds, as soon as break-out is initiated. With increase in speed, the value of the frictional coefficient increases up to a first peak. Then onwards, further increases in speed result in the frictional coefficient falling to a minimum value and subsequent increase with continuing speed increase. This is a general description of the friction-speed relationship and can be further modified for various operating conditions and seal design and material.

Figure 1.4*: Variation of friction with rubbing speed*

Pressure-energized Seals

For a flexible seal of the pressure-energized type, the effective pressure is equal to the sum of the preload and the fluid pressure. For higher fluid pressures, the value of the preload pressure is less significant and may even be ignored. These seals are prone to extrusion and wedging at higher pressures particularly with large clearance gaps. Wedging leads to a considerable increase in friction. The basic friction equation for these seals is given by

$F_s = ?\ V\ (p + P)^2\ ?\ a.b$ **or**

$F_s = K\ ?\ V\ (p + P)^2 \times D$

Where p is the preload interference pressure

P is the fluid pressure

V is the rubbing velocity

K is a constant depending on the seal type.

The tendency for a seal to wedge is enhanced by a rough surface, lack of lubrication and high reciprocating speeds. Wedging is unlikely to occur with small clearance gaps in the range of 0.055-0.127mm (0.002-0.005 inch). For clearances greater than 0.25 mm (0.010 inch) the possibility of wedging always exists.

Friction of Reciprocating Seals

For these seals, the coefficient of friction primarily varies with seal design and also with pressure, with the frictional loss varying as the frictional coefficient and rubbing speed. Friction is found to increase with increase in operating pressure for these seals, but the ratio of friction loss to the theoretical pulling power of the cylinder is reduced (the friction loss being expressed as a percentage of the theoretical pulling power when used as piston or rod seals in a cylinder). A reduction in the ratio of friction loss to the theoretical pulling power means that the coefficient of friction is smaller. This is because the liquid pressure in the lubrication film increases, thereby resulting in the effective filling of the sealing surface roughness.

The heat that is generated on account of friction, affects the seal material at temperatures above 50º C. In such cases, the seal material may swell or lose its resistance or hardness. Also oxidation of the oil occurs at high temperatures with the result that lubricant properties are affected and leading to further damage of the seal material.

1.9 Wear and Seal life

Normal wear of the seal material can cause the seal to lose its ability to function. This wear is greater at the time of starting and at low speeds. Seals may also lose their ability to function on account of erosion of the seal material which occurs when the fluid under pressure flows over the sealing surface and impinges on an area of deterioration. The first indication of wear is when the seal is no longer capable of maintaining the required contact with the sealing surface, at low pressures. The sealing may continue to be adequate at high pressures as long as the pressure is maintained. Wear is also often aggravated by lack of lubrication, shaft irregularities, excessive frictional heat, a soft seal compound etc.

The life of a seal cannot be predicted with any amount of certainty since it is in turn dependent on a lot of factors. The normal life expectancy of seals also varies considerably from one application to the other. While a life expectancy of around 400 hours may be considered normal for a hydraulic cylinder seal, the normal life of a lip-type seal on the other hand may be only 1000 hours. In the event of the seal life being significantly lesser when compared with the average, for a particular application, then it is likely that an unsuitable seal was selected and that the operating conditions turned out to be more severe than was originally expected.

1.10 Texture

The texture of the surface against which a seal rubs, has a significant bearing on friction, wear and seal life. Texture refers to both, surface roughness or irregularities as also the pattern of these irregularities. While the roughness is capable of sampling measurement, pattern can be described only empirically. The standard method of measuring roughness is by taking an average value of the profile variation from a center line over a sampling length, as shown in figure 1.4 below. This method is known as Center Line Average (CLA), commonly expressed as Ra in British standards and also adopted as ISO standard.

It is designated as Arithmetical Average (AA) in the United States.

Figure 1.5*: Average Roughness value- R_a*

The R_a value in most measurements is directly obtained either in micro meters or micro inches. **ISO R1302** relates the nominal values of R_a to equivalent roughness grade numbers. The table below relates the different nominal R_a values and their equivalent ISO roughness grade number.

Table 1.4

Relationship between nominal R_a values and ISO Roughness grade number

Nominal R_a values		**ISO Roughness Grade Number**
?m	**?in**	
50	2000	N12
25	1000	N11
12.5	500	N10
6.3	250	N9
3.2	125	N8
1.6	63	N7
0.8	32	N6
0.4	16	N5
0.2	8	N4
0.1	4	N3
0.05	2	N2
0.025	1	N1

Surface roughness may also be expressed in terms of maximum roughness, depth or the distance between the peak and base line measurements over the reference length as shown below. Maximum roughness designated as R_t is measured in the same units as R_a. Both, the values of R_a as well as R_t play a significant role in determining the optimum surface finish.

Figure 1.6: *Roughness depth- R_t*

Recommended surface finishes

Surface finishing helps provide a surface that causes the least seal wear. Rod seals can be damaged by fine abrasive particles which may adhere to a rough surface and should therefore have a low surface roughness value, a surface similar to the best hard chrome and resistant to corrosion. The ideal value would be 0.16-0.40 ?m (R_a) or 1.0-2.5 ?m (R_t).

Evaluation of the surface properties must also take into account the operating pressure. Friction is greater under higher operating pressures, since the oil film between the sealing surface and the seal is thinner. Under such conditions, a surface quality approaching lower values must be chosen. The surface quality for a static seal housing will be around 1.6?m (R_a) or 10?m (R_t).

The roughness and pattern of the surface finish can vary according to the machining process adopted. The table below contains some typical values that are likely to be obtained with different process, although the values may themselves vary, depending on the machine tool and the material used.

Table 1.5

Surface finish from various machining processes

Machining Process	**Surface finish**	
	?m	**?in**
Super finishing	0.025 - 0.25	1 - 10
Polishing	0.05 – 0.5	2 - 20
Lapping	0.05 – 0.5	2 - 20
Buffing/Burnishing	0.125 –0.5	5 - 20
Honing	0.125 –1.625	5 - 65
Grinding	0.125 –1.75	5 - 70
Diamond bored and Turned	0.25 – 0.5	10 - 20
Turning	0.5 – 6.25	20 - 250
Boring	0.5 – 6.25	20 - 250
Reaming	0.9 – 3.00	35 - 120
Broaching	0.9 – 3.00	35 - 120
Milling	0.9 – 6.25	35 - 250
Shaping	1.5 – 12.5	60 - 500
Planing	1.5 – 12.5	60 - 500