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DESIGN CONSIDERATIONS IN MECHANICAL FACE SEALS FOR IMPROVED PERFORMANCE

I. BASIC CONFIGURATIONS

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ABSTRACT

Basic assembly configurations of the mechanical face seal are described and some advantages associated with each are listed. The various forms of seal components (the primary seal, secondary seal, etc.) are illustrated, and functions pointed out. The technique of seal pressure balancing and its application is described; and the concept of the PV factor, its different forms and limitations are discussed. Brief attention is given to seal lubrication since it is covered in detail in a companion paper. Finally, the operating conditions for various applications of low pressure seals (aircraft transmissions) are listed, and the seal failure mode of a particular application is discussed.

INTRODUCTION

In a highly technical society the fluids which must be sealed range from water (automobiles, reactors, submarines) and oil, to liquid oxygen and toxic chemicals. The mechanical face seals developed for these applications have many diverse forms, from the low cost automotive water pump seals to the very sophisticated seals for liquid oxygen turbopumps such as are used on the space shuttle. It is well recognized that seals can have a significant cost impact in regards to maintenance and downtime. An additional consideration that has received recent attention is the health hazard. It is now realized that personnel exposure to even low levels of some substances can have serious health consequences; often the health damage can not become manifest until late in life.

In regard to chemicals which may be a health hazard, ref. (1) points out that one source of worker exposure comes from leaks in valves and pumps, and states that prevention of a health hazard (occupational disease) is highly desirable since it tends to be chronic, untreatable, and fatal; and may go unnoticed if it does not result in an unusual group of clinical manifestations (the case of a rare neoplasm of the liver found in workers exposed to vinyl chloride is an example). An additional point in regard to the seriousness of industrial carcinogens is made by ref. (2), in which it is stated that, unlike cigarettes smoking where the risk of lung cancer in ex-smokers is only slightly above that of lifetime non-smokers, industrial carcinogens cause an extra risk which does not appear to drop after exposure ends. Specifically, ref. (2) lists numerous substances which appear to be associated with excessive lung cancer mortality, among these are polycyclic aromatic hydrocarbons, bis chloromethyl ether (BCMHE) and chloromethyl methyl ether (CMME). In this regard, the Occupational Safety and Health Administration (OSHA) has developed exposure limits for over 300 substances and additional substances will probably be added as research information is developed.

In general, face seal application is largely empirical, being based on past experience. However, current technical needs (e.g., breeder reactors) and the previously cited occupational disease considerations have placed an additional burden on a technology, which seems to sense, to have been barely adequate in certain applications.

The objective of this study is to review the basic seal configurations and design considerations; a companion paper, ref. (3), summarizes the current thinking on seal lubrication, which is a major factor in improved life. In this paper, discussion is limited to the conventional face seal. High-performance special-purpose seals that employ thrust bearing geometry machined into the primary seal surfaces (hydrodynamic seals) are not addressed; an introduction to those can be found in ref. (4).

DISCUSSION

Applications

The mechanical seal is recognized as one of a family of devices used in the general area of fluid sealing of rotating shafts. This general area of sealing is represented by the pyramid in fig. 1. At the bottom, or base, are packings which were the earliest and still the most frequently used solution to sealing problems. Moving up the pyramid, there is an increasing complexity of sealing devices including O-rings, lip seals and mechanical seals, beginning with the simpler varieties used in the appliance and automotive industries. Moving up the pyramid further there are aircraft, marine, and chemical process industry applications. Finally, at the peak there are sophisticated aircraft and nuclear requirements, and rocket component sealing systems.

Arrangements of Seal Components

One conventional arrangement of seal-face components is shown in fig. 2. The primary-rings has axial flexibility, usually contains the primary seal diameter, and also has angular flexibility. The axial flexibility allows accommodation of axial displacements that arise for various reasons (e.g., face run-out, thermal growth differences and tolerance variations). The secondary rings can be of various designs, such as elastomeric 'O'-rings, bellows, or piston rings.
the rotating primary-ring with the sealed liquid at secondary rings.

Bence of centrifugal force on both the primary and shaft diameter.
or similar means which assemble directly on a smooth
attached to the shaft by set screws, friction fits,
straight shaft auctions without affecting the seal
pumps, is the rotating primary-ring configuration
mating ring are rotating.

are encountered where both the primary-ring and the
O.D. or the I.D. (see figs. 2 and 3). Occasionally
be identified; these step a rotating primary-ring with
the I.D. These are:

The most common configuration, especially in
pumps, is the rotating primary-ring configuration
(fig. 3); the principle advantage of this configuration
lies in the relative ease of attachment to
straight shaft sections without affecting the seal
face alignment. The primary-ring assembly is easily
attached to the shaft by set screws, friction clips,
or similar means which assemble directly on a smooth
diameter.

Nonrotating primary-ring configurations (fig. 2)
is often found in applications where speed is limited,
or where the rotating speeds are high. The principle
advantage of the nonrotating primary-ring is the
absence of centrifugal forces on both the primary and
secondary rings.

The rotating primary-ring with the sealed liquid
at the O.D. has a number of advantages as compared to
the rotating primary-ring with the sealed liquid at the I.D. These are

1. Centrifugal force tends to retard leakage
2. Centrifugal force tends to centrifuge solid
particles away from the primary seal and
springs, thus there is a tendency to be self
cleaning
3. The sealed pressure tends to put the seal
rings into compression

On the other hand the rotating primary-ring I.D. con-
figuration has advantages in that it is easier to in-
stall; and in sealing corrosive fluids, the seal can
be designed so that metal parts do not come into con-
tact with the corrosive fluid.

Tandem Seal Arrangements

For dangerous liquid, seals are often used in
tandem (fig. 4) with a barrier liquid between the
seals. The barrier liquid may be pressurized to
level slightly above that of the sealed process liquid
in order to preclude any leakage of the sealed liquid;
in other designs, the barrier liquid is held at a
pressure equal to or even slightly below that of the
process liquid. Selection of the barrier liquid pres-
dure depends on restrictions regarding leakage rates
of the barrier liquid into the process liquid, or on
the process liquid leak rate into the barrier liquid.
It should be noted that holding barrier liq-
uid pressure above that of the sealed process liquid
is no absolute guarantee against leakage of the pro-
cess liquid into the barrier liquid (and then from
leaking into the atmosphere). In this regard ref. (5)
cites the experience with deep well irrigation pumps
in which water leaked across the seal into the bearing
lubricant side even though the lubricant was at a
higher pressure than the water. Evidently the seal
acts as a small pump, transferring the fluid from a
lower pressure to a higher pressure. The mechanism
responsible for this pumping action was investigated
by several researchers (refs. (6) and (7)) and one
explanation is that certain combinations of parallel
misalignment and primary-seat surface waviness re-
sult in a pumping effect against the higher pressure.
(Surface waviness and parallel misalignment are dis-
cussed in a later portion of this paper.) Obviously,
if the process liquid is very toxic, and very low
levels of atmospheric contamination must be main-
tained, then the possibility of process liquid transport
across a seal into a region of higher pressure is a
major concern.

It is apparent that many different combinations
of tandem seal configurations can be constructed from
the four basic seals (rotating primary-ring and non-
rotating primary-ring, either of which can be O.D.
or I.D. pressurised). The advantages and disadvantages
of each involve cooling, leakage, safety, and cost
considerations which are outside the scope of this
paper. Several of many possible configurations are
shown in fig. 4; and it should be noted that tandem
seals are often used with heat exchangers to maintain
the seals at a desired temperature level.

The Primary-Seal

The primary-seal performs two functions; these
are, (a) effectively retards the pressurized fluid
and (b) acts as a thrust bearing using the sealed
liquid as a lubricant. (There is a very close paral-
lal between seal face operation and thrust bearing
operation.) When selecting materials for use as seal
faces, a general rule is that all good face seal ma-
terials must be good bearing materials, but not all
good bearing materials will necessarily make good seal
faces. Carbon-graphite materials of various types,
used in combination with some other compatible hard material, are used as face materials in the vast majority of mechanical seal applications. Table I lists a number of materials used for seal faces and the typical environments and associated material combinations that are frequently used.

In seals of all types, a basic difficulty is found in the fact that the seal lubricating film is very thin (in the range of 1 micron or less) and therefore, very small surface irregularities, elastic displacements (thermal bumps, etc.) and face runout motions have a dramatic effect on the lubrication of the seal. Thus the primary-seal cannot, in general, be visualized as two perfectly flat and parallel surfaces; some possible geometries are illustrated in Fig. 5. There have been many hypotheses put forth to explain the mechanism (or mechanisms) responsible for the development of the lubricating film pressure that acts to separate the primary-seal faces. These hypotheses include the following: surface angular misalignment, surface waviness, surface asperities, vaporization of the fluid film, axial vibration, and thermal deformation; current work is reviewed in the companion paper (ref. 2).

With reference to the possible primary-seal geometries of Fig. 5, waviness (geometry a) and angular misalignment (geometry b) are most likely sources of hydrodynamic pressures that contribute to face-seal lubrication. Coning (geometry c) affects the hydrostatic force balance (not significant in low-pressure seals) and this is also discussed in ref. (2). Externally imposed axial vibration (geometry d) can produce useful squeeze-film pressures, but it is judged not to be significant because of the high damping that is associated with viscous fluids and secondary-seal friction. Parallel misalignment and shaft whirl (geometry e and f) do not produce significant fluid-film pressures directly but can affect the transport of fluid into and out of the primary seal. In the case of a rotating ring (geometry e), parallel misalignment is present, some points on the seal surface enter and leave the primary-seal during each revolution. This radial velocity component can affect both leakage and lubrication.

The Secondary-Seal

The secondary-seal elements may be divided into three basic types:

(a) Compression packings
(b) Automatic packings
(c) Bellows diaphragms

Compression packing was one of the earliest used in mechanical seals (see Fig. 4). The principle feature of the compression packing system is that it utilizes a mechanical load on the packing. The typical arrangement, shown in Fig. 6(a), consists of a small packing box, one or two rings of compression-type packing and a gland follower actuated by the same spring load used to load the seal faces. Dynamic packing rings used with this construction include a variety of materials, some of which are molded synthetic elastomers, asbestos, and synthetic binders, carbon-base packings and plastic materials, the most common of which would be PTFE.

The term "automatic packings" apply to all of those packing rings and devices that are self energized by the sealed pressure and do not normally require an auxiliary mechanical load to maintain sealing contact. Figure 6(b) illustrates a typical mechanical seal using an automatic packing ring. Some of the variations commonly used in mechanical seals include O-rings, U-rings, and piston rings. Seals using piston rings as the secondary seals generally use conventional piston rings of various designs. Figure 6(c) illustrates a typical high temperature piston ring assembly. These are usually special purpose seals operating in severe environment beyond the capabilities of the elastomers described previously.

The third category of secondary seals, bellows and diaphragms, can be subdivided into two groups:

(a) Elastomer bellows and diaphragms
(b) Metal bellows and diaphragms

Figure 6(d) illustrates the typical elastomeric bellows seal assembly. Sliding packing surfaces and the friction associated with them are eliminated. Within temperature and pressure limits and compatibility with the media, the elastomeric bellows is one of the most widely used of all mechanical seals.

Figure 6(e) illustrates a formed metal bellows arrangement. This was the original construction used for all metal seal construction.

Figure 6(f) is a later development illustrating a welded metal bellows seal assembly usually found in applications involving high temperatures and highly reactive media where synthetic elastomers are incompatible with the media. The welded sealing bellows requires less space, has softer spring rates, wider operating ranges and has higher pressure capabilities than the formed bellows.

Because of the all metal construction, these welded metal bellows, fabricated from corrosion resistant alloys, will be found in high temperature service and in various types of corrosive media. Several additional advantages of this construction are the elimination of sliding interfaces and, therefore, the elimination of abrasion and fretting corrosion. The metal bellows also eliminates the need for separate springs since the bellows serves both the sealing function and the face loading function.

In general, the secondary-seal has a dramatic effect on the performance of the seal, but this is not well understood. Also the frictional force of the secondary-seal can change substantially with time, and this alters the seal's performance. In extreme cases, the secondary seal friction increases to a point where the seal sticks open; an example is "O" ring swelling due to lack of compatibility with the sealed fluid.

Loading Elements

Figure 7 illustrates the various springs that may be found in conventional mechanical seals starting with the single spring (fig. 7(a)), the multiple spring (fig. 7(b)), wave spring (fig. 7(c)), belville springs (fig. 7(d)) and bellows (fig. 7(e)).

The single spring has the advantage that it is a single component having a relatively large wire cross section with less susceptibility to degradation through corrosion. The disadvantage are that a new spring is required for every shaft size, the single spring is a long assembly, and centrifugal force may affect the coils at high speed causing them to open up.

Multiple springs permit the use of a standardized spring for a variety of sizes. Loads are varied by simply varying the number of springs in the assembly. Multiple springs will operate at a relatively shorter operating length than the single spring design. Disadvantages are described as the number of added parts...
in an assembly, and the potentially greater effect on spring loads for a given corrosion rate.

Wave springs (fig. 7(c)) are used in seals designed for very small axial spaces. Their advantages are principally lightweight and limited space requirements. The disadvantages can be described as a low available working length, degradation of the spring loading due to corrosion of the thin materials, and the probability of void under conditions of high vibration or excessive movement.

Belleville springs, which are made up of a series of disk washers (fig. 7(d)), have been used in special seal construction. The advantages of this technique is that rates can be changed by simply adding or subtracting washers. Projected loads from belleville springs will be generally more diametrically uniform than any of the above springs. The disadvantages of the belleville spring is its initial cost due to tooling required and the relatively high spring rates that must be reduced by a multiple number of convolutions. There is also a greater likelihood of fouling the spring system through deposits of foreign material on the inside convolutions of the belleville plates.

Figure 7(e) illustrates a self-contained bellows arrangement which was the original construction used to obtain an all metal seal assembly. A later development is the welded nesting bellows, a seal assembly wherein a bellows is usually found in applications involving high temperatures or highly reactive media where synthetic elastomers would be incompatible with the sealed liquid. The welded bellows require less space, lower spring rates, wider operating ranges and has higher pressure capabilities than the formed bellows. Finally, a seal using magnets instead of springs is shown in fig. 7(f); this provides a very compact assembly.

Pressure Balance

Pressure balance is defined as the ratio of two areas. Referring to fig. 8 (a typical balanced seal) the primary seal area, which is the area bounded by the outer and inner diameters of the sealing face, has been designated \( A_H \). The closing force area, which is the area subjected to the sealing pressure and bounded by the secondary seal balance diameter and a primary face diameter has been designated \( A_P \). The pressure balance ratio is the value of \( A_H \) over \( A_P \).

Expressed in terms of the face diameters, the pressure balance is derived as shown in fig. 8.

In simpler terms, it can be stated that the unit hydraulic loading transmitted to the seal face of a balanced seal is some value less than the pressure sealed. The range of pressure balance used for balance seals will generally vary over a range from 50 to 85 percent dependent upon the characteristics of the application. The great majority of seals are balanced in the range of 70 to 75 percent.

When sealing an incompressible liquid and operating on a lubricating film, the pressure drop across the sealing face as calculated for incompressible flow is a straightline function.

The average film pressure will, therefore, equal 50 percent of the pressure sealed. If this factor is equated against the seal's hydraulic balance, it will be seen that the residual load available to maintain the faces in contact on a 70 percent balance seal would be theoretically 20 percent of the pressure sealed, which is the net load available to maintain the faces in contact.

If we consider a gas as the fluid sealed, the flow characteristics would be that of a compressible fluid. The pressure profile across the sealing face would occur in the form of a curve and the required balancing characteristics would be adjusted accordingly.

The unbalanced seal (fig. 9), is one in which the full value of the pressure sealed is impressed upon the sealing faces. Unbalanced seals will generally be used for low pressure service and balanced seals will be applied to high pressure applications.

Arbitrary recommendations ranging from 100 to 200 psi are usually established for the operating pressure limits between balanced and unbalanced seals. In practice, there is no fixed value for this decision point and all of the operating conditions as well as the characteristics of the media sealed may influence the decision. Unbalanced seals are operated successfully in service that range as high as 1200 psi involving special operating conditions and duty cycles.

Figure 10 illustrates some variations that may be applied to the primary seal face detail. A wide face and a narrow face are illustrated. The hydraulic balance of either face can be identical. Nevertheless, the two faces may perform differently. Torque requirements and heat generation will be a function of the total load and hence a function of the total face area. The narrower face will result in lower torque, lower heat generation for a given set of operating conditions. This would imply that all faces should be made as narrow as possible. This must be tempered by some practical considerations. The most important point is that the lapped faces are flat to the limits of a manufacturing tolerance. When subjected to mechanical and pressure loadings, further variations in face flatness will occur. Since the film pressure is no longer uniform, contact will take place at some points and the load will be distributed between some fluid film and some contact area. For a given variation in flatness, the shorter leakage path of the narrow face will result in higher leakage rates than the wide face seal arrangement.

There are no fixed boundary lines or hard-and-fast rules for face width. In general, narrower faces are used for high speed where the lower torque and heat generation may be critical to the performance. Wider faces will be found on high pressure, low speed applications where lower stresses will provide lower distortion, thus maintaining the flatness characteristics, and this promotes lower leakage rates.

PV Factor

An arbitrary mathematical relationship, often used to evaluate the severity of an application, is the PV value which is the product of the pressure expressed in psi times the velocity expressed in feet per minute.

The resulting PV value is a factor that is useful for establishing a relative degree of severity of an application or for establishing a rough comparison of performance between several applications operating under differing operating conditions. It has developed through informal usage by the seal manufacturers, the seal users, and material suppliers. There are several variations in the calculations that have evolved and it is necessary to be sure that the approach is uniform before comparable relationships can be established and inferences made from those values.

Figure 11, formula 1, is the simplest version as it is the simplest product of the pressure and velocity. In the calculation, there is no consideration given to film pressure differentials across the face or geometric characteristics. It is often used to identify the test material to be used which is conducted with a straight mechanical load on the sealing faces and with no pressure differential across the

In this calculation, there is no consideration given to film pressure differentials across the face or geometric characteristics. It is often used to identify the test material to be used which is conducted with a straight mechanical load on the sealing faces and with no pressure differential across the
Formula 2 uses the balance factor to reduce the pressure sealed to the unit loading the seal faces actually experience. In the case of an unbalanced seal, the balanced factor is 100 percent; therefore, the calculated value would be the same for formula 1 or formula 2. Only the balanced seal would show a proportionately reduced PV value.

Formula 3 is a further modification that has been used by some seal engineers. In this case, the unit loading is further reduced by subtracting the pressure exerted by the lubricating film between the faces. We have previously indicated that an incompressible fluid has a substantially straight line pressure drop and that the average pressure is one-half of the differential. This value is then subtracted from the gross hydraulic load resulting in a net differential hydraulic load which is used for the PV calculation.

The concept of the PV value is an imperfect relationship but, if used in the proper context, it can be a useful reference tool. Of the three versions, formula 2 provides the most logical and realistic comparison; but, the most important concern is to know the basis of the computation when making comparisons of face material performances.

In applications where the pressure differential is very low, such as in helicopter transmission seals in which the pressure differential may be zero, the spring loading magnitude is critical. In these cases the PV factor is calculated by using the contact pressure produced by the springs on the primary-seal area. Examples of PV factors used in applications sealing a gear box lubricants are given in Table II; the first 9 entries are helicopter transmission applications and the data was given by ref. (8). The last two entries in Table II are reported problem areas. Comparison of the PV factors of the last two entries with the first nine does not suggest operation outside the bounds of the state-of-the-art; this illustrates that the PV factor is not an absolute guide for prediction of successful operation.

Figures 12 through 14 depict the type of surface distress that is sometimes found in these types of very low pressure seals. The seal was operated under the same conditions listed in line 10 of Table III; these conditions are:

2089 N per min (6870 ft/min) sliding speed;
7.8 N/cm$^2$ (11.36 psi) face pressure due to spring load of 20,904 (ft/min x psi) PV factor. Instead of a face contact pressure some seal engineers use closing force per unit length of the circumference. On this basis the load due to the springs for the application in line 10 Table II is 2.14 N/cm or 1.22 lbf/in.; and briefly stated, for the seal to operate successfully the primary-seal face must act as an oil lubricated thrust bearing and produce a load capacity equivalent to 2.14 N/cm of seal circumference. However, as indicated in figures 12 through 14, heavy surface distress occurred because of lack of adequate lubricating (load support) film. The seal shows severe heat checking (fig. 12) with temperatures probably reaching over 811 K (1000° F) as indicated by the soft pearlite produced at local hot spots (fig. 12). The wear profile of the rotating seat matched that of the carbon ring and this is shown in fig. 14.

Seal Lubrication

The engineering of seals can involve fluid mechanics, heat transfer, lubrication theory, thermodynamics, chemistry, physics, metallurgy and dynamics, to mention a few of the most frequent areas of concern. Seal problems may consist of superposition of effects which can be intercorrelated. Usually, each effect can be analyzed by itself, but then integrated effects must be evaluated for a complete analysis of a sealing system. As indicated previously, seals are characteristic of surfaces in relative motion separated by a very narrow gap. In order to maintain proper operation, very small differences in the dimension of seal parts must be maintained. Deformations in geometry due to imposed thermal gradients, frictional heating, pressure and mechanical and inertial forces must be held to a minimum. In most cases, any deformations must be no more than microvalues.

Seals operate in many lubrication regimes, depending upon the type of seal, sealed fluid, the application and related characteristics.

Figure 15 illustrates the various seal lubricating regimes that can exist (ref. (8)). In a companion paper (ref. (2)) seal lubrication is discussed in detail; the current theories are reviewed and seals are classified according to the lubrication operation mode.

CONCLUDING REMARKS

The seal descriptions in this paper show that mechanical face seals appear in many diverse forms and that a basic difficulty of all is the very small thickness of the lubricating film (on the order of 1 micron). And since the film is very thin, very small irregularities or motions of the primary-seal can have a dramatic effect on the performance of the seal. (For a detailed discussion on seal lubrication the reader is referred to ref. (2).)

Generally, seal lubrication is not well understood and this is evident when compared to bearing lubrication in which performance can be closely predicted; in seals, prediction of performance (lubrication or life) is often not possible. The importance of the secondary seal is stressed, in general, data is lacking in regard to secondary seal friction which must be offset by the spring load or by seal pressure balance; here current practice depends on a vast amount of experience. Finally, the limitations of the PV factor should be kept in mind; in this regard, insights provided by current thinking on lubrication can help guide the application of seals.

REFERENCES


### Table I. End Face Seal Materials and Environment Combinations

<table>
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<tr>
<th>Environment</th>
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*The seal diameter is the primary ring nose inside diameter.*
Figure 1. - Shaft seal usage.
THE PRIMARY RING HAS
(a) AXIAL FLEXIBILITY (SPRING LOADED)
(b) SECONDARY-RING SEALING DIAMETERS
(c) ANGULAR FLEXIBILITY (NOSE WILL TEND TO ALINE ITSELF TO ANGULAR
MISALINEMENT OF THE SEAT FACE)

Figure 2. - Arrangement of seal components (nonrotating primary ring).

Figure 3. - Mechanical face seal with rotating primary ring.
Figure 4. - Tandem seals, rotating primary-ring type.
(a) WAVINESS.
(b) ANGULAR MISALINEMENT.
(c) CONING.
(d) AXIAL VIBRATION.
(e) PARALLEL MISALINEMENT.
(f) SHAFT WHIRL.

Figure 5. - Possible primary-seal geometries.
Figure 6. - Types of secondary seals.

(a) COMPRESSION PACKING.

(b) AUTOMATIC PACKING, ELASTOMERIC.

(c) AUTOMATIC PACKING, PISTON RING.
Figure 6. - Concluded.
Figure 7. - Face loading elements.
Figure 8. - Pressure balance for sealed pressure at the O. D.

PALANCE RATIO = \frac{A_P}{A_I} = \frac{D_0^2 - D_B^2}{D_0^2 - D_I^2}

Figure 9. - Unbalanced seal.
Figure 10. - Wide and narrow seal faces.

Figure 11. - Pressure balance formulas.

1. $PV = \Delta P \times V$
2. $PV = B_H \times \Delta P \times V$
3. $PV = (B_H \times \Delta P - 0.5 \Delta P)V$
Figure 12. - Micrograph of mating ring sealing face. X12.

Figure 13. - Micrograph of mating ring cross-section. X200.

"Hot spots" on sealing face are evidenced by localized annealing of ring material. Cooler portions of the sealing face were not annealed.
Figure 14. - Surface profile traces of mating seal members.

Figure 15. - Seal lubrication models.