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DYNAMIC SEALING PRINCIPLES

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ABSTRACT

The fundamental principles governing dynamic sealing operation are discussed. Different seals are described in terms of these principles. Despite the large variety of detailed construction, there appear to be some basic principles, or combinations of basic principles, by which all seals function. They are:

1. Selection and control of seal geometry - low friction packing materials, fixed and floating bushing seals, labyrinth seals, stepped face seals and mechanical face seals, lip seals, circumferential shaft riding seals, hydrodynamic seals.

2. Control of leakage fluid properties - liquid buffer seals, controlled heating and cooling seals.

3. Control of forces acting on leakage fluids - slinger seals, magnetic seals, ferromagnetic seals, viscoseals.

Theoretical and practical considerations in the application of these principles are discussed. Advantages, disadvantages, limitations, and application examples of various conventional and special seals are presented. Fundamental equations governing liquid and gas flows in thin film seals, which enable leakage calculations to be made, are also presented. Concept of flow functions, application of Reynolds lubrication equation, and nonlubrication equation flows, friction and wear; and seal lubrication regimes are explained.

NOMENCLATURE

A	cross sectional area, cm^2; $in.^2$
C	conversion constant
C_D	flow discharge coefficient, dimensionless
D	hydraulic diameter, 2h; cm; in.
G	mass flow per unit area, kg/sec-cm^2; lbm/sec-$in.^2$
F	net seal face load, N; lb_f
$\overline{F}$	pressure profile load factor, dimensionless

$\overline{f}$	mean Fanning friction factor, dimensionless
h	film thickness (gap), cm; in.
h_{char}	characteristic film thickness, $\left(h_1^2 h_2^2/h_m\right)^{1/3}$
L	flow length from entrance to exit, cm; in.
$\dot{M}$	mass flow rate, kg/min; lbm/min
m	molecular weight of gas
N	speed, rpm
P	static pressure, N/m^2; psi
Q	volume leakage flow rate, scms; scfm
$\mathbb{R}$	universal gas constant, N-m/mole K; 1545.4 ft-lbf/(lb-mole)(^{o}R)
$\mathcal{R}$	gas constant, $\mathbb{R}/m$, J/kg-K; in.-lbf/(lbm)(^{o}R)
R_1	seal inner radius, cm; in.
R_2	seal outer radius, cm; in.
Re	leakage flow Reynolds number, $\rho Uh/\mu$, dimensionless
T	temperature, K; ^{o}F
U	leakage flow reference velocity, m/sec; ft/sec
u	velocity in leakage flow direction, m/sec; ft/sec
W	flow width, cm; in.
Y	compressibility expansion function, dimensionless
α	flow coefficient, dimensionless
β	linear sealing face deformation angle, rad
γ	recovery factor, dimensionless
δ	geometric balance ratio or modulus, dimensionless
η	friction coefficient, dimensionless
μ	absolute viscosity, $N\text{-sec}/m^2$; lbf-sec/in.^{2}

ρ density, kg/m^3; $lbf\text{-}sec^2/in.^4$

τ shear stress, N/m^2; $lbf/in.^2$

ϕ flow function, dimensionless

Subscripts:

a ambient

i inner

m mean

o outer

s sealed

w wetted

1 entrance condition

2 exit condition

INTRODUCTION

In recent years, fluid sealing has become a very important area of technology due to a number of factors. These include ecological constraints, the necessity for having equipment that operates economically, and also new demands on seals due to higher pressures, temperatures, and speeds in rotating machinery. In the area of ecology, new legislation has resulted in requirements that seals have low or no leakage. Generally better seals are required than are currently used. During the oil embargo of the winter of 1974 most of us came to realize that fuel economy and energy conservation are of the utmost importance. Improved fluid sealing contributes greatly to economical operation. In many new areas of technology there are identifiable seal problems. An example is the Wankel rotary combustion engine. One of its major limitations is the seals. Periodically claims are made that the sealing problem is solved but in time it is seen that it really has not been satisfactorily resolved. Generally in all areas of rotating machinery there is a trend to operate the equipment faster and at higher pressures and temperatures to get better thermodynamic efficiency. This puts a more severe requirement on the sealing technology.

In many undergraduate machine design courses, bearings are designated as X's on the shaft and seals may not be identified at all. It was suggested that a specialist be consulted. The design engineer must consult with seal vendors, but it is essential to have an understanding of the basic principles so a judicious choice may be made. It is very important to have the right seal design during the initial concept stages of the design. Traditionally space for the seal is allotted only after the overall design is completed.

Many types of seals exist. In Table I two general categories of seals are shown - static and dynamic. Static seals, such as O-ring seals, metallic diaphragm type seals, and gaskets are very important and widely used. However, this presentation will only be concerned with dynamic seals. Dynamic seals can be categorized by their motions. A further restriction will be that only rotary dynamic seals will be discussed. There are also oscilatory seals and reciprocating seals and even limited motion seals. Many rotary seals concepts are also applicable, in most cases, to the other motion seals. Emphasis will be on overviewing dynamic seals, classifying the various types of dynamic seals, and selecting seals for different applications. The fundamentals of operation of these seals and a limited number of examples of seal systems will also be presented. A very important part of sealing is the seal system itself. That is, seal performance depends to a great extent on the environment around the seal.

Table II illustrates the various disciplines involved in fluid sealing. Note that these areas encompass the area of tribology. Fluid mechanics, heat transfer, solid mechanics and dynamics are important in many applications. In materials science both metallurgy and the chemistry and physics of the material - both the bulk and surface properties of the material - are extremely important disciplines. Fluid sealing is differentiated from many other areas of engineering technology in that seals operate in a microworld. Rubbing contact seals operate with effective gaps on the order of 50 microinches. Extremely small deformations that are not important in many other areas of engineering are extremely important in fluid sealing. Microdeformations must be carefully controlled for successful seal operation.

Many factors must be considered in seal selection and design. Some of these factors are shown in Table III. Many of these factors are common to any type of engineering hardware that must be designed. In sealing probably the foremost requirement is "what is the allowable leakage rate?" The allowable leakage rate depends on the application. The requirement may require a zero leak seal. However, in reality we never have a truly zero leak seal situation. There may appear to be no leakage across the boundary. However, there can be diffusion across the boundary; vapor leaking out of a liquid gas interface or amounts of the fluid itself leaking. For example, a rubber diaphragm static seal will actually allow diffusion of water through the diaphragm. If you are sealing hydrogen even with a solid material, you get diffusion of the hydrogen through the solid material. Many water pump seals appear to be operating as zero leak seals. In reality, however, it is very possible that one does not see any droplets of liquid but in this seal vaporization occurs at the interface. This is also the case with helicopter transmission seals. Experiments at NASA have shown that although liquid oil does not leak out of the seal, there is hydrocarbon vapor leaking out of the interface. At the other end of the leakage spectrum a labyrinth seal typically may leak a pound of the sealed substance per second. This may be an acceptable leakage rate for some applications. Thus, one of the foremost requirements is to determine the allowable leakage rate. Another requirement is the available physical space for the seal and the seal duty requirements - seal pressure differential, seal temperature and temperature gradient, the rotating speed, and surface velocity. The sealed fluid media must also be considered: Does it have good boundary lubrication properties? Is it abrasive in nature? Is it toxic? Another factor is the maintainability and life requirements of the seal - is the seal life to be the one or two year warranty period or five years (some power applications desire a seal life of 40 000 hrs)? Some rocket seals only have to operate for 30 seconds. Another important consideration is the necessity for external accessories. Is there room and does cost allow an auxiliary cooling system to be applied? Can a buffer system with its complex controls be used? Another item is the wear and/or rubbing characteristics of the seal materials. Will the wear rate be low enough for the design life of the seal. Are the rubbing seal surface materials compatible (is the friction low)? Will thermoelastic instability (a situation where catastrophic failure occurs at the sealing interface) be avoided? Control of all types of distortions is important. Distortions may be due to centrifugal forces, axial temperature gradients, pressure, and/or mechanical forces. Excessive distortions must be avoided or a seal has to be chosen that can accommodate these distortions. Here a decision on initial cost against the total life cycle cost must be made. This may be a very important consideration because in some applications seals can significantly increase the cost of the equipment. In some cases a trade-off must be made of cost with reliability and life of the seal. The lower initial cost may mean problems in the field and high replacement costs. All of the factors mentioned are important but the particular application usually determines the relative importance of each.

DYNAMIC SEAL CLASSIFICATION

Dynamic seals exist in many configurations and sizes and can be classified in many ways. However, operation can be described in terms of a few fundamental principles. Differen' seals will be described in terms of these basic approaches. Despite the large variety of detail construction there appear to be some basic principles or combinations of basic principles by which all seals function. (These are shown in table IV.) They are:

(1) Seals that depend on the selection and control of the sealing geometry. These can be further reduced into three subcategories:

(a) Positive rubbing contact seals - these include mechanical face seals, circumferential shaft riding seals, lip seals, and shaft packings.

(b) Seals that operate at close clearances. Close clearance seals generally operate with clearances on the order of 0.025 to 0.25 centimeter (0.1 to 1 mil). In this category are hydrodynamic seals, hydrostatic seals, and floating bushing seals. Hydrostatic seals further can be classified in terms of the two operating modes - externally pressurized mode or self-energized.

(c) Fixed geometry clearance seals. The seal is the gap between the shaft and stationary sleeve. The gap must be large enough to accommodate shaft distortions and dynamics. In this category the clearance depends on the size of the shaft - generally the clearances can be on the order of 0.001 centimeter per 1 centimeter (1 mil per 1 in.) radius of the shaft. Basically there are two types of fixed geometry clearance seals - fixed bushing seals and labyrinth seals.

(2) Seals that depend on control of fluid properties. These include controlled heating and cooling seals and ferromagnetic seals.

(3) Seals that depend on control of fluid forces. Three types of seals fail into this category.

(a) Centrifugal

(b) Screw pump or viscoseals

(c) Magnetic seals

SELECTION AND CONTROL OF GEOMETRY

As previously mentioned, there are three classes of seals in this category - positive contact, close clearance, and fixed geometry. The seal leakage equations can be found in appendix A and are discussed in more detail in reference 1.

Positive Rubbing Contact

Various rubbing contact type seals are shown in figure 1. For very light duty applications "O" rings, molded packings, and compression packings can be used.

Mechanical face seal. - A photograph of a mechanical end face seal is shown in figure 2. This is a very widely used seal in many applications, such as in the processing industries. Figure 3 illustrates the basic elements of the mechanical face seal. A rotating seal seat is mounted to the shaft and held at close proximity to a nonrotating sealing ring. The sealing ring is held in close proximity to the seat by a mechanical spring. The sealing ring is allowed to move axially to accommodate the axial motion of the seal seat - such as runout. Antirotation lugs prevent seal ring rotation (not shown in fig. 3). Relative motion of the ring and stationary housing occurs across a secondary seal.

Figure 3 shows the secondary seal as an "O" ring. This seal also can be a piston ring for higher temperature applications. The purpose of the secondary seal is to allow the seal ring to track axial motions of the seal seat. It is practically impossible to locate the seal seat to be perfectly perpendicular to the axis of rotation, hence axial runout occurs. In order to accommodate this runout for low leakage, small axial movement of the seal ring is allowed. That is, the spring essentially forces the seal ring against the rotating seal seat to accommodate the runout or wobble.

The particular configuration shown in figure 3 is an internally pressurized seal. That is, the pressurized fluid is located on the inner diameter of the seal and is being sealed from the ambient pressure located at the outside diameter (preventing the sealed liquid from leaking to this environment). The seal shown is a pressure balanced seal. At the primary seal interface leakage occurs and a pressure drop of the fluid exists. For laminar viscous flow (a very common sealing situation) the pressure profile is linear as shown. The force due to this pressure drop is called the seal separating or opening force. Holding the seal against the seal seat are the spring force and a hydrostatic closing force (a net seal pressure force) that acts against the seal ring as shown in figure 3. A common practice is to balance the hydrostatic pressure force with the pressure drop force and let the spring force apply a slight contacting pressure. For liquids this contacting force is generally 27 to 53 newtons per centimeter (6 to 12 lb/in.) circumference and for gases it is less than 3/4 pound per inch. There are other forces involved. The inertia force of the sealing ring becomes very important at high speeds. If the secondary seal is an elastomer or a piston ring type, a friction force exists at the secondary sealing interface. One of the fundamental problems is to decide on what the secondary seal diameter should be. For a face seal this diameter is usually determined by practical experience as well as analytic consideration of the force balance. An important parameter is the pressure profile load factor. See appendix B. For liquids theoretically parallel surfaces predict a pressure profile load factor of about 0.5 (ref. 1); however, in practice there are inherent distortions and a factor of about 0.6 is typical. However, it will vary from application to application. A seal vendor can be of great assistance in selecting the proper face seal for an application where a mechanical face seal is the proper choice.

For very light duty applications - low speed, low pressure - the seal may be completely pressure loaded. That is, the seal pressure acts over the entire sealing interface (see fig. 4(a)). On the other hand, an application could require a seal that is pressure unloaded - that is, the separating force would be greater than the closing force (fig. 4(b)). Pressure balancing of face seals is of fundamental importance. In reference 1 there is a more detailed discussion of pressure balancing.

Figure 5 illustrates a common application of a mechanical face seal as an end face seal in a centrifugal pump. These seals are located at both ends of the pump shaft. Figure 6 shows more details of an end face seal. Note the seal ring is rotating here - that is, the spring acts on the rotating piece as shown in figure 6.

A rotating sealing ring can accommodate shaft center-to-bore center offset or dynamic shaft whip during operation (due to the gyromoment an eccentric inward pumping component does not exist). The nonrotating sealing ring main advantage lies in its ability to accommodate shaft center-to-bore center misalinement. Also a nonrotating sealing ring must be used at high shaft speeds.

The secondary seal and spring function can be combined into one integral unit as illustrated in figure 7 where a bellows is used. However, the use of bellows is limited due to collapse of the bellows fingers when higher pressures are sealed. The mean effective diameter of the bellows changes; the mean effective balance diameter is analogous to the secondary seal diameter. Figure 7 also illustrates a mechanical seal system in terms of a lumped parameter dynamic system. A simple one-dimensional model is shown where the seal ring is represented as a lumped mass. The bellows has both stiffness and damping properties. (In low viscosity sealing media, the bellows must have a finger-type Coulomb damper because the natural frequency of the bellows can be very low in the range of the operating speed.) The fluid film also has stiffness and damping properties and these properties are modeled as shown in figure 7. The seal seat can be represented as a massless displacement forcing function. An analysis of the dynamic behavior of fluid film seals using these analogies can be found in reference 3.

There are many variations in mechanical face seals. There are, of course, advantages and disadvantages to a particular design. Despite hardware differences the pressure balancing principle is of paramount importance.

A widely used criterion for determining the limitation of seal-face materials is the PV factor, the product of the unit pressure acting on the sealing interface (P.S.I.) and the rubbing velocity (fpm). For any given combination of seal face materials, the limiting PV value depends on such factors as surface, quality, lubricating ability of seal fluid, rate of heat conduction from the sliding interface, etc. There are PV limiting values for heat generation and wear (1). Caution should be exercised in interpreting this limitation because of such factors as local stress levels greatly exceeding the unit pressure. Conditions used to obtain PV data should be carefully examined.

The composition of the sealed liquid can vary in the sealing interface; even boiling can occur. Figure 8 illustrates the sealing interface of an oil seal from a study by Orcutt (ref. 2). The seal ring was a transparent optical flat. Hence the interface was visually observed and the temperature at the interface was measured using an infrared temperature measuring technique. Figure 8 from reference 2 shows that the oil film only extends out a certain radial distance then the oil vaporizes and oil vapor leaks out of the seal interface. Figure 8 also shows the measured temperature profile and shows that the seal was operating at a relatively high temperature. This was due, in part, to the high bulk oil temperature and also due to shear heating.

The maximum temperature was about 500° F at the vapor-liquid interface where boiling was occurring. High temperature and/or high speed liquid seals must be cooled due to the high shear heating that occurs at the sealing interface. These seals will be speed limited due to the shear heating that causes the sealing interfaces to distort.

Mechanical face seals are discussed in much greater detail in reference 7.

Circumferential shaft seal. - Another positive contact seal is the circumferential seal shown in figure 9. The seal ring is usually segmented into three segments in order to accommodate radial misalinement of the shaft and also the dynamic motion of the shaft and still maintain a very small gap. Also, as shown in figure 9, there is a retainer cover which prevents leakage at the segment joints and a garter spring which keeps the segmented seal rings in close proximity to the seal shaft. The seal shown in figure 9 is completely unbalanced. However, there is a balanced version as shown in figure 10 where pressure relief slots which result in only a small net unbalance force both in the axial and the radial direction. It should be noted that circumferential seals can never be perfectly balanced due to the configuration. A face seal can be, at least in theory, completely pressure balanced. Also lapping a flat surface is much easier to perform in the face seal geometry than it is for shaft riding type seals characterized by curved surfaces. Also, circumferential seals can be very sensitive to installation - fracture of the carbon segments can occur during blind assembly.

The circumferential seal is commonly used when the shaft undergoes large axial movement. These seals are usually more tolerant of pressure reversals than face seals. Another advantage of this seal can be in sealing in an oil environment where coking can occur. The circumferential seals can still seal even though the sealing segments are frozen. However, the gap will be larger and resulting leakage higher. A face seal may catastrophically fail when coking occurs in the interface.

Lip seal. - A very common seal that is used in automotive applications and appliances is a lip seal, shown in figure 11. Some of the elements of this seal are the case, the stiffener ring and the primary lip, which is held in close proximity to the shaft by a garter spring force (usually on the order of 1.24 N/cm (0.7 lb/in.) of circumference). These seals are usually used for low or zero pressure differentials - for example, to prevent an oil mist that is lubricating the bearing from leaking to the outside environment. However, in some applications they have been used for shaft speeds to 3 kilometers per minute (10 000 ft/min) and have sealed 69 newtons per square centimeter (100 psi) pressure differentials. Lip seals are inexpensive, compact, and easy to install. However, since lip seals are made of elastomeric materials, two major problems can be encountered: (1) chemical compatibility of the elastomer with the sealed fluid; the seals usually swell upon contact with the sealed liquid. In cases where the interaction is incompatible, destruction of the sealing interface can occur; (2) stress relaxation - elastomers have the problem that due to hysteresis frictional energy, the internal molecular chains can fracture and, in time, especially at the higher temperatures; the elastomer may age harden and the sealing effectiveness can be lost. This is especially true in the high speed applications where sealing interface temperatures as high as 478 K (400° F) have been measured due to the heat generation.

In analyzing lip seals elastohydrodynamic theory is used; however, the pressure-viscosity effect is neglected. In order to find the film thickness distribution, an elastic analysis is first performed, then a hydrodynamic fluid film analysis using Reynolds lubrication equation. Wiebull statistics are used in predicting the lives of lip seals. The industry now is trying to set up standards on accelerated life tests. However, this approach has been generally unsuccessful to date.

There are many innovations for lip seals to try to improve performance and life. Placing helical grooves on the interface has been successful in some applications. Recently a wavy type of lip seal has been introduced and appears to increase the life of the lip seal. The "wavy" sealing surface results in a sealing interface that is more effectively cooled than a standard lip seal where continuous surface line contact occurs. Generally a molded lip will give better performance than a trimmed lip. Reference 4 has more information on lip seals.

Soft packing seals. - Another very common seal is the soft packing; it is a relatively inexpensive, simple, sealing material and has been used for many years. A common type of soft packing is compression packing shown in figure 12. Wrapped abraded pliable material is compressed by tightening of bolts in a stuffing box. This compression yields a low leakage sealing interface. A very common material is asbestos. Because of potential health problems, it is not used as commonly today as in the past. Elastomers and metal foils are used instead. Many of the soft packings are impregnated with solid lubricants. The packing can also be lubricated with a grease tap. The lantern ring in figure 12 acts as a distribution manifold for the grease. Generally compression packings are limited to peripheral speeds of 50 meters per second (10 000 ft/min) but sealed pressures as high as 345 newtons per square centimeter (500 psi) have been achieved with large axial lengths. Fluid temperatures to 808 K (1000° F) have been sealed. Unfortunately, the packing material does wear away due to high heat generation and periodic adjustment of the compressive force is required to keep the packing in close proximity to the shaft. Other types of packings that are used are automatic packings, cup-type, and floating packings which are similar to piston rings. Recently, it appeared, due to the recent ecological requirements for low leakage seals, that packing seals would lose a large share of the market. However, recent graphite metal foils have been introduced which are sealing effectively and can be used in the small space required by the packing. So at a time when the future of packings was seriously being treatened a new innovation, in this case a material, came along which appears to have (at least temporarily) solved the problem. It appears that packings will always be used for certain applications. In applications where packings are replaced with lower leakage seals such as mechanical face seals, a great deal of redesign is usually required due to the larger spatial requirements.

Close Clearance Seals

Hydrodynamic seals. - For applications where positive contact seals are not practical, close clearance seals may be used. One type is the hydrodynamic seal shown in figure 13. This hydrodynamic seal is primarily used to seal gases. Essentially the sealing ring interface is the same as an ordinary mechanical face seal. However, a fluid film bearing geometry has been added to the interface in order to give positive separation of the surfaces. In figure 13 the bearing geometry is a shrouded Rayleigh step gas bearing called self-acting lift pads. These lift pads have pockets on the order of 0.00127 to 0.00254 centimeter (0.5 to 1 mil) deep and pocket-to-land width ratios in the circumferential direction about 2:1. Axial and radial grooves around each pad keep the pressure the same around the pad.

During rotation of the seat, the high pressure gas is dragged into the pad and is compressed as it passes over the step at the end of the pad. This creates a lifting action or force separating the primary seal ring and the rotating seat.

The pressure drop in leakage occurs across what is known as the sealing dam of the sealing ring. The fluid film bearing also contributes a high film stiffness to the seal (190 000 to 950 000 N/cm (100 000 to 500 000 lbf/in.)) such that the seal ring can dynamically track the motions of the seal seat. This is especially important in high speed applications where the runout can be excessive and the unbalance forces induced could not be tolerated without the fluid film geometry seal surface. Another hydrodynamic seal is a spiral groove seal shown in figure 14. Spiral grooves are incorporated in the sealing interface and operation is similar to the lift pad seal. However, with a wide radial face a "pumping action" will make this seal very efficient and zero net leakage operation can be achieved. This seal has been successfully used in sealing liquids. Consult references 5 and 8 for further information about both types of seals.

Externally pressurized hydrostatic seals. - Another type of close clearance seal is an externally pressurized hydrostatic seal, shown in figure 15. A buffer fluid is pressurized to a higher pressure than the sealed fluid. In effect, the buffer fluid leaks against the leakage of the sealed fluid. This type of seal requires additional plumbing and controls. Usually the pressurized fluid is at least 3.5 newtons per square centimeter (5 psi) higher than the sealed fluid. Under all conditions of operation the buffer pressure must be higher than the sealed pressure. An example of its use is applications where abrasives are present in the sealed fluid. In effect the buffer fluid "flushes out" the sealed fluid so that the abrasives will never destroy the sealing interface. This type of seal is also used in applications where toxic fluids are sealed. The buffer fluid has to be compatible with the sealed fluid. If it is not, a more complex seal system is required whereby the entrained buffer fluid must be separated from the sealed fluid in order to prevent the buffer fluid from contaminating the process fluid.

Self-energized hydrostatic seals. - Self-energized hydrostatic seals are also used as close clearance seals. A radial step seal is shown in figure 16. A shallow step on the order of 0.00127 to 0.00254 centimeter (0.5 to 1 mil) deep is located over part of the radial distance of the sealing ring. An ordinary sealing interface (dam) exists over the remainder of the length. This type of seal has an equilibrium restoring system similar to a servo system. This restoring force system can be better understood by examining figure 16. In case A, at a normal design gap the seal separating force is due to a pressure drop across the recess region and one across the sealing dam region. Also, acting on the sealing ring is the seal closing (hydrostatic pressure) force as shown in the figure. Now to understand how this servo principle operates, examine case B where a very close gap exists across the sealing dam. In that case very little pressure or no pressure drop takes place across the recess. The entire pressure drop is across the dam portion. In this case the seal ring forces are unbalanced such that the seal opening force is greater than the closing force hence we get a restoration opening to case A. On the other hand, if the gap is very large, the pressure drop would be as shown in case C. Here the presence of the step does not appreciably affect the pressure drop across the sealing interface. Essentially a linear pressure drop occurs across the entire seal interface. In this case the closing force would be greater than the seal separating force and this closing force would give a restoration to the equilibrium condition in case A. This seal is used successfully; however, hydrostatic seals are susceptible to self-excited instabilities similar to those that are observed in hydrostatic bearings. This is particularly the case when sealing gases. Generally hydrostatic seals are used in very high pressure differential applications. The effect of rotation can usually be neglected in these cases (rotative speeds may be too low to yield sufficient hydrodynamic lift forces and the gap is too large for significant hydrodynamic pressure generation). Again, a finite amount of leakage must be tolerated. Also a jacking gas may be used to achieve surface separation prior to rotation.

Floating bushing seal. - The third type of close clearance seal is the floating bushing seal shown in figure 17. Here a bushing acts as a flow restrictor and the bushing is kept in close proximity to the seal shaft by allowing the bushing to float radially. Rotation is prevented by the use of an antirotation pin as shown in figure 17. This type of seal can accommodate large shaft movements and still behave as a close clearance seal. Floating bushing seals generally operate in a laminar flow regime, and are effective for sealing liquids. Again a finite amount of leakage must be permitted when this type of seal is used. If the shaft misalinement is large, rings of seals similar to the bushing seal can be staged to accommodate the misalinement. In high temperature applications a wear ring of carbon is retained by a metal ring. The composite thermal expansion of both rings is designed to match the thermal expansion characteristics of the shaft.

Fixed Geometry Clearance Seals

Fixed bushing seal. - The last category of classifying seals by geometry is the fixed clearance seal. Two types are found in this category: The fixed bushing seal and the labyrinth seal. In this case very limited axial or radial motion can be tolerated. Fixed gap operation between bushing and the shaft occurs and again this is a flow restrictor type of operation. This seal is illustrated in figure 18. In general, the clearances are large and fixed bushing seals will usually operate in the turbulent flow regime.

In special applications much ingenuity can be used. For example, in some particular applications with the bushing seals shown in figure 19: (a) a geometry that deforms causes the clearance to reduce as pressure increased and, (b) a situation exists where the high pressure causes the seal gap to increase when the pressure is increased. Thus there is a great deal of flexibility and a lot of ingenuity can be used in designing these seals for a particular application.

Labyrinth seals. - The second category of fixed geometry seals is labyrinth seals. Labyrinth seals have been used for many years - since the early days of rotating machinery. These seals are illustrated in figure 20. Many geometric variations are possible with this type of seal. In effect this seal consists of stages of knife edges analogous to orifice or throttling stages. Common industrial practice is to have the knife edges located on the stator. If rubbing occurs, the knife edges will expand away from the shaft. Also the knife edges will wear away because the rotor usually has a very hard coating to minimize the wear. In aircraft applications, due to fatigue and aeroelastic problems, the knife edges are located on the rotor. Here the stator usually contains a rub tolerant material. Abradable sprays, honeycomb, porous surface materials, and soft plated surfaces are commonly used. Figure 21(a) shows a straight through type labyrinth, (b) shows a double surface or closed type labyrinth which is not too commonly used. Another variation is shown in figure 21(c). Figure 21(d) illustrates a canted knife edge labyrinth which is more effective in sealing than the straight through labyrinth. On figure 20(c) a staggered labyrinth design is shown; the purpose of the steps is to change the flow kinetic energy into frictional energy. This is the key physical mechanism of labyrinth seal operation. In effect the kinetic energy of the flow through the labyrinth is converted to frictional energy which in turn is dissipated as heat. The total pressure is not recovered. Figure 20(b) shows a more effective way of dissipating the kinetic energy because of the high turning loss associated with that type of labyrinth configuration. However, this configuration is limited to certain industrial applications where split casings can be used because of the assembly problem. Figure 22 illustrates a very commonly used labyrinth seal in aircraft applications where two stages of labyrinths of four knife edges are used as inner air seals.

The labyrinth seal leakage equations generally can be derived from inviscid flow theory and will now be described.

The inviscid flow conservation of momentum equations for one-dimensional, incompressible flow is

$$-A\ dP = \dot{M}\ du$$

$$= \rho u A\ du$$

Integrating between any two control surfaces points 1 and 2 along a stream tube

$$-\int_1^2 dP = \rho \int_1^2 u\ du$$

or

$$P_1 + \frac{\rho u_1^2}{2} = P_2 + \frac{\rho u_2^2}{2}$$

This is the classical incompressible Bernoulli equation. For flows where the entrance velocity is negligible, thus

$$u_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho}}$$

The mass flow is thus

$$\dot{M} = A\rho_1 \sqrt{\frac{2(P_1 - P_2)}{\rho}}$$

For a perfect gas, the above equation becomes

$$\mathring{M} = \frac{C_D A P_1}{\sqrt{R T_1}} \left[\sqrt{2\left(1 - \frac{P_2}{P_1}\right)}Y\right]$$

where Y is a compressibility expansion function or

$$\frac{\dot{M}\sqrt{T_1}}{AP_1} = Y\sqrt{2\left(1 - \frac{P_2}{P_1}\right)}\frac{C_D}{\sqrt{R}}$$

We define the flow function as

$$\phi = \frac{\mathring{M}\sqrt{T_s}}{AP_s}$$

The flow function is a function of the pressure ratio and the particular overall geometric configuration. The pressurized fluid conditions are now designated by the subscript "s." The flow function is a measure of the flow effectiveness.

The mass flow equation can be rewritten in the following form:

$$\dot{M} = C\phi\alpha\gamma\frac{AP_s}{\sqrt{T_s}}$$

where

$\dot{M}$ mass flow rate, lbm/sec

ϕ flow function

α C_D, the discharge coefficient or flow coefficient that accounts for real flow effects such as viscous friction, vena contacts, etc.

γ recovery factor, used for straight through staged restrictions, e.g., in stages of orifices or labyrinth knife edges

A flow cross sectional area, in.^{2}

C conversion constant, e.g., C = 0.777 for units in lbf, lbm, in., sec system

P_s sealed gas pressure, psia

T_s sealed gas temperature, ^oR

The resulting equation is named after Egli and was first presented in 1935 (ref. 6). Three factors are found in Egli's equation that can be obtained from graphical plots shown in figure 23. The first is the flow function ϕ and it depends on the pressure ratio and the number of knife edge restrictors. α is the flow coefficient; it is analogous to the discharge coefficient in orifices and, in fact, is the flow area deficiency or discharge coefficient. The flow coefficient is a function of the gap and the thickness of the labyrinth knife edge. Finally γ is the carryover factor, and it depends on the gap-to-knife edge spacing ratio. The carryover factor has a nonunity value in straight-through labyrinth configurations. In effect, this accounts for the carryover of the kinetic energy that is not dissipated in a straight through labyrinth. If the labyrinth knife edges are spaced too close to one another, the kinetic energy is not completely dissipated, and kinetic energy from the previous knife edge flow is "carried over" and results in higher leakage. If the knife edges are placed too close to one another, a double set of knife edges would only be effectively acting as a single knife edge. There are a lot of optimization and trade-off studies that can be made with labyrinth seals. Many organizations have their own proprietary data on labyrinth seals. Egli's equation gives one an engineering estimate of the leakage. More sophisticated corrections include pressure ratio, Reynolds number, surface porosity, and other geometry corrections. Additional information on labyrinth seals can be found in reference 6.

CONTROL OF FLUID PROPERTIES

Freeze Seal

The second major category of classifying seals is to control the sealing forces. One type of seal in this class is a freeze seal shown schematically in figure 24. (Although the seal in fig. 24 is a limited motion seal, the same principle applies to rotary seals.) This seal has been used primarily by the Atomic Energy Commission for sealing valve stems and has successfully sealed sodium, fluorine, and lead. Basically, the liquid metal is solidified in the annulus around the shaft and acts as the seal. In operation frictional heat causes a thin fluid film to develop between rotating shaft and the annulus of the solidified material. Properly designed, the freeze seal will have a starting power no greater than a packing seal and the running power will be less than a typical packing used in the stuffing box design. A typical gap is 30 mils. This gap is small enough to prevent extrusion of a solid sodium plug up to a pressure differential of 15 psi, and large enough to prevent the formation of a strong bond between the shaft and the frozen material. This type of seal has been used to temperatures of 1200° F, sealing flow rates of 1400 gallons per minute and a sealing head to 100 feet. Among the seal's disadvantages is that this type of seal will leak if the auxiliary coolant system fails. High wear occurs if abrasives are present or the fluid precipitates. Other details of using this seal and a typical temperature profile are shown in figure 24. This seal is described in reference 9.

Ferromagnetic Seal

Another seal that has been recently developed and has created a great deal of interest is the ferromagnetic seal illustrated in figure 25. The elements of this seal are knife edges on the shaft and magnets and pole blocks located on the stationary housing. A magnetic circuit through the seal is induced as shown in figure 25. Colloidal suspensions of ferrite particles are dispersed in the sealed fluid medium. These suspended particles are in the colloidal scale, sizewise. The particles move with the fluid's Brownian motion - that is, thermal agitation is occurring and the particles do not tend to separate or coagulate. Coalescing is further prevented by the use of antidispersion coatings which are claimed to be a proprietary secret. Any barrier fluid can be used and the limitations may be more on the bulk fluid than on the ferritic particles. Generally the limitation is the vapor pressure and the magnetic saturation temperature of the barrier fluid. One cited advantage of this seal, as shown in figure 25, is that operation with a fairly large gap and fixed geometry with low leakage is possible. It is too early to appraise what the full impact of this seal will be. However, it has been very successful to date in applications where a good dynamic seal was not available. It has become popular as a rotary vacuum seal. For example, an electric motor is located in an outside ambient environment and its output shaft drives an element that is inside a vacuum environment. Among the limitations of this seal are fluid degradation due to the high shear layer that occurs at high speeds. (The ferromagnetic fluid is essentially stationary except for a very thin boundary layer at the rotating surface where the slip (shearing) occurs.) Also, interfacial instability and other instabilities may be a problem. Especially undesirable is the application case where it is not desired to have the barrier fluid mix with sealed liquid. Further, the ferromagnetic particles must be replenished periodically. A problem with ferrite particle agglomeration has been observed, particularly with water. The particles settle out. The start-up performance of shaft seals may be poor when the seal system has been idle for extended periods. In time we may see many new applications of this sealing concept.

CONTROL OF FLUID FORCES

Slinger Seal

The third major category of seal classification is seals that control the forces. The first group utilizes the centrifugal force due to rotation. Such a seal is the slinger or rotating fluid ring seal shown in figure 26. This seal is comprised of a rotating disk enclosed in a confined housing. The centrifugal force acts on the liquid and a rotating head of fluid results. If the pressure on one side of the disk is increased, the pressure will force the interface up and a liquid seal results that operates analogous to a manometer. This type of seal has been used in space power applications. However, it is limited by the frictional heat that is generated, the maximum sealed pressure differential, and the stability of the interface between the liquid and the gas. There are many variations of this type of centrifugal seal. Some are shown in figure 27.

Viscoseal

Another seal in this category is a viscoseal shown in figure 28. It is essentially a fixed geometry seal. It usually operates at gaps of 0.0058 to 0.0127 centimeter (2 to 5 mil). This seal is an axial seal where the pressurized liquid, as it is leaking axially down the shaft, is pumped back to the high pressure end by the action of helical grooves similar to screw threads. In fact, it operates as a screw pump. The shearing action of the grooves pump the leaking fluid back to the high pressure end. Its performance depends on the shaft speed, axial pressure gradient, fluid viscosity, and seal gap. If sufficient axial distance is available a "zero leak" seal appears possible. However, diffusion from the interface occurs. This seal successfully operates in the laminar flow regime. A problem is encountered at high speed operation (turbulent flow regime) where the interface between the liquid and the gas becomes unstable and gas or air of the ambient environment is entrained.

Magnetic Seal

The final example of controlling seal forces is the magnetic seal (fig. 29) where the elements are the same as the mechanical face seal. However, the seal spring is force replaced by a magnetic force. The magnetic seal consists of a magnetized ring with an optically flat sealing surface attached to the housing and a rotating sealing ring fabricated from a magnetic stainless steel and is movable axially along the shaft. The advantage of this seal is that a constant closing force acts regardless of the axial seal movement. In some applications where large axial excursions occur a mechanical spring force may be too large and the magnetic seal may overcome this problem. The experience with this seal is limited.

SEAL OPERATING REGIMES

Seals operate in many lubrication regimes depending on the type of seal, sealed fluid, application, etc. It is useful to use a friction coefficient against seal duty parameter plot to understand the various seal operating regimes that can exist.

For illustrative purposes consider a lift-off type seal that is in rubbing contact at start-up (and shutdown). The way the seal may change from one lubricating regime to another in an application can be illustrated by considering figure 30. The figure shows the friction coefficient variation a seal undergoes from startup under a load (e.g., due to spring force and pressure) in the boundary lubricating regime to the steady state operating speed in the full-film lubricating regime. (The mechanism for achieving full-film operation could be due to an external pressurization source or self-generated by hydrodynamic lubrication.) As the seal starts, the parts are in solid-to-solid rubbing contact and the seal seat (rotating member) begins to turn under essentially dry conditions and starts to follow the path AB. If sufficient lubricant is available, the lubricant is ordinarily drawn between the sliding surfaces at once (by capillary action or a forced pumping mechanism) and the seal immediately enters the mixed-film or thin film region following the path BC. When the speed reaches the value corresponding to point D, the seal enters the full-film laminar flow regime in which it remains until coming to operating speed at point E. At point F, the friction would again increase due to operation in the turbulent flow regime. It can be seen in figure 30 that if the lubricant were not present the seal would be forced to operate dry at a speed corresponding to point A, the resulting temperature rise could be extremely high, due to the high friction.

Now some of the details will be examined more closely. Figure 30 shows two distinct lubricating regimes - boundary lubrication and full film (hydrodynamic). These curves were originally proposed for journal bearings but the same principles apply for a seal. Friction coefficient and film thickness are plotted against a seal duty parameter $\mu N/F$, where μ is the fluid viscosity, N is the rotational speed, and F is the net seal face load. To the right of the dashed vertical line is the region of full film fluid lubrication; that is, thick film lubrication, where the surface asperities are completely separated by an oil film of such thickness that no metal-to-metal contact can occur (see fig. 30). Hydrodynamic lubrication theory applies and the flow is laminar. At sufficiently large values of the seal duty parameter, turbulent flow can occur (transition occurs at E and turbulent flow exists in region F). The friction here rises significantly and increases at a more rapid rate with speed than in the laminar flow regime. To the left of the dashed vertical line is the region of boundary or thin-film lubrication. As noted in figure 30, the film thickness in boundary lubrication is so small that asperities contact through the oil film. The mixed film regime can be identified as the one that has partial hydrodynamic and boundary lubrication. This is also the regime where elastohydrodynamic effects may be important. In full fluid film lubrication, since the asperities do not contact, only bulk lubricant properties are important. In boundary thin film lubrication, the bulk properties of the surfaces and surface physics and chemistry are of primary importance since there is solid-to-solid contact by asperities. Lubricant chemical properties can influence the type of damage that occurs.

In summary the lubrication regimes can similarly be associated with the seal film thickness. The three regimes from this point of view are shown in figure 30. That is, the full film lubrication regime is characterized by the film thickness being several times greater than the surface roughness. The mixed film regime has the film thickness on the order of the film thickness. In the boundary regime asperity contact characterizes the interface.

Note the friction coefficient for a hydrodynamic film can be calculated from

$$\eta = \frac{\tau A}{\text{Net normal closing force}}$$

where τA is the traction force.

The friction coefficient behavior is different for a gas and is shown in figure 31. Since gases are poor boundary lubricants, the friction coefficient values are almost those obtained for sliding solid-on-solid for low seal duty parameter values. Since a gas has a much lower viscosity than a liquid, friction forces can be one to three orders of magnitude less in the full film lubrication regime. For a gas seal to operate in this regime usually requires the incorporation of a lift geometry to the sealing faces. Since operating gaps are inherently smaller for gas seals (due to the low viscosity), gas film seals are more sensitive to face distortions. This, coupled with the poor boundary lubricating properties of gases, means stable self-induced hydrodynamic gas film seal operation is unlikely. The friction coefficient variation is qualitatively similar to liquids in the full film regime until compressibility effects become significant. Generally compressibility effects become important before turbulence. However, for large gaps and sufficiently high pressures or speeds turbulent flow can be achieved.

SEALING SYSTEM EXAMPLE - GAS TURBINE SUMP SEAL

A typical bearing compartment seal system that is found in gas turbine aircraft engines is illustrated in figure 32. These seals are protecting the bearing environment from the hot environment of the gas turbine engine. The sump is pressurized with air from the fan or low compressor stage air which prevents hot gas from leaking into the sump. In the example, a labyrinth seal with a honeycomb rub strip is used. A limited amount of air leaks into the sump. Note that a windback seal is shown.

A windback seal operates in a similar way to a viscoseal. It pumps out droplets of oil out of the sealing interface. In this application the seals not only prevent the hot gas from leaking into the bearing compartment avoiding problems with oil coking and potential fires but also these seals establish the bearing thrust load. The net thrust load acting on the ball bearing is determined by the radial location of the balance piston labyrinth seal. Also notice in figure 32 the oil drains from the bottom of the sump whereas the gas vents out the top to the outside environment. This is just one example of a sealing system and this example illustrates that seals can serve purposes other than strictly sealing. Seals are important for secondary flow management and controlling the thermal gradients in machinery as well as protecting and establishing thrust loads on shaft bearings.

CONCLUDING REMARKS

In conclusion many dynamic sealing principles have been presented. An attempt has been made to cite the advantages and limitations of each seal type. Some of the factors in seal selection and design have also been pointed out.

APPENDIX A

FUNDAMENTAL LEAKAGE EQUATIONS FOR LIQUIDS AND GASES

Close clearance seals can operate in either the laminar or turbulent flow regimes. Usually the flow path distance will be much greater than the sealing gap, hence viscous friction is extremely important in these seals, as contrasted to labyrinth seals where thin knife edges are used. Usually the approximation is made where the rotational effect on leakage is neglected. This may not be true for high speed liquid seal operation. See reference 1 for more discussion.

For turbulent flow exact physical knowledge is unknown. Hence an exact differential analysis model such as the one that describes laminar flow is impossible to solve or impractical for design analysis purposes. Thus approximate solutions must be found. A widely used method in fluid mechanics and hydraulics is the approximate integrated average method. Although the integral models only satisfy mean conditions in the flow field, they have shown good results on gross quantities such as seal leakage and pressure distribution. However, it will be seen that an empiricism will be required to find a solution.

The viscous friction is balanced by the pressure drop. This is the classical fluid flow case. This model is widely used to describe pipe and duct flows.

Consider the control volume shown in figure A1 for situations when the fluid inertia is negligible. The momentum conservation is a balance between the pressure and viscous friction force which is

$$A\,dP = -\tau_w\,dA_w \qquad (A.1)$$

Now introduce the following parameters

Hydraulic diameter:

$$D = \frac{4A}{\frac{dA_w}{dX}}$$

Mean Fanning friction factor:

$$\bar{f} = \frac{\tau_w}{\frac{\rho u^2}{2}}$$

into equation (A.1) this results in

$$D \frac{dP}{dX} = -2\ u^2 \bar{f} \tag{A.2}$$

Substituting the mass flow definition

$$\dot{M} = \rho u A \tag{A.3}$$

yields the following useful form

$$dP = \frac{-2\dot{M}^2 \bar{f}\ dX}{\rho D A^2} \tag{A.4}$$

The pressure at a position, X, along the leakage length can be found by integrating from the inlet to any position X

$$\int_{P_1}^{P} dP = -\int_{0}^{X} \frac{2\dot{M}^2 \bar{f}}{\rho D A^2}\, dX \tag{A.5}$$

Constant Area Flows (Parallel Flows)

Equation (A5) can be readily integrated for constant flow area.

$$P = P_1 - \frac{2\dot{M}^2 \bar{f} X}{\rho D A^2} \tag{A.6}$$

at the seal exit $X = L$, $P = P_2$, hence the mass flow rate can be found from

$$\dot{M} = \sqrt{\frac{\rho D A^2 (P_1 - P_2)}{2 \bar{f} L}} \tag{A.7}$$

Substituting this equation (A.7) into equation (A.6) results in a linear pressure drop equation that is

$$P = P_1 - (P_1 - P_2)\,\frac{X}{L} \tag{A.8}$$

Hence for either laminar or turbulent flow the pressure distribution is independent of the fluid properties and film thickness. For radial flow between coaxial parallel disks and parallel plates, the hydraulic diameter D is given by

$$D = 2h \tag{A.9}$$

Generally, the mean friction factor is related to Reynolds number by a relation of the following form.

$$\bar{f} = \frac{k}{Re^n} \tag{A.10}$$

It is useful to express the Reynolds number in the following form.

$$Re = \frac{2\mathring{M}}{W\mu} \tag{A.11}$$

Now, both laminar and turbulent flow cases will be considered.

(a) Laminar flow. - For laminar flow, the friction factor is derived from an exact classical, viscous flow differential equation solution and the derivation is presented in reference 1. This equation has been experimentally verified. The resulting mean friction factor - Reynolds number relation is

$$\bar{f} = \frac{24}{Re} = \frac{12\mu W}{\mathring{M}} \tag{A.12}$$

Thus

$$\mathring{M} = \frac{\rho h^3 W(P_1 - P_2)}{12\mu L} \tag{A.13}$$

Note the strong cubic dependence of the leakage on the gap. If the gap is doubled the leakage would increase eight times for laminar flow. Although there is lesser dependence on gap for turbulent flow and in labyrinth seal, tight control of the flow cross sectional area is essential. The clearance between the seal and shaft that forms the flow path should be as small as possible. However, the minimum clearance possible is limited by shaft deflections, variations in bearing film thicknesses, fabrication and assembly tolerances, unequal expansions, etc. Because of variations present in any seal the gap in equation (A.13) can be the "mean effective-separation" between the sealing surfaces. This equation is used to estimate leakage in rubbing contact seals where an effective gap of 40 to 60 microinches is commonly used.

(b) Turbulent flow. - The Blasius relation of friction factor - Reynolds number appears to satisfactorily describe a large class of fully developed flows even though it is experimentally determined. Thus in equation (A.10), $k = 0.079$ and $n = 0.25$. Substitution in equation (A.7) yields

$$\dot{M} = \frac{2^{1/7}\rho^{4/7}h^{12/7}W(P_1 - P_2)^{4/7}}{(0.079)^{4/7}L^{4/7}\mu^{1/7}} \tag{A.14}$$

which gives the functional relation of the variables in quasi-fully developed turbulent flow. Note that the leakage dependence on film thickness is no longer cubic but less than quadratic. However, the leakage is still most sensitive to gap.

Dependence is the same for flow width but less so (4/7) on density flow length and pressure differential can be seen. Also notice the weak dependence on molecular viscosity - a characteristic of turbulent flow. However, turbulent flow is characterized by large scale momentum exchange (eddies). This macroscopic fluid behavior can be represented as apparent shear stresses, hence a turbulence viscosity which can be orders of magnitude higher than the molecular viscosity. Hence turbulent flow leakage is characterized by a high effective viscosity which means lower leakage than predicted by laminar flow models, but also higher shear heating.

Variable Area Flows

Equation (A5) can be integrated for both radial flow and constant width flow with small deformations of the sealing surfaces. The following results are obtained.

Radial flow ($W = 2\pi r$)

(a) Laminar flow

$$\dot{M} = \frac{\pi \rho h^3 (P_1 - P_2)}{6\mu \ln \frac{R_2}{R_1}} \tag{A.15}$$

$$P = P_1 - (R_2^2 - R_1^2) \frac{\ln \frac{r}{R_1}}{\ln \frac{R_2}{R_1}} \tag{A.16}$$

(b) Radial flow

$$\dot{M} = \frac{2\pi \rho^{4/7} h^{12/7} (P_1 - P_2)^{4/7}}{(0.079)^{4/7} \mu^{1/7} \left[\frac{1}{R_1^{3/4}} \quad \frac{1}{R_2^{3/4}}\right]^{4/7}} \tag{A.17}$$

The pressure distribution can be found from

$$P = P_1 - \frac{(P_1 - P_2)\left[\frac{1}{R_1^{3/4}} - \frac{1}{r^{3/4}}\right]}{\left[\frac{1}{R_1^{3/4}} - \frac{1}{R_2^{3/4}}\right]} \tag{A.18}$$

Laminar Flow with Small Linear Deformation and Constant Width

Surface deformations can be represented by an "effective" or "apparent" linear tilt of the surface. The film thickness at any distance along the leakage flow path X can be found from

$$h = h_1 + \beta X$$

where β is the relative surface tilt angle.

(a) Laminar flow

$$\mathring{M} = \frac{\rho W h_{char}^3 (P_1 - P_2)}{12\mu L} \tag{A.19}$$

where

$$h_{char} = \left(\frac{h_1^2 h_2^2}{h_m}\right)^{1/3} \tag{A.20}$$

The mass leakage flow rate is identical to the parallel surface equation (A.13) except a characteristic film thickness is used. The pressure distribution is

$$P = P_1 - (P_1 - P_2)\,\frac{X}{L}\left[\frac{h_2^2(2h_1 + \alpha X)}{2h_m(h_1 + \alpha X)^2}\right] \tag{A.21}$$

(b) Turbulent flow

$$M = \frac{2^{1/7} W h_{char}^{12/7} \rho^{4/7} (P_1 - P_2)^{4/7}}{(0.079)^{4/7} \mu^{1/7} L^{4/7}} \tag{A.22}$$

The pressure distribution is also identical to that found for laminar flow (eq. (A.21)).

These results are summarized in table AI. Similar leakage equations can be derived for gases. This is done in reference 1. The flow can be assumed to be isothermal (due to small gap assumption) and the perfect gas relation is used to relate the variable density with pressure variation. Because of compressibility the pressure profile is no longer linear for parallel surfaces. Some of these results are summarized in table AII.

APPENDIX B

SEAL PRESSURE BALANCING FUNDAMENTALS

One of the prime objectives in fluid film face seal design is to insure that the face loading is sufficiently low so high heat generation and high wear are prevented; however, contact or close clearance operation must be maintained at all operating conditions. Seal balance can be achieved, at least theoretically, by properly adjusting the secondary seal diameter (see figs. B1 and 3). A common term used by seal designers is the geometric balance ratio or modulus. This modulus is defined as the ratio of the hydrostatic closing area to primary sealing face (dam) area and is used to determine the location of the secondary seal diameter. It is desirable to predict this location analytically.

Unfortunately, a fluid film seal may only be "balanced" at one combination of operating conditions. "Balancing" is strongly dependent on film thickness variation. For gases the pressure profile factor varies with the sealed gas pressure differential. The pressure profile load factor is defined as the ratio of sealing face pressure opening or separating force to sealed hydrostatic pressure closing force. The pressure profile factor is defined as the ratio of the net or average sealing (dam) face pressure to sealed pressure differential. From hereon, the pressure profile load factor will be referred to as the load factor. Both the load factor and geometric balance ratio have other names in the literature and sometimes defined in slightly different ways. Since the load factor can equal the geometric balance modulus at only one set of operating conditions, it is therefore impossible to completely balance an ordinary face seal for all situations. Engineering judgment must be employed to select the proper design.

The importance of the load factor and the geometric balance ratio can be illustrated by considering a face seal force balance. The basic equation defining seal closing force is (see fig. B1)

$$\text{Net closing force} = F_s \pm (F_f + F_I) + A_{HS}\,\Delta P - W\int_0^L P\,dX$$

Design philosophies differ; however, a common pressure balancing practice for fluid film seals is to select the spring force F_s, to overcome only the frictional forces F_f and the inertial forces F_I. (The frictional forces are due to the secondary seals (e.g., O-rings, piston rings) and the antirotation lugs (e.g., torque pins) rubbing on the housing.)

A fundamental consideration in designing pressure balanced seals is the selection of the secondary seal diameter. This diameter determines the hydrostatic closing force as illustrated in figure B1. By proper positioning of the secondary seal diameter, this closing force can be equal to the sealing dam pressure opening force or, at least theoretically, any degree of seal face loading. The secondary seal diameter can be found from the geometric balance ratio δ where

$$\text{Geometric balance ratio, } \delta = \frac{A_{HS}}{A_{SD}} = \frac{R_s^2 - R_i^2}{R_o^2 - R_i^2}$$

Another important parameter is the pressure profile load factor, $\overline{F}$ which is defined as the pressure (pneumatic) opening force normalized to the sealed pressure differential force acting over the entire sealing dam (seal face) area or

$$\overline{F} = \frac{\text{Pressure opening force}}{\Delta P \ A_{SD}}$$

Across a sealing dam

$$\text{Opening force} = W \int_0^L P \, dX$$

For a linear pressure drop (valid for incompressible fluids and parallel sealing surfaces)

$$P = P_1 - (P_1 - P_2) \frac{X}{L}$$

Thus

$$\text{Opening force} = \frac{WL \ \Delta P}{2}$$

and

$$\overline{F} = \frac{1}{2}$$

If the seal opening force is equated to the hydrostatic closing force

$$W \int_0^L P \, dX = \Delta P \ A_{HS}$$

and substituting this condition into the load factor relation results in

$$\overline{F} = \frac{A_{HS}}{A_{SD}} = \delta$$

When this situation exists, that is the load factor is equal to the geometric balance ratio, the seal is said to be perfectly balanced.

Once the load factor is known, the seal balance diameter can be simply calculated.

$$\text{Seal balance diameter} = \overline{F}(2R_2 - 2R_1) + 2R_1$$

(For a perfectly balanced seal, the seal balance diameter equals the secondary seal diameter.) For some cases the sealing dam opening force can be evaluated analytically and hence the load factor can be predicted analytically.

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TABLE I. - SEAL CATEGORIES

I. Static

II. Dynamic
- Motions
  - Rotary
  - Oscillatory
  - Reciprocating
- Classification - types, selection
- Fundamentals
- Seal systems - examples

TABLE II. - SEALING DISCIPLINES

- Tribology (lubrication science)
- Fluid mechanics and heat transfer
- Solid mechanics
- Elastohydrodynamics
- Dynamics
- Materials
  - Metallurgy
  - Chemistry and physics
    - Bulk
    - Surface

TABLE III. - SEAL SELECTION AND DESIGN

1. Allowable leakage rate
2. Available space
3. Seal duty requirement (ΔP, T, V)
4. Sealed fluid media
5. Maintainability and life requirements
6. Necessity for external accessories
7. Wear and/or rubbing behavior of seal materials
8. Differential growths
9. Cost - initial against total life cycle

TABLE IV. - DYNAMIC SEAL CLASSIFICATION

1. Selection and control of geometry

   A. Positive (rubbing) contact

      1. Mechanical face
      2. Circumferential
      3. Lip
      4. Soft packing

   B. Close clearance

      1. Hydrodynamic
      2. Hydrostatic
         Externally pressurized
         Self-energized
      3. Floating bushings

   C. Fixed geometry clearance seals

      1. Fixed bushing
      2. Labyrinth

2. Control of fluid properties

   Freeze
   Ferromagnetic

3. Control of fluid forces

   Centrifugal
   Screw pump
   Magnetic

TABLE AI. - MASS LEAKAGE FLOW RATES FOR FULLY DEVELOPED FLOW RATES

	Laminar flow	Turbulent Flow
Constant area, parallel surfaces	$M = \dfrac{\rho W h^3 (P_1 - P_2)}{12\mu L};$	$\dot{M} = \dfrac{2^{1/7} \rho^{4/7} h^{12/7} W (P_1 - P_2)^{4/7}}{(01079)^{4/7} L^{4/7} \mu^{1/7}}$
Radial flow	$\dot{M} = \dfrac{\rho \pi h^3 (P_1 - P_2)}{6\mu \ln \frac{R_2}{R_1}}$	$\dot{M} = \dfrac{2\pi \rho^{4/7} h^{12/7} (P_1 - P_2)^{4/7}}{(0.079)^{4/7} \mu^{1/7} \left(\dfrac{1}{R_1^{3/4}} - \dfrac{1}{R_2^{3/4}} \right)^{4/7}}$
Small linear deformations ($h = h_1 + \alpha X$) and constant width	$\dot{M} = \dfrac{\rho W h_{char}^3 (P_1 - P_2)}{12\mu L};$ where $h_{char} = \left(\dfrac{h_1^2 h_2^2}{h_m} \right)^{1/3}$	$\dot{M} = \dfrac{2^{1/7} W h_{char}^{12/7} \rho^{4/7} (P_1 - P_2)^{4/7}}{(0.079)^{4/7} \mu^{1/7} L^{1/7}}$

TABLE AII. - MASS LEAKAGE FLOW RATE EQUATIONS FOR VARIOUS QUASI-FULLY DEVELOPED (SUBSONIC) FLOW RATE SITUATIONS

Case	Laminar Flow	Turbulent flow
Constant area, parallel surfaces	$\dot{M} = \dfrac{Wh^3(P_1^2 - P_2^2)}{24\mu\mathcal{R}TL}$	$\dot{M} = \dfrac{Wh^{12/7}(P_1^2 - P_2^2)^{4/7}}{2^{3/7}(0.079)^{4/7}\mathcal{R}^{4/7}T^{4/7}\mu^{1/7}L^{4/7}}$
Radial flow	$\dot{M} = \dfrac{h^3(P_2^2 - P_1^2)}{12\mu\mathcal{R}T \ln \dfrac{R_1}{R_2}}$	$\dot{M} = \dfrac{3^{4/7}h^{12/7}(P_1^2 - P_2^2)^{4/7}}{2(0.079)^{4/7}\mu^{1/7}\mathcal{R}^{4/7}T^{4/7}\left(\dfrac{1}{R_1^{3/4}} - \dfrac{1}{R_2^{3/4}}\right)^{4/7}}$
Small linear deformations ($h = h_1 + \beta x$) and constant width	$\dot{M} = \dfrac{Wh_{char}^3(P_1^2 - P_2^2)}{24\mu\mathcal{R}TL}$	$\dot{M} = \dfrac{Wh_{char}^{12/7}(P_1^2 - P_2^2)^{4/7}}{2^{3/7}(0.079)^{4/7}\mu^{1/7}\mathcal{R}^{4/7}T^{4/7}L^{4/7}}$

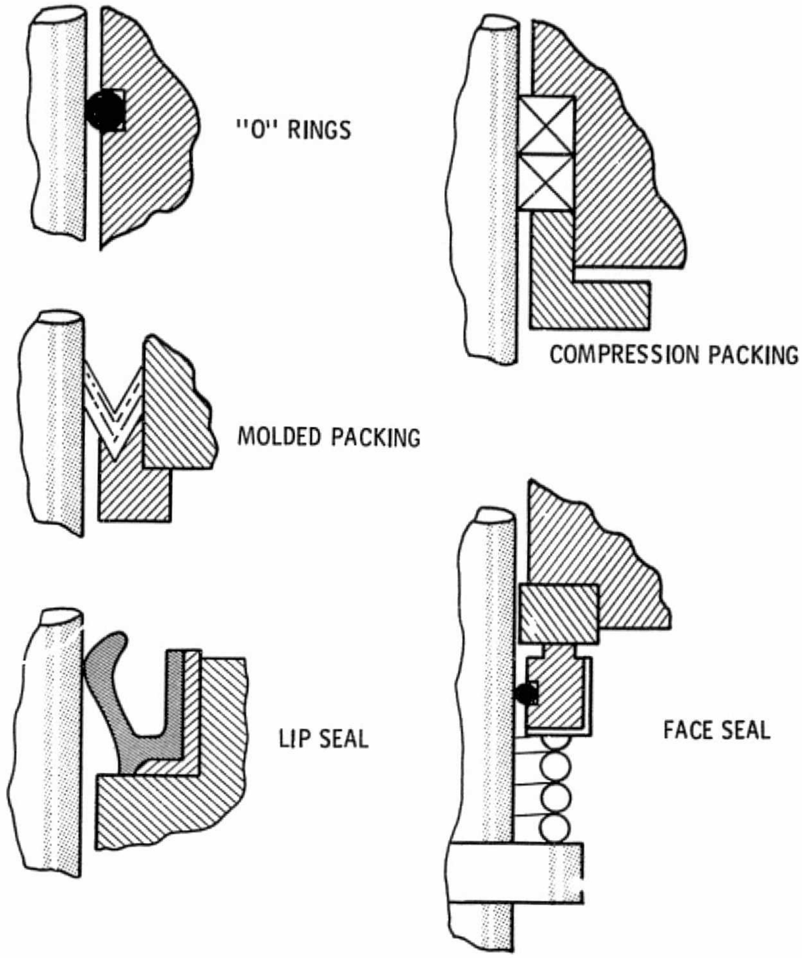

Figure 1. - Examples of rubbing contact type seals.

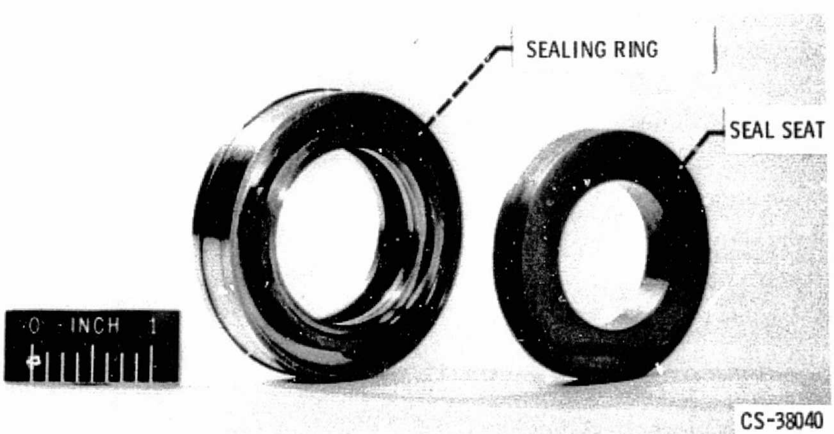

Figure 2. - Mechanical face seal.

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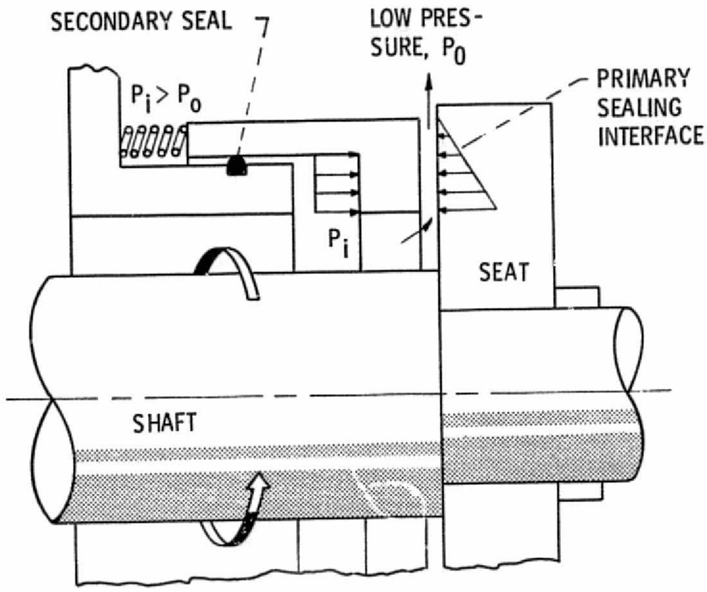
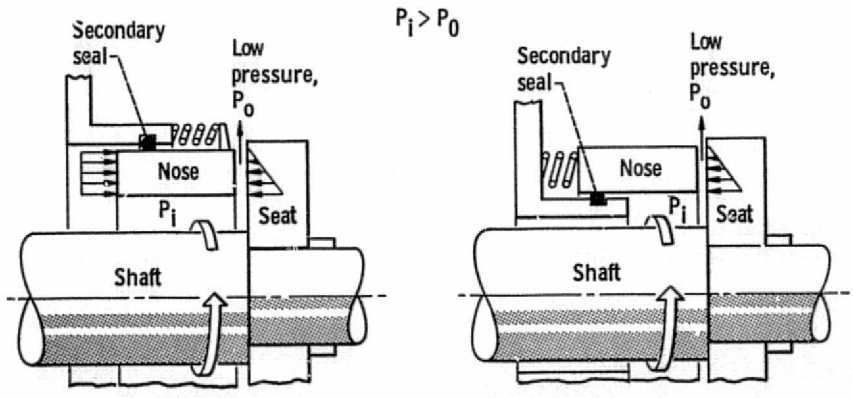


Figure 3. - Pressure balanced contact seal.



(a) Pressure loaded.

(b) Pressure unloaded.

Figure 4. - Unbalanced face contact seals.

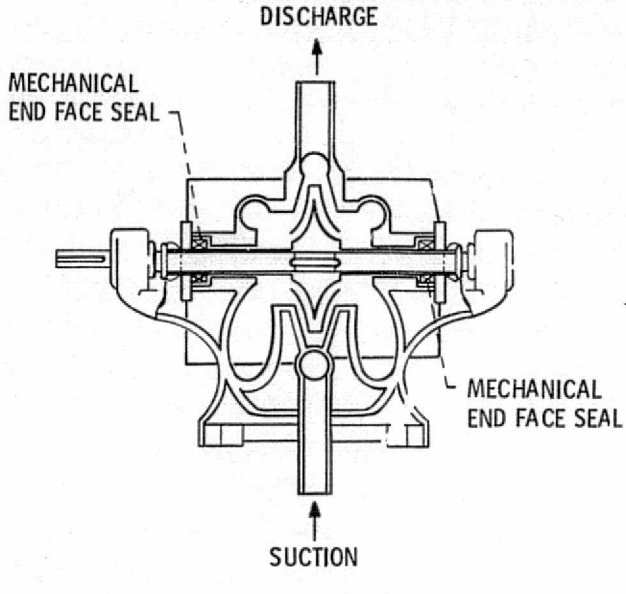


Figure 5. - End face seal application in a pump.

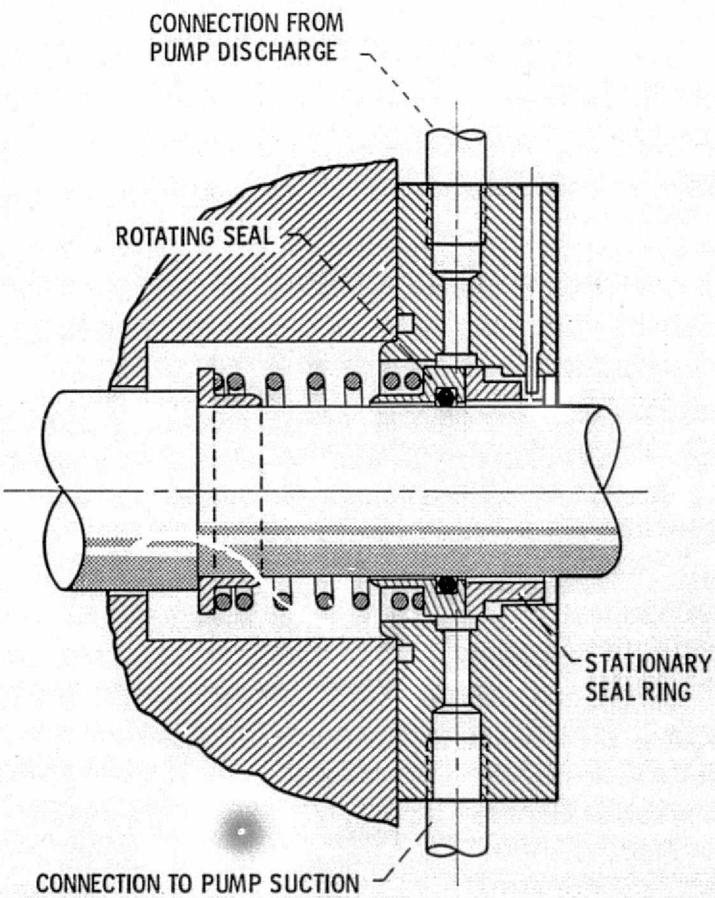


Figure 6. - Pump end face seal - rotating seal ring.

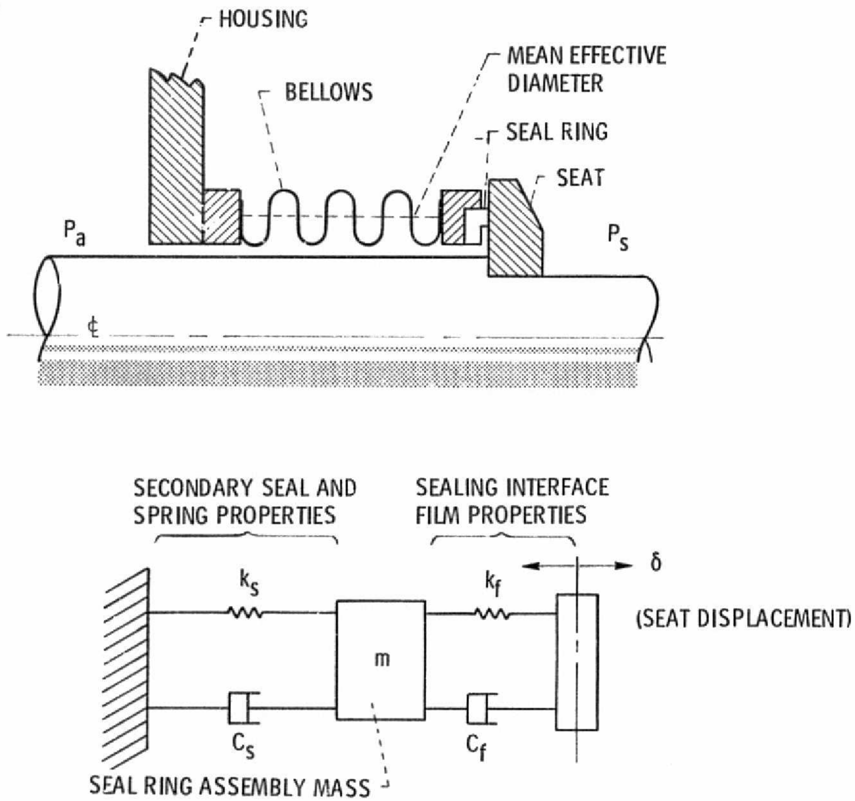
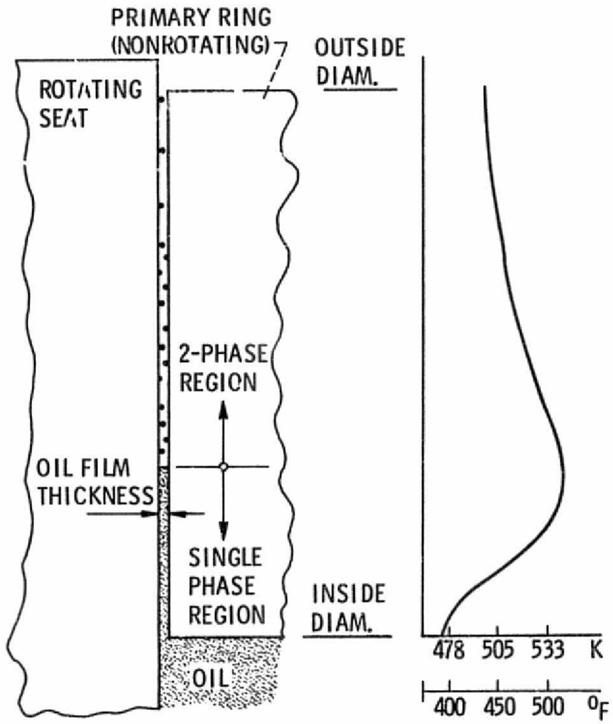


Figure 7. - Lumped parameter model for analyzing seal dynamics.



(a) SCHEMATIC OF OIL FILM IN PRIMARY SEAL.

(b) PRIMARY SEAL RADIAL TEMPERATURE PROFILE (DATA FROM REF. 2).

Figure 8. - Conventional oil lubricated radial face seal; seal siding speed, (37 ft/sec).

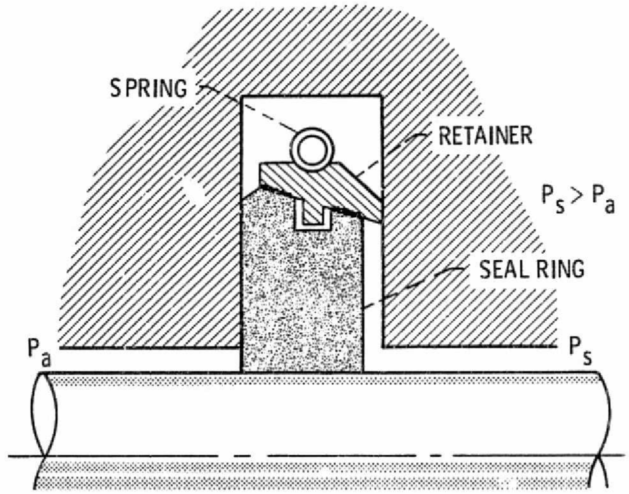


Figure 9. - Circumferential shaft riding seal.

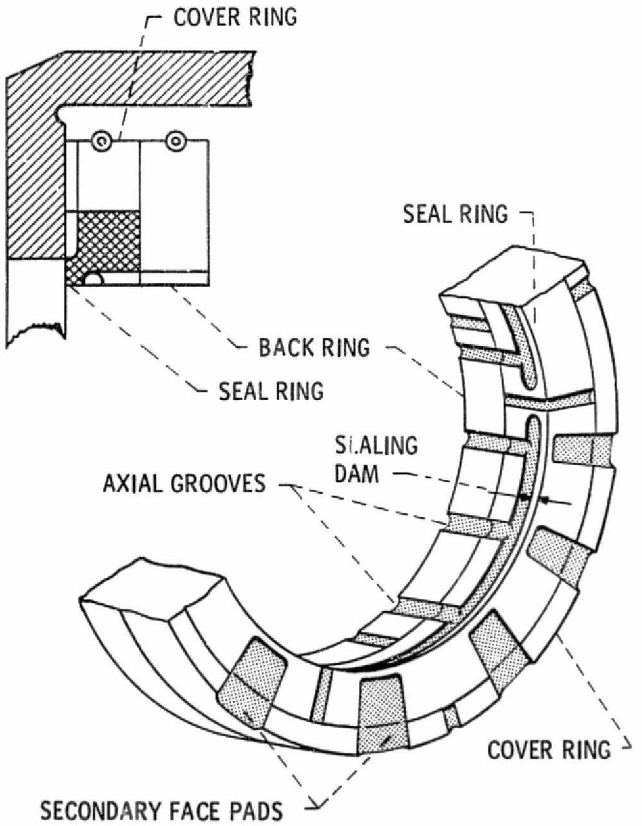


Figure 10. - Partially "pressure balanced" circumferential shaft riding seal.

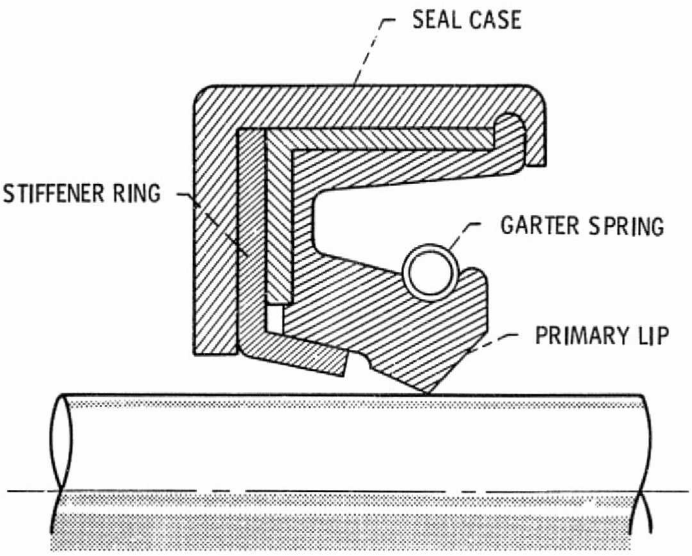


Figure 11. - Lip seal components.

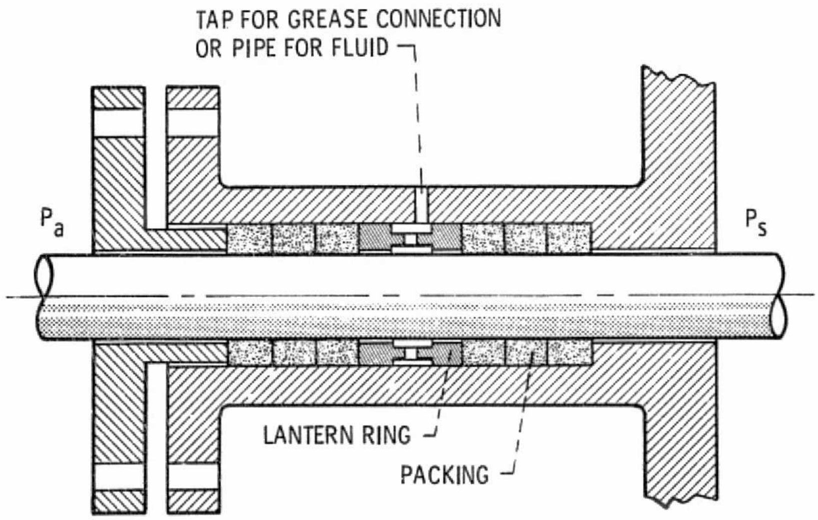
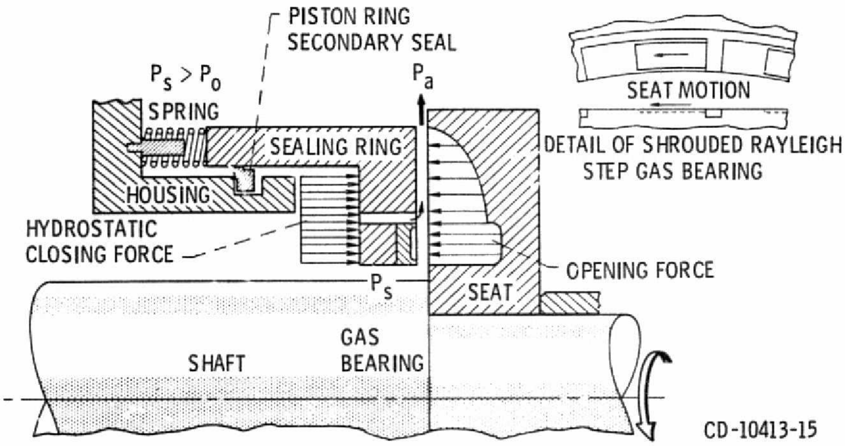
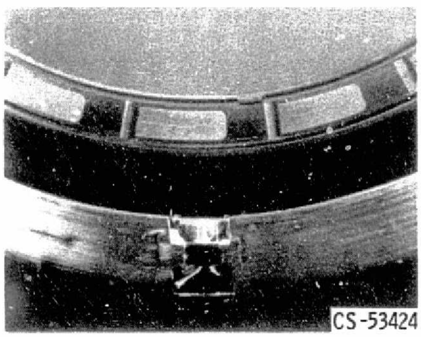


Figure 12. - Typical stuffing box with gland packing seals.



(a) SELF-ACTING LIFT PAD HYDRODYNAMIC SEAL.



(b) PHOTOGRAPH OF SEAL RING SURFACE.

Figure 13. - Self-acting lift pad seal.

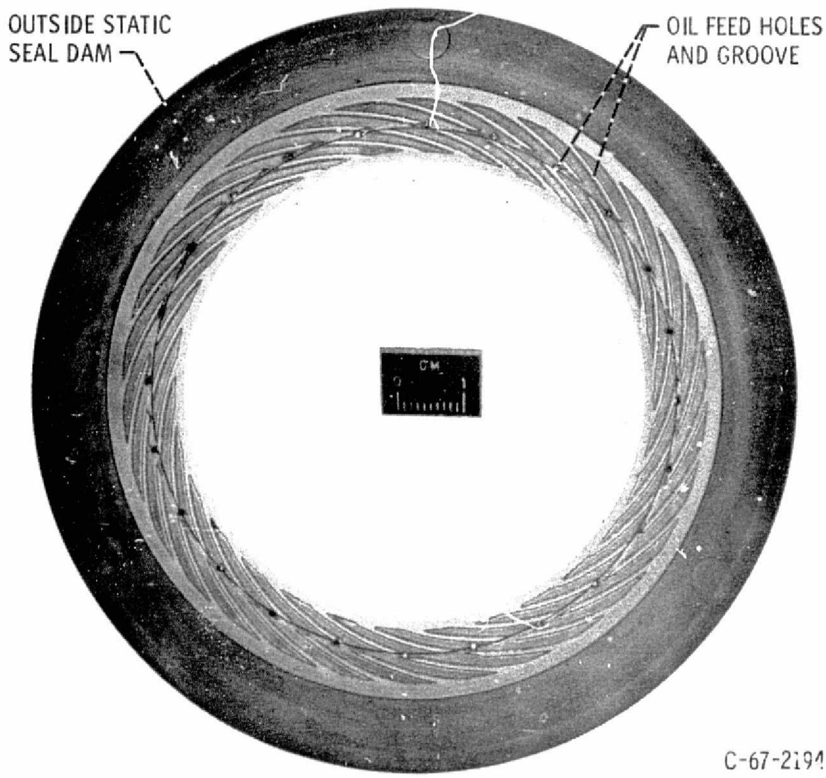


Figure 14. - Spiral feedgroove seal.

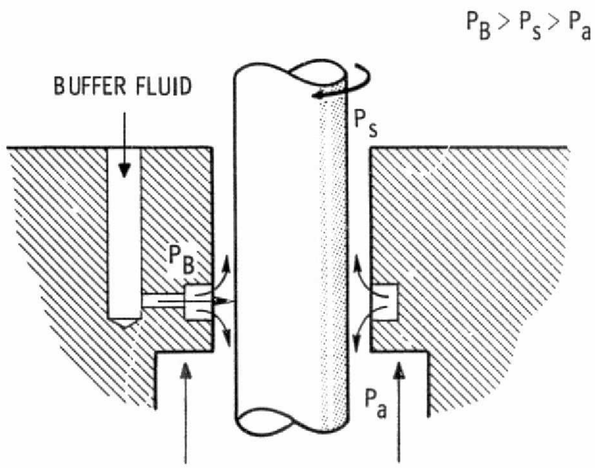


Figure 15. - Externally pressurized hydrostatic seal.

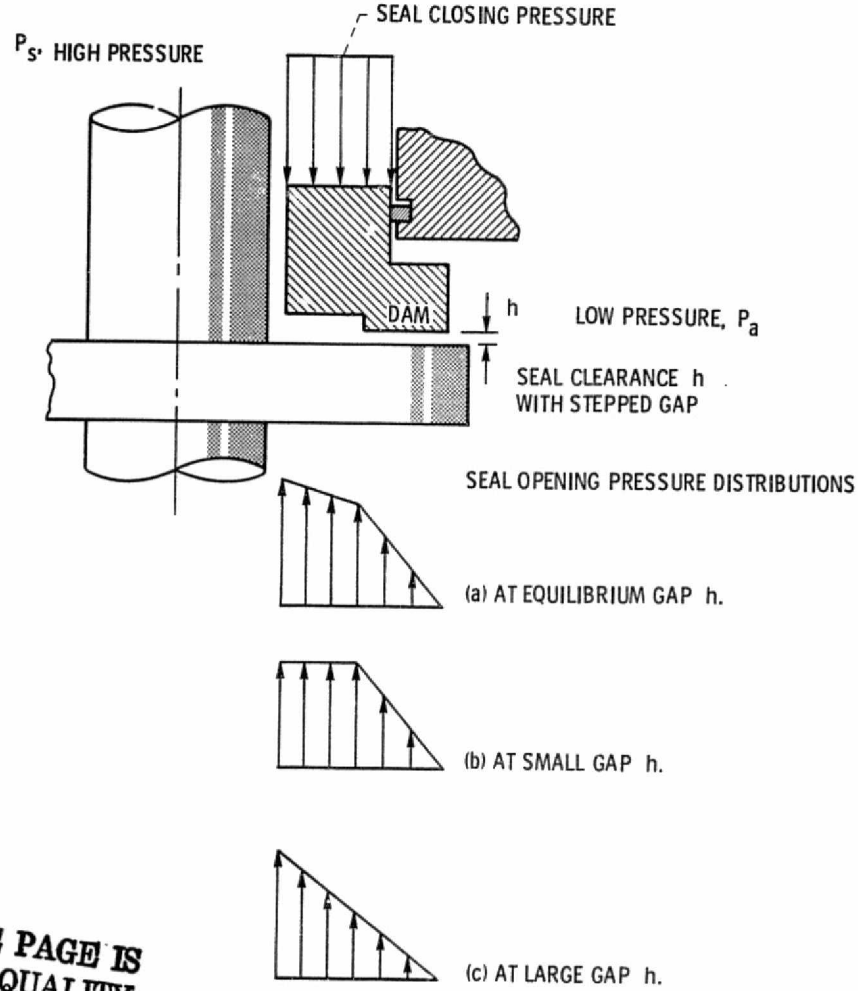


Figure 16. - Self-energized hydrostatic step seal.

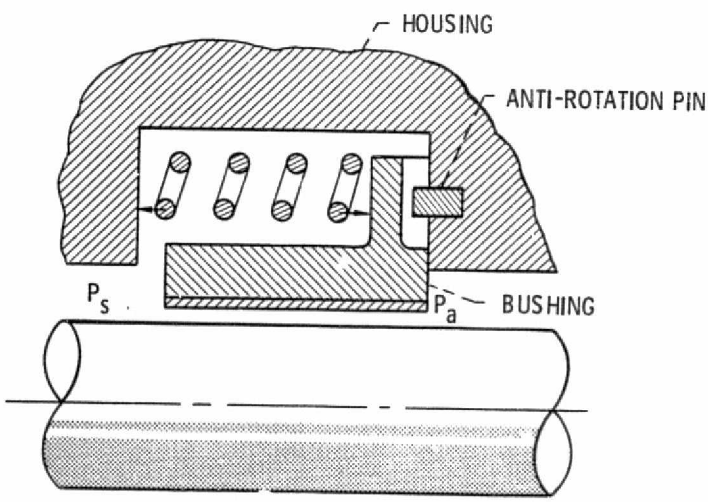


Figure 17. - Floating bushing seal type.

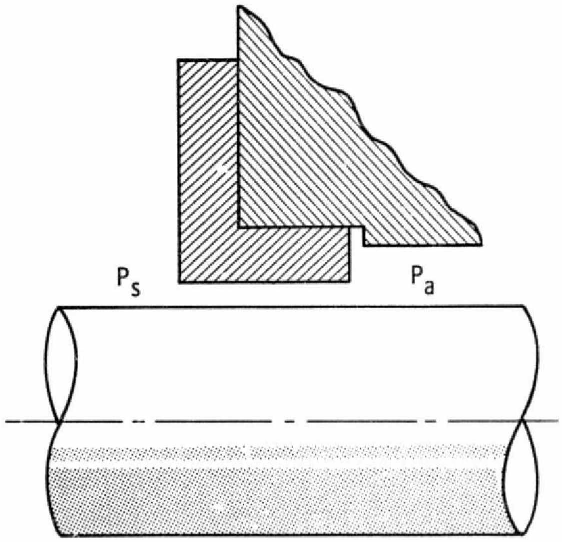
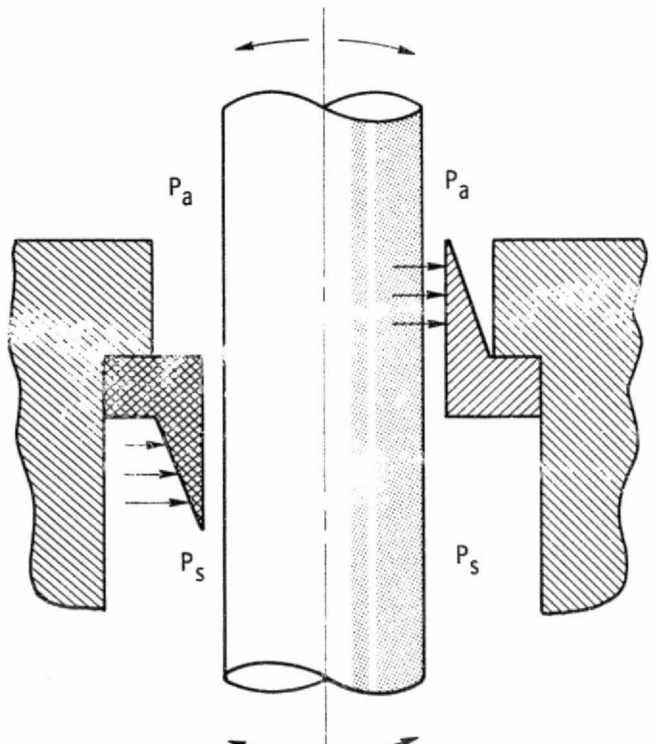


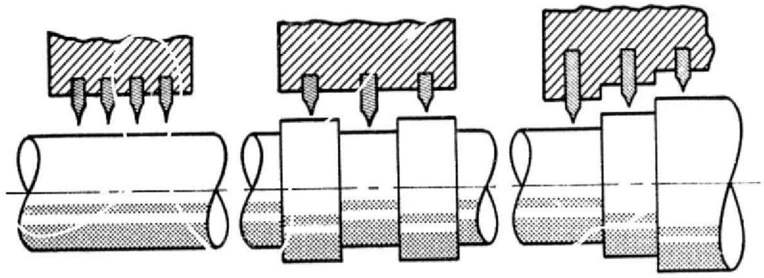
Figure 18. - Fixed bushing seal.



(a) HIGH PRESSURE CLOSING CLEARANCE.

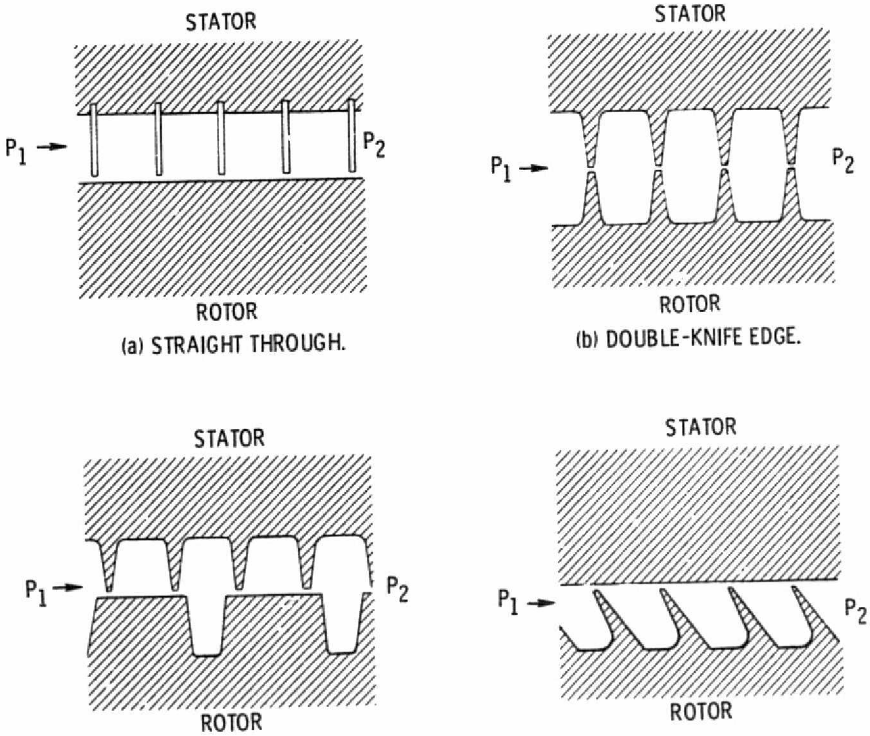
(b) HIGH PRESSURE OPENING CLEARANCE.

Figure 19. - Deformable bushing seal designs.



(a) STRAIGHT THROUGH. (b) HIGH-LOW. (c) STEPPED.

Figure 20. - Labyrinth seal types.



(a) STRAIGHT THROUGH. (b) DOUBLE-KNIFE EDGE. (c) MODIFIED DOUBLE-KNIFE EDGE. (d) CANTED-KNIFE EDGE.

Figure 21. - Labyrinth seal configurations.

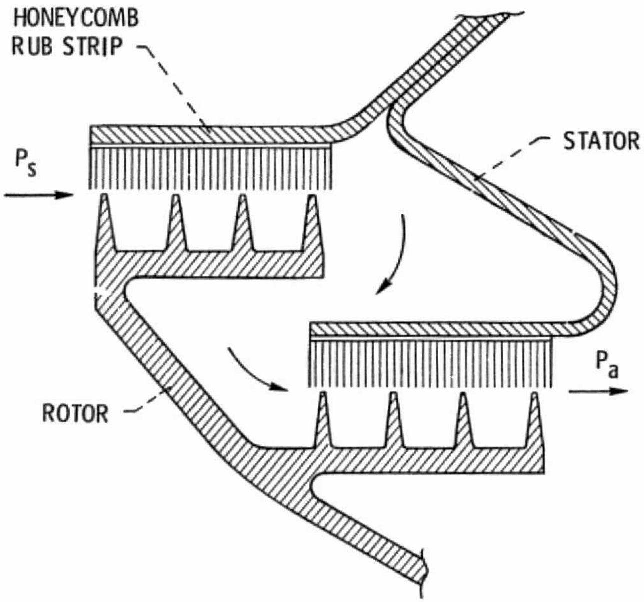


Figure 22. - Common labyrinth seal system used as aircraft gas turbine inner air seals.

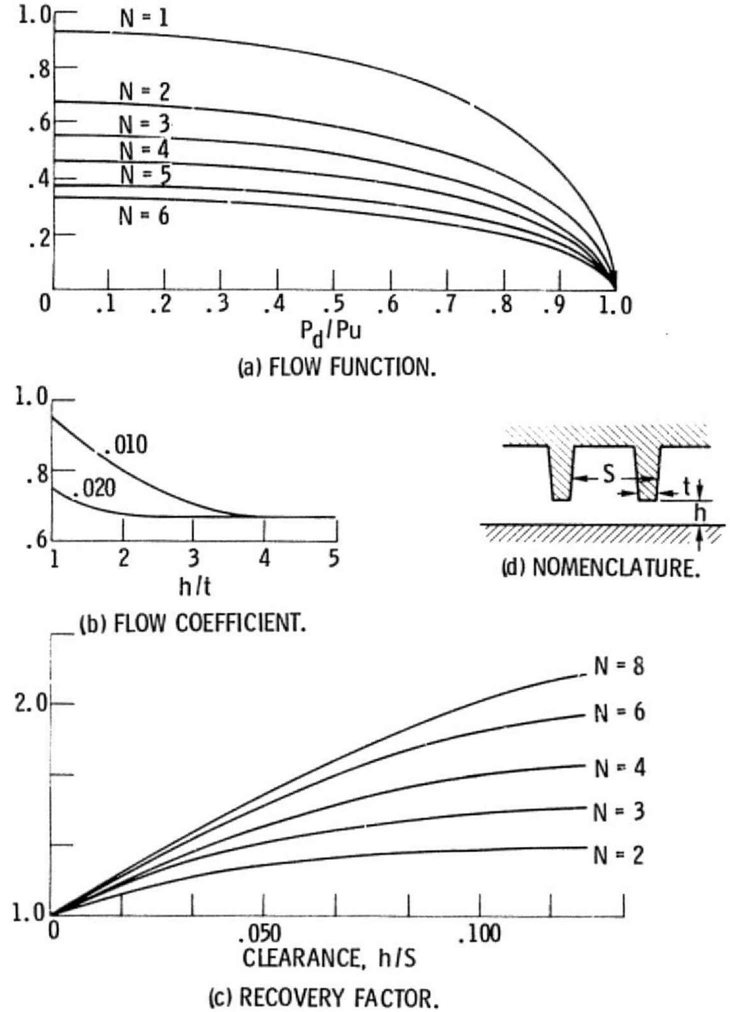


Figure 23. - Plots of functions used in Egli labyrinth seal leakage equation.

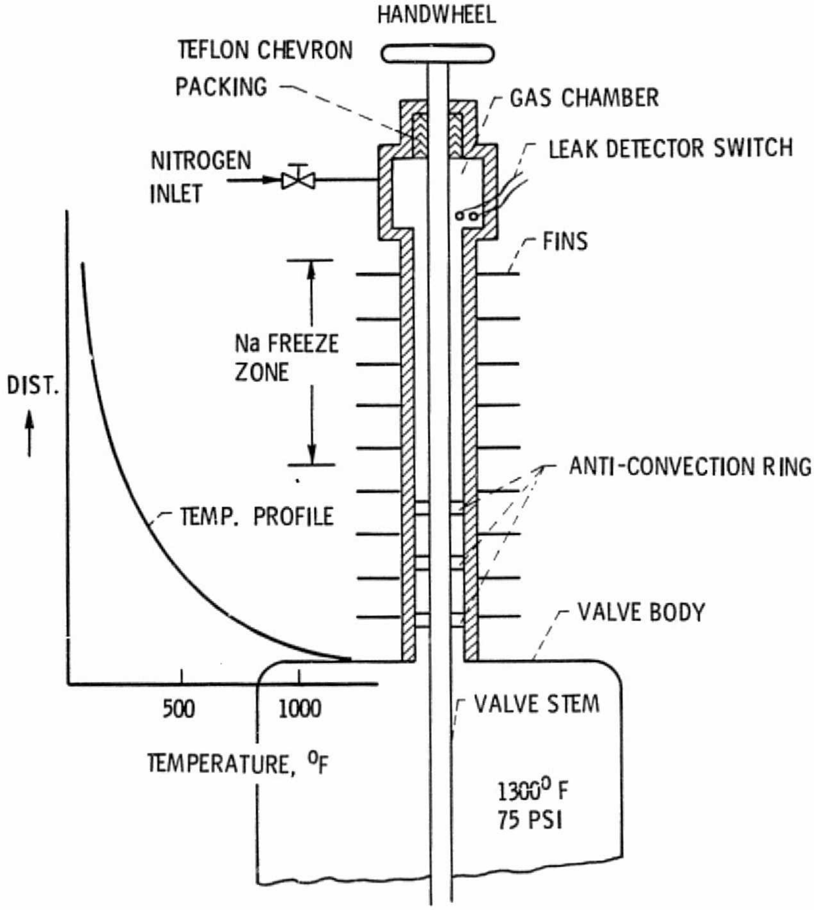


Figure 24. - Freeze seal schematic.

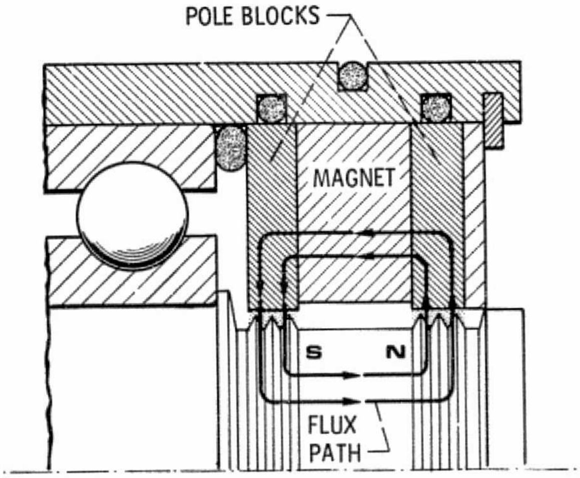


Figure 25. - Ferromagnetic seal.

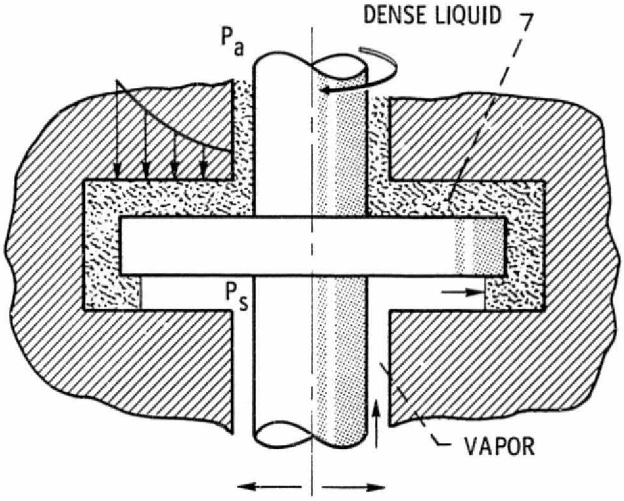
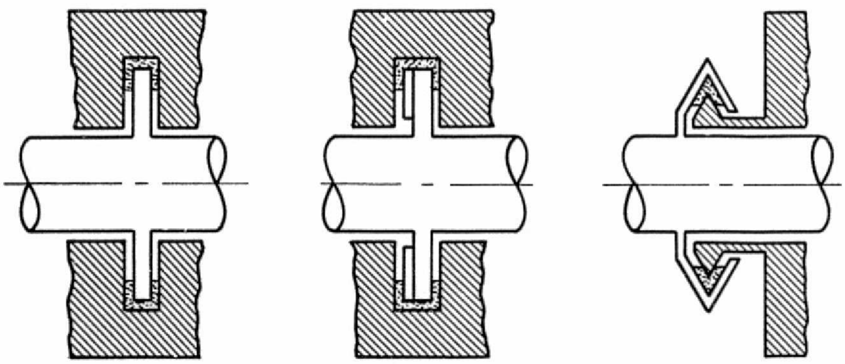


Figure 26. - Slinger or rotating fluid ring seal.



(a) ROTATING DISK. (b) ROTATING VANED DISK. (c) ROTATING HOUSING.

Figure 27. - Some centrifugal seal configurations.

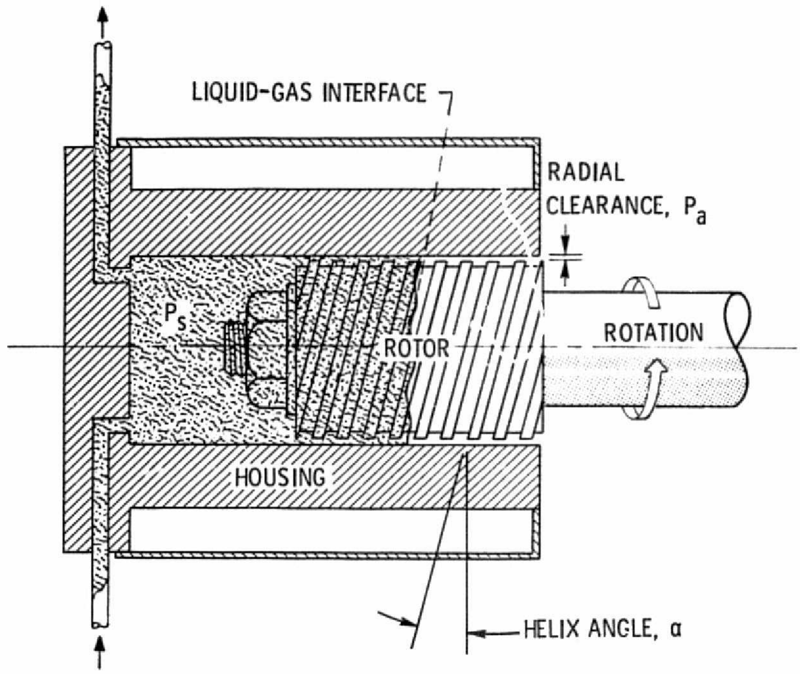


Figure 28. - Viscoseal.

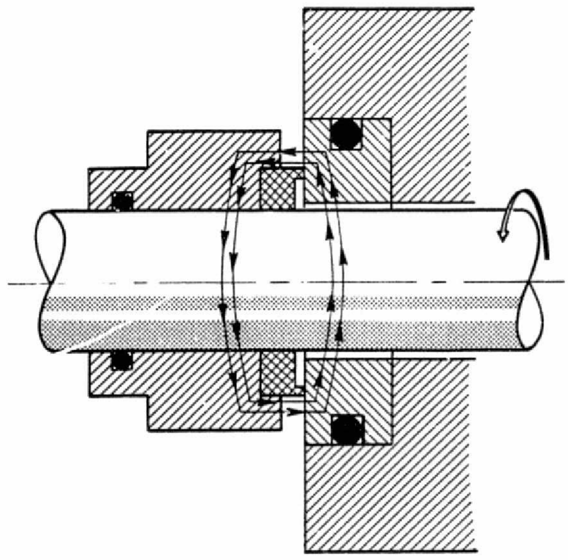


Figure 29. - Magnetic face seal.

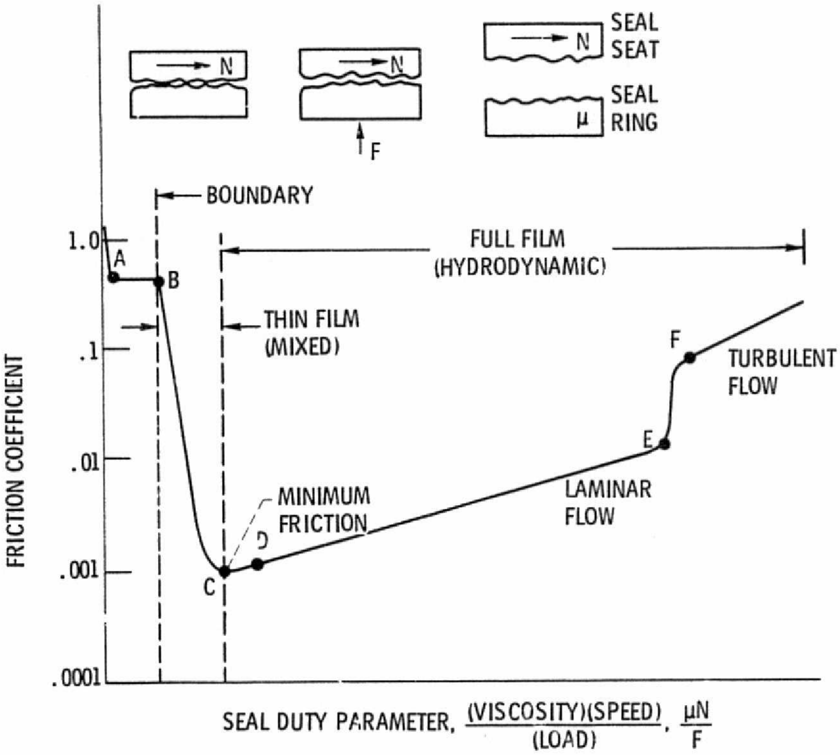


Figure 30. - Friction coefficient variation with liquid lubrication.

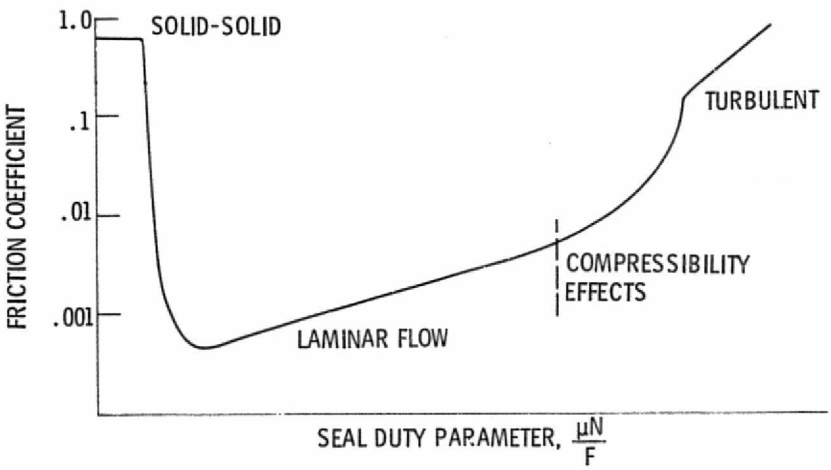


Figure 31. - Friction coefficient variation with gas lubrication.

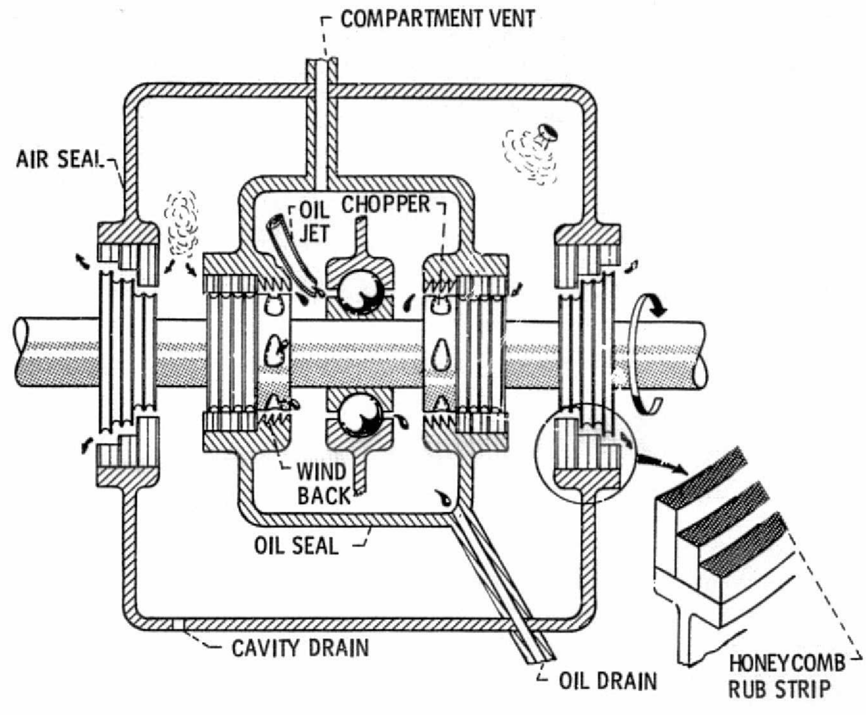


Figure 32. - Aircraft gas turbine engine sump seal system.

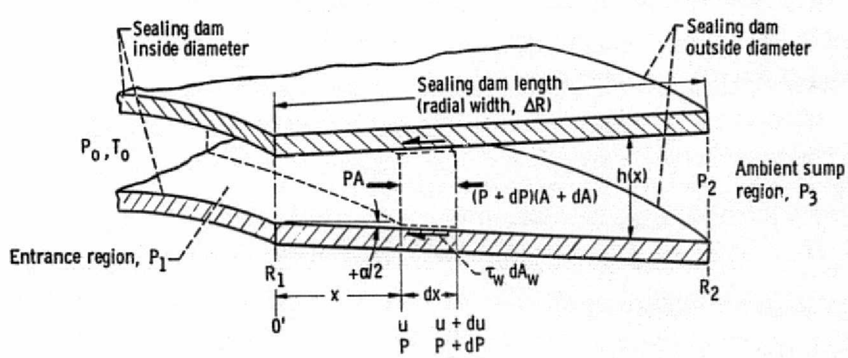


Figure A1. - Model and notation of sealing faces for leakage flow analyses.

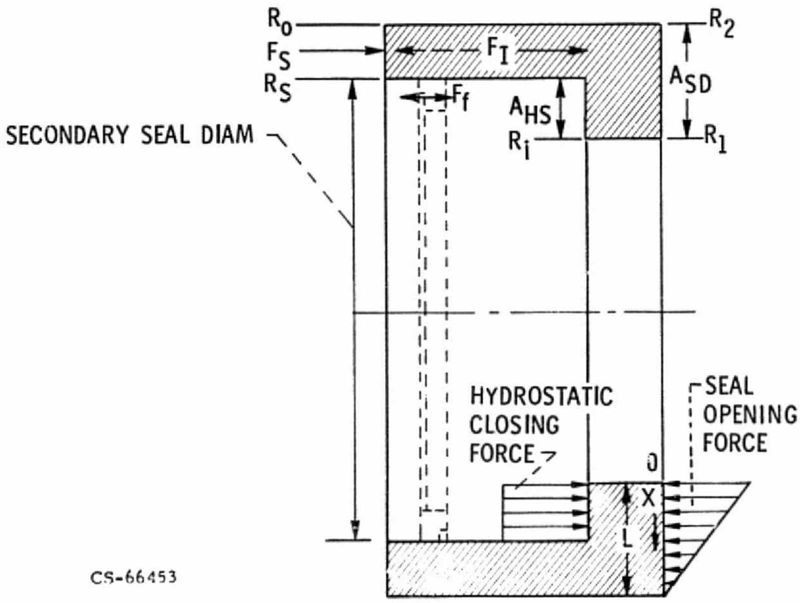


Figure B1. - Schematic of radial face seal identifying the nomenclature.