Alternatives in Gearbox Seals for Main Drive Gearboxes

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ABSTRACT

A variety of sealing devices have been used to seal Marine Main Reduction Gears. One means of sealing Main Reduction Gears today is the labyrinth seal. While the labyrinth seal has been successfully employed in some applications, it has performed unsatisfactory in others. This paper discusses mechanical seals as alternatives to labyrinth seals for use in Marine Main Reduction Gears. Along with a discussion of Labyrinth Seals, various types of mechanical seals are considered, along with their respective advantages and disadvantages. Split Circumferential Seals are described in particular. Seal design features, as well as test results, are presented.

KEY WORDS: Circumferential Seal; Labyrinth Seal; Lip Seal; Hydrodynamic Seal; Marine Main Reduction Gear; Mechanical Seal, Shaft Sealing, Gearbox

INTRODUCTION

Main Reduction Gears (MRGs) predominately utilize labyrinth seals to provide sealing. Since no such thing as a perfect seal exists, one must weigh associated positives and negatives when selecting a sealing device. While one may be tempted to focus on the devices positives, if one cannot effectively manage its negatives, then other seal devices must be considered. Two alternatives to the labyrinth seal are the lip and circumferential seals. A discussion of these sealing solutions is presented here.

After a discussion of the typical MRG sealing requirements, a discussion of labyrinth seals emphasizes its features and performance characteristics. Lip seals are introduced as a device capable of alleviating some of the performance problems of labyrinth seal under certain conditions.

The circumferential seal is analyzed in some detail. Along with a discussion of seal features, analytic design tools are highlighted. The hydroload concept is described. Relative advantages and disadvantages with respect to labyrinth and lip seals are discussed. Installation of the split housing design is described, as well as replacement procedures. Test results are presented regarding oil leakage rates, life, and shock robustness.

SEALING REQUIREMENTS

The primary purpose of a main reduction gear seal is to prevent the loss of lubricating oil from the gearbox. Inadequate lubrication can adversely affect gearbox performance, and ultimately lead to failure. This oil leakage affects not only gearbox function, but has adverse consequences in the ship compartment as well. Leakage can occur as oil mist, solid oil, or a combination of both. In the case of oil mist, cleanliness problems can develop as oil settles on surfaces. Such mist may require venting to the atmosphere, which can lead to environmental problems. In addition, an oil mist rich environment is undesirable for nearby personnel. Likewise, solid oil splashing from the reduction gear can cause environmental problems when one considers its disposal. In the typically confined areas around the MRG, oil leakage that settles on the disk brake can be a fire hazard. This can result in the rubber boot catching fire. The need to prevent this has been an overriding concern when selecting a sealing device. The rolling of a ship causes difficulties for sealing with clearance devices in particular.

In addition to keeping lubricating oil in, for the variety of reasons described above, the seal must prevent outside debris from entering the gearbox. Such debris can cause damage to gears and bearings, preventing proper function and lessening working life. It may be critical for the seal to prevent seawater from entering the gearbox under an emergency condition.

In addition to controlling leakage into and out of the gearbox, a number of other requirements are common. Due to the nature of the MRG system, a split design is necessary for installation and disassembly. The seal footprint is limited. Therefore, the design of a satisfactorily performing seal is constrained to the dimensions available. As seals are either close clearance or contacting devices, eccentricities of the rotating shaft become a factor is design, possibly influencing the seal device selected. Particularly in a military application, the ability to continue to function adequately after sustaining a shock load may be a critical need.

For a specific application, only a portion of the above described requirements will be applicable. It is up to the engineer to determine the critical requirements, and select the seal accordingly. Along with satisfying the performance characteristics described above, consistent performance over the lifetime of the seal is expected.

THE LABYRINTH SEAL

Labyrinth seals are commonly used in MRGs. Two major reasons for this are simplicity and reliability. As a clearance device, one need not worry about labyrinth wear (as long as rubs do not occur). Also, due to the nature of clearance devices, maintenance is low, since replacements due to wear are unusual under normal circumstances. Shaft speed, which influences wear (and therefore life) in a contacting seal, is not an issue with the non-contacting labyrinth seal. In a contacting device, seal friction with the shaft results in power consumption, whereas for labyrinth seals, little or no friction is generated.

Design Features

A labyrinth seal is a clearance device that reduces fluid energy through a series of throttles. A labyrinth seal example is shown in Figure 1. Two major design considerations when designing the labyrinth seal are the clearance between the teeth and the shaft, and the overall length the sealed fluid must travel. Other considerations include the type of tooth geometry, the number of teeth, and the placement of the teeth. Two types of teeth are shown in Figure 1, straight and stepped. Stepped teeth may not be possible with large axial movements. In addition, the placement of oil drains between the labyrinth stages allows for the collection of the escaped oil back to the MRG.

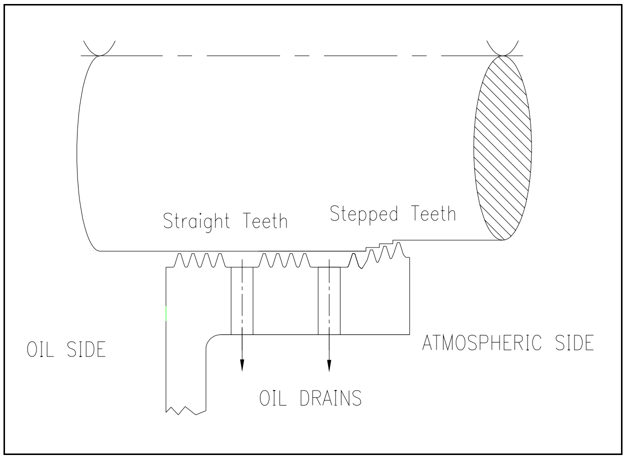

Figure 1 – Three Stage Labyrinth Seal with Oil Drains

The tip clearance chosen depends on a number of factors. One would like to decrease this clearance, since larger clearances result in larger leakages. However, one also wants to prevent shaft rubs due to shaft excursions. The addition of an abradable sleeve on the rotating shaft can be used. This sleeve will wear away in the event of a rub, without causing the negative consequences of metal on metal rubs. However, such a sleeve may not be applicable in many circumstances.

Another design feature is the overall axial length of the seal. The more teeth one has, and the more drain holes, the more tortuous the path oil must follow to escape. However, practical concerns limit ones available length. Especially in cramped compartments, this consideration can become a major factor.

For assembly and disassembly, the seal must be split. This should not affect seal performance when compared to a non-split seal assembly.

Challenges in the Marine Environment

While the advantages of a labyrinth seal are numerous, certain drawbacks lead one to rethink their applicability under certain circumstances. The oil system in the MRG can be such that oil is directly impinging in the vicinity of the seal. One example would be if the input shaft bearing is located nearby. For a labyrinth, which has a clearance, this oil can escape through the clearance in the form of solid oil or oil mist. Oil mist is disagreeable for the reasons described earlier. In the case of solid oil, leakage onto the disk brake has been suspected of causing fires. The many advantages of a labyrinth will be negated if a compartment fire results from its use. Rolling of the ship can also facilitate the leakage of oil. This escape of oil is due mainly to the clearance nature of the seal.

While oil leakage from the MRG is unacceptable, one must also look at material entry into the MRG as well. Debris entering the MRG can cause wear, affect performance, and shorten component life. Debris passes through a clearance device more easily than debris through a contacting device.

Especially in military applications, the ability of the seal to prevent water leakage into the MRG in the event of compartment flooding is usually of importance. The labyrinth seal provides no protection under these conditions. Water will pass through the clearance, and flood the MRG, adversely affecting functionality.

It has been observed that in some cases, a pressure differential will develop across the seal. Reverse pressures, where the area outside the gearbox is at a slightly lower pressure (typically under 1 PSI, usually under 5 inches of water), will draw oil from the MRG into the area outside the gearbox.

THE LIP SEAL

Lip seals differ from labyrinth seals in a number of ways. Perhaps the most significant of these is lip seals are contacting devices, whereas labyrinth seals are clearance devices. Depending on the operating conditions, a lip seal can outperform labyrinths, and alleviate many of the drawbacks associated with them. However, lip seals have operating limitations.

Design Features

Figure 2 shows a typical lip seal. Lip seals are ubiquitous, and the number of design variables makes for a wide variety of types to choose from. Each type has features specific to the environment that it is designed for. In Figure 2 for example, a dust prevention lip is added, possibly an important feature in a MRG application.

A typical lip seal consists of a flexible elastomeric leg held in contact with the rotating shaft by a garter spring or some other form of spring. Additional load is provided by an interference fit between the flexible leg and the shaft. A balance occurs here between making sure the radial load is great enough to prevent loss of contact due to surfboarding on oil and shaft eccentricities, but not high enough to result in high wear.

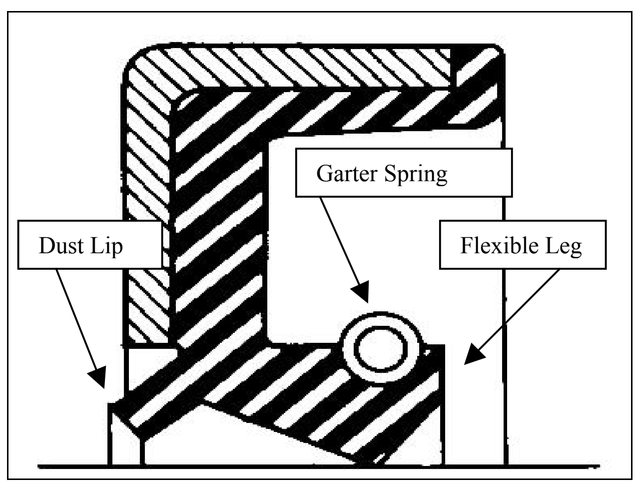

Figure 2 – Lip Seal with Dust Prevention Lip (Buchter 1979)

A thin film of oil is desirable between the flexible leg and the rotating shaft. A film thickness of ~0.0001 inch has been empirically shown to be the best in terms of preventing leakage and assuring low wear.

(Buchter 1979) Using the equation for flow between plane parallel walls, for an incompressible fluid in the laminar region, the total flow is,

$$Q = 2 \cdot \pi \cdot R \cdot \frac{h^3}{12\mu} \cdot \frac{\Delta P}{L}$$

Equation 1

wherein R is the sealing radius, h the clearance, μ the viscosity, ΔP the pressure differential, and L the axial length of the sealed surfaces. We have integrated the unit circumference parallel plate flow over the circumference of the seal.

An analysis of Equation 1 clearly shows the importance of controlling the film thickness. Whereas other variables linearly affect flow, clearance affects flow as a cubic. This sensitivity of leakage to clearance results is the need to carefully control this variable. A knowledge of the operating conditions is necessary to do this. For example, in the case of a slight negative pressure, outside the gearbox pressure being less than the MRG pressure, one must control the forces on the seal in order to prevent a clearance developing greater than the optimal. Also, one must prevent large downward forces, which will increase friction, wear, and power consumption.

The flexible leg can be modified to include hydrodynamic features. In the Helixseal configuration, ribs on the back side of the lip function as hydrodynamic elements, abating oil leakage. Similarly, the Hydroseal makes use of hydrodynamic grooves on the sealing surface. Since these features are angled and depend on shaft windage, they are unidirectional. Care must then be taken to ensure that upon assembly, the hydrodynamic features pump oil back into the MRG, and not out of it.

The axial length required for a lip seal is no more than that of a labyrinth, and can be less. For assembly and disassembly, the seal must be split. This should not affect seal performance when compared to a non-split seal assembly.

Lip seals have been successfully used in preventing oil leakage in MRGs. Oil splash onto the lip must pass through the small film between the lip and the shaft. This has two consequences. It reduces the kinetic energy of the oil, allowing any leaked oil to be drained more easily, and also helps condense any oil mist.

A desirable feature in some applications is to prevent water from entering the MRG in the event of compartment flooding. Figure 3 illustrates a solution to this requirement using the lip seal. In this double seal arrangement, one seal seals oil from escaping the MRG, while the other seals water, debris, or any gases from entering the MRG.

All of these design features make the lip seal a considerable alternative to the labyrinth seal. Being a contacting seal, oil does not have a direct clearance path out of the MRG. Also, with the double seal in Figure 3, water influx into the MRG can be abated.

The MRG application necessitates the installation of a split lip seal. This is possible, and does not affect seal performance. Replacement kits allow for change out of worn seals which no longer satisfy sealing requirements.

Challenges in the Marine Environment

Speed limitations reduce the applicability of the lip seal. It is generally regarded that the maximum surface speed for adequate performance is 30 ft/sec. While the MRG output shaft may fall into this category, the input shaft usually exceeds this speed. At speeds about 30 ft/sec, friction and associated wear result in reduced sealing effectiveness.

Shaft orbiting can pose problems for lip seals. If orbiting conditions are such that clearances develop between the shaft and lip seal leg, oil leakage can occur. Lip seals can only track the shaft to the extent of the flexibility of the sealing leg. In addition to developing leakage path clearances, orbiting can results in higher wear rates, and the need for more frequent replacement. In addition, any misalignment of the shaft will results in an asymmetric interference fit around the circumference of the seal.

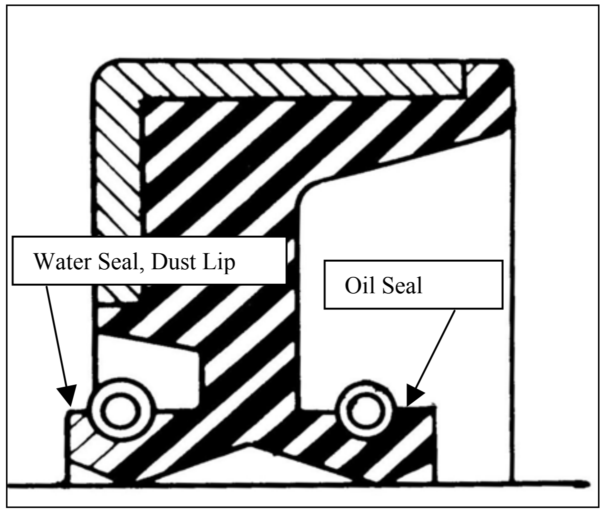

Figure 3 – Double Lip Seal (Buchter 1979)

THE FACE SEAL

The face seal design is widely used in industrial applications. Faces seals can be designed to be either contacting (rubbing) or non-contacting (film riding). Choice of a contacting or non-contacting depends of the demands of the application in question.

Design Features

In Figure 4, we can see the major components of a rubbing face seal. The face seal consists of two radial flat surfaces, with one rotating with the shaft and the other stationary with the machine housing. By an analysis of the forces on the system, the designer can ensure that the two surfaces will stay in contact over the range of machine operating conditions. Of critical importance is the flatness and surface finish of these mating surfaces. Flatness is held to 0.3 microns to 0.6 microns in many applications. Since the leakage is a cubic function of the clearance between these two surfaces, contact is critical.

In addition to a primary sealing surface between the carbon element and the mating ring, sealing also takes place by the secondary seal. In Figure 4, an o-ring is used for secondary sealing. Other types of secondary seals used include bellows and piston rings. While the secondary seal is sealing across two stationary surfaces, one should carefully design the secondary seal to prevent leakage. Being a sealing system, the successful operation of a face seals requires successful sealing for the secondary as well as the primary seal.

A face seal can perform successfully at higher rotational speeds. At high speeds, one must consider the inertia generated, and design to ensure contact of the primary sealing surfaces. In addition, rotational speed, as well as unit loading, affect wear rates and thereby seal life.

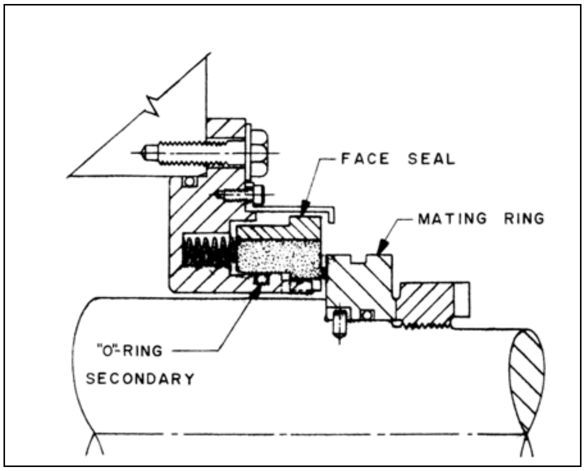

Figure 4 - Rubbing Face Seal with O-ring Secondary Seal

Hydrodynamic Face Seal

Hydrodynamic lift (or suction) can be generated between the primary sealing surfaces. If the case of high rotational speeds, creating a suction force will aid in keeping sealing surfaces in contact with one another. If high unit loading is developed, a lift force can lessen the unit loading, and therefore increase seal life. A face seal with hydrodynamic features is shown in Figure 5.

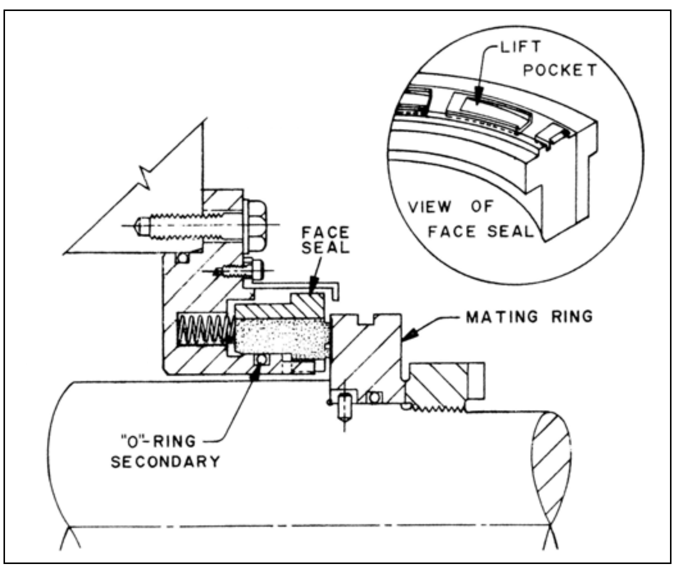

Figure 5 - Hydrodynamically Lifted Face Seal

Challenges in the Marine Environment

While a face seal can be split, it creates technical challenges to keeping the primary sealing surfaces flat. While flatness can be achieved, it is a challenging task which results in a complex design. Unlike the lip seal, it can perform well at high rotational speeds, but will wear, unlike a non-contacting seal. Due to the requirement of a split design for MRG applications, the design complexity of a face seal has prevented them from being used.

THE CIRCUMFERENTIAL SEAL

The circumferential seal is a contacting segmented device which has been used successfully in the MRG environment. It has performed satisfactory is cases where the labyrinth seal has failed. Being a contacting device, it shares many of the advantages over a labyrinth seal in common with the lip seal. It has a wider range of applicability than the lip seal, which make it suitable for use under many operating conditions.

Design Features

A typical circumferential seal is shown in Figure 6. The various components of the seal are highlighted. The segmented circumferential ring is held in contact with the shaft by a garter spring. Also, axial springs seat the ring along the sealing face of the housing. These spring forces are adjusted to assure contact throughout the operational range of the seal. The backplate can be used to anti-rotate the circumferential seal ring, while the retaining ring holds the assembly together. Anti-rotation locking can also be achieved using an anti-rotation feature on the sealing face of the housing.

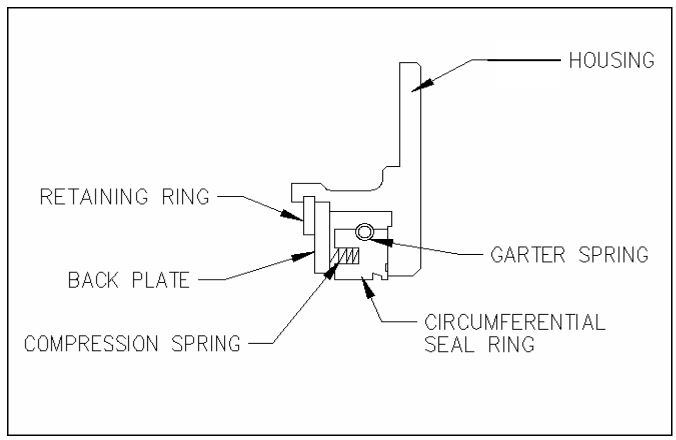

Figure 6 - Cross Section of Circumferential Seal

The segmented circumferential seal ring in Figure 6 has a number of features detailed in Figure 7. Since sealing takes place on two surfaces, the bore (dynamic) and the face (static), both surfaces must be carefully controlled. In addition, a close fit tongue and socket abates leakage through the joints. Bleed grooves in the bore and face of the circumferential seal ring provide pressure relief to the seal, lessening loading due to pressure differentials.

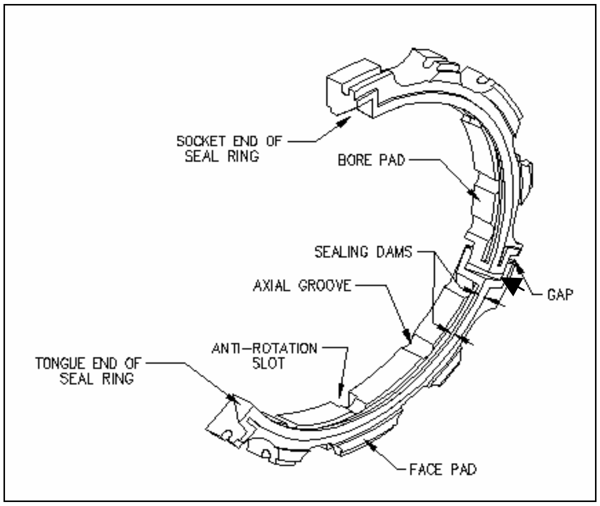

Figure 7 - Circumferential Seal Ring

The net loading on a circumferential seal is an important factor. If the loading is too low, unseating can occur, resulting in a clearance developing and increasing leakage. Likewise, if loading is too high, frictional effects will be high, resulting in high heat generation and high wear. This balance is critical to proper function.

When considering force balance of a circumferential seal, an important factor is shaft eccentricity. Since the inertia increases as the square of the shaft speed, at low speeds, this inertia effect is less important. However, at high speeds, this effect should be considered along with spring forces and pressure.

Figure 8 shows us the force balance on a circumferential seal, taking into account pressure and spring forces. Radial and axial forces are considered. One can see that the pressure relief grooves minimize pressure loading to the sealing dams, where pressure breakdown occurs. By controlling spring and pressure forces, and taking into account inertia and frictional effects, one can be confident in maintaining contact, and minimizing wear and friction.

High runouts, discussed as a concern for lip seals, are more easily handled by the circumferential seal. The garter spring around the segmented circumferential seal ring allows for tracking of the shaft and constant contact.

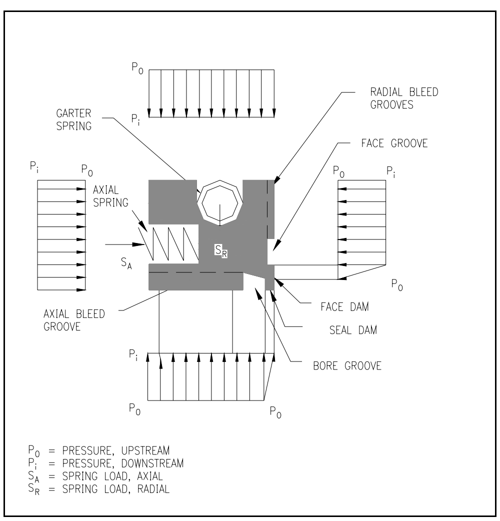

Figure 8 - Circumferential Seal Force Diagram

Power consumption

Especially for contacting sealing devices, a look at the power consumption should be considered. An approximation of the power consumption of a seal has been devised (Stein 1961). This quantity depends on both the frictional force generated, and the speed of the rotating shaft.

$$HP = \frac{F \cdot U}{550 \cdot 12}$$

Equation 2

where F is frictional force in pounds, and U is the surface speed in inches/second. This frictional force may be found by,

$$F = \frac{2 \cdot \pi \cdot R \cdot U \cdot L \cdot \mu}{h}$$

Equation 3

Where R is the shaft radius, L is the sealing width, μ is the viscosity in lb-sec/in.^{2}, and h is the clearance in inches.

Hydroload Concept

In MRG applications, the low differential pressure environment, where circumferential seal seating is produced mainly by the springs, “surf-boarding” of the segments on a film of oil can result in high oil leakage. Since increasing springs loads does not adequately solve this problem, the hydroload seal was developed as a solution (Stein 1978).

A hydroload seal makes use of negative lift pockets in the bore of the carbon. These pockets produce a downward force, preventing the formation of the oil film. Without the “surf-boarding” effect, hydroload seals have performed well in oil rich environments.

Hydroload Negative Lift Calculations

Analytic tools are available which calculate the amount of seating force generated by a given hydroload pocket (Stein 1978). Equations 4(a), 4(b), and 4(c) calculate the bearing force per unit width (W) for the three geometries in Figure 9 respectfully.

$$W = \frac{6 \cdot \mu \cdot U \cdot B^2}{\Delta h^2}\left(\ln\left(\frac{1+\alpha}{\alpha}\right) - \frac{2}{1 + 2 \cdot \alpha}\right)$$

Equation 4(a)

$$W = \frac{6 \cdot \mu \cdot U \cdot B^2}{\Delta h^2}\left[\frac{E \cdot (E + 2\alpha)}{2 \cdot \alpha(1 + 2\alpha)(1 - E)\left(E\left(2 + \alpha - 2\alpha^2\right) + \alpha(1 + 2\alpha)\right)} + \ln\left(\frac{1+\alpha}{\alpha}\right) - \frac{2}{1 + 2\alpha}\right]$$

Equation 4(b)

$$W = \frac{6 \cdot \mu \cdot U \cdot B^2}{\Delta h^2} \cdot \frac{E(1 - E)}{(1 - E) \cdot \alpha^3 + E(1 + \alpha)^3}$$

Equation 4(c)

Where α is the ratio of the entering clearance h_1 to the change in clearance Δh. Also of importance in Equation 4(b) and 4(c) is the E term. E is the ratio of the parallel part of B at height h_1 to the total length B. Of note in these equations is the relationship between the bearing force generated and the depth of the pocket. Since the bearing force is inversely proportional to the square of the pocket depth, a doubling of pocket depth will decrease the bearing force by a factor of four.

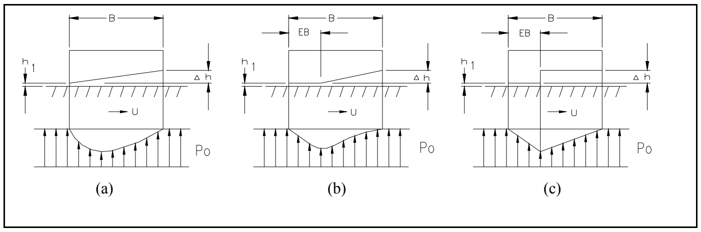

Figure 9 - Negative Lift Geometries

Figure 11 illustrates the geometry of a typical hydroload seal. Except for the bore, the seal does not differ from the standard circumferential seal. A detailed view of the bore is shown in Figure 10. The pocket pumps oil in the bore of the seal back into the MRG. Misty oil also has a tendency to condense in the pocket, preventing misty oil from escaping the gearbox. Unlike in clearance devices, like the labyrinth seal, oil cannot directly pass through a clearance opening. Since the negative lift generated does depend of the viscosity of the fluid in the bore, in a air rich environment, lift forces will be lower than in an oil rich environment. Hydroloads are unidirectional, and therefore assembly procedures are designed to ensure proper installation.

The force breakdown of a hydroload is the same for that of a standard circumferential seal, with the additional force generated by the hydroload pockets taken into account. Other forces, such as inertia and friction, must also be considered when developing a loading model of the seal in operation.

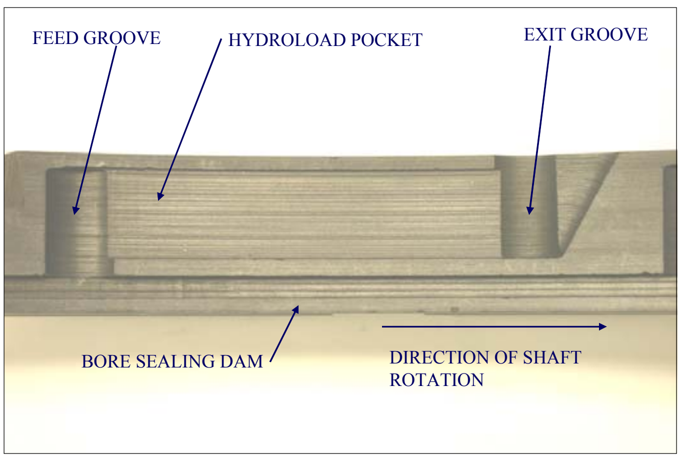

Figure 10 - Hydroload Seal Bore

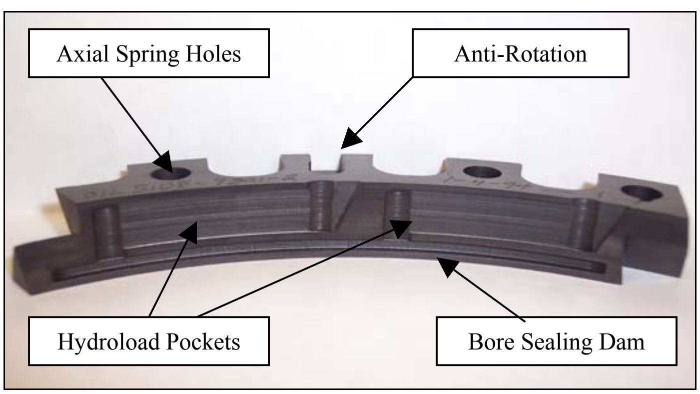

Figure 11 - Detailed view of Hydroload Seal Ring

The use of hydroload seals alleviate many of the concerns regarding surf-boarding associated with the standard circumferential seal. Seal assemblies employing hydroload pockets today are satisfactorily performing in a variety of MRG units.

In one case, leakage from a labyrinth seal was unacceptably large. This leakage was the supposed reason for several fires outside the gearbox. The labyrinth was replaced with a face to face hydroload sealing arrangement. As shown in Figure 13. One ring sealed oil while the second sealed debris and water in a flooding situation. This sealing replacement alleviated the leakage problems and associated fires observed with the labyrinth seal.

Operating Conditions and Performance

A wide range of operating conditions can be successfully handled with the hydroload circumferential seal. In one main drive shaft gearbox, an 8.5” sealing diameter seal has performed well at 200^{o}F maximum oil temperature, and 160 ft/sec (4320 rpm). Maximum allowable oil leakage in this example is 1.25 cc/hr. This is a horizontally split seal.

The presence of an oil rich environment allows for possible seal lives in excess of 20,000 hours. Throughout the lifetime of the seal, little to no reduction in sealing performance is observed. With respect to shaft runouts, 0.020” magnitudes can be tolerated. Depending on shaft speeds, larger runouts also can be effectively handled.

While the hydroload seal can effectively seal oil in the MRG, the placement of a second hydroload seal, face to face with the oil seal, allows for the possibility of sealing water and air contaminants as well. Where the labyrinth seal will draw oil out of the MRG under a reverse pressure, the face to face hydroload seal will continue to stay in contact with the shaft, and provide for continued sealing.

In addition to being able to preventing oil leakage during operation, the ability to seal oil when the shaft is not rotating is of importance. While the pockets actuate with shaft rotation, surf-boarding does not occur in the static case. Therefore, with surf-boarding absent, the contacting seal is effective at preventing leakage.

The Face to Face Hydroload seal

In Figure 12**,** we see a typical main reduction gear unit. This MRG makes use of a face to face hydroload seal at location D. A cross section of a similar seal is shown in Figure 13. This seal has an 8.5” sealing diameter, and mounts to the MRG by 18 5/8” bolts around its flange. Two hydroload seals are used, each sealing on two locations, the rotating bore and a housing face.

Our face to face seal assembly in Figure 13 has on the bottom vertical centerline a radial hole drilled through the seal plate, which aligns with a threaded hole on the lower half of the seal housing. Any liquid that collects between the two hydroload seals will drain through the hole to a tube installed in the lower half of the seal housing. In the case of small quantities of oil or water leakage, a canister will collect any liquid that passes through the tube.

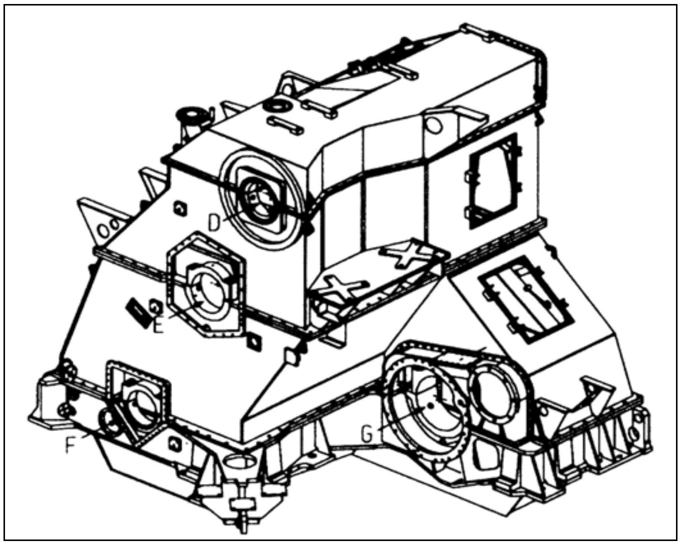

Figure 12 - Main Reduction Gear

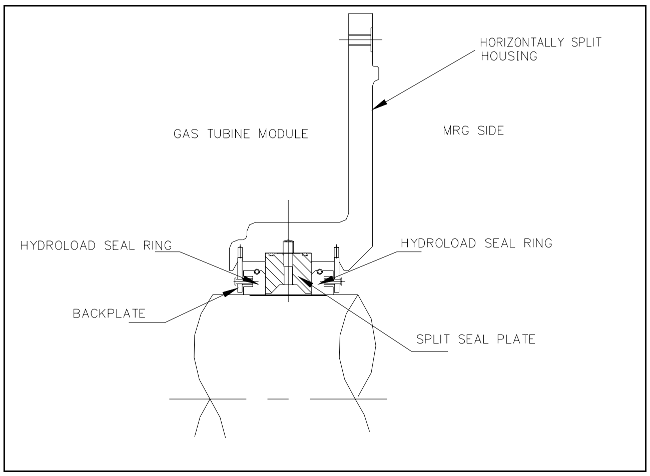

Figure 13 - Face to Face Hydroload Seal Assembly

Assembly / Disassembly

As will all seals, proper installation ensures proper functioning of the seal. While installing the seal assembly shown in Figure 13 requires some care, with attention, assembly is not a difficult process.

An assembly procedure is used for installation. During the assembly process, it is important that critical sealing surfaces do not develop nicks or scratches, and that all parts are clean and free from potentially damaging debris. Care must be used when handling the seal rings, as any damage incurred on them may affect sealing performance.

Assembly requires a number of steps due to the split nature of the seal. The horizontally split housing is seen in Figure 14. Two people are required for complete assembly. Scribe marks are located on the seal housing for ensuring proper alignment. After installing the split seal plate, in the correct direction, the two sets of hydroload rings are assembled. A service manual provided by seal manufacturers provides a step by step procedure on assembly.

Likewise, disassembly is done following a disassembly procedure. As with assembly, preventing damage to the seal rings is important. Disassembly involves removing the upper housing, the backplates, and then the lower half of the housing. Then, each set of seal rings are removed, and finally the seal plate.

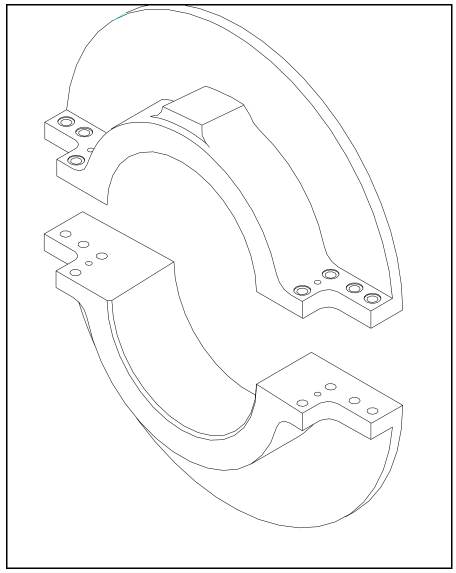

Figure 14 – Horizontally Split Seal Housing

Test Results

The seal assembly shown in Figure 13 was tested under a variety of conditions. The oil side seal was tested sealing air with oil mist as well as with solid oil. Oil leakage observed ranges from zero to 0.5 cc per hour when sealing oil laden air. In the case of oil flooding, the seal showed no leakage up to 4000 RPM and 35 psi. (Stein 1978)

The hydroload seal ring used for sealing water was tested at typical and higher water pressures one would expect. Table 1 shows the leakage rates found at various pressure differentials for a given speed. (Stein 1978)

Further testing has shown that expected leakage rates range from 0 cc/hr up to ~2 cc/hour, depending on the application. With these quantities of leakage, a small container connected to the drain hole provided in the seal assembly would be sufficient in collecting any escaped oil.

Water Pressure (psi)	Leakage Rate (cc/hr)
0	0
10	1.33
20	3.62
30	7

Table 1 - Water Leakage Rates at 1860 RPM (Stein 1978)

Shock Test Results

One seal requirement, especially in military applications, is the ability to survive a shock load without unacceptably affecting seal performance. High impact medium weight mechanical shock per MIL-S-901C Section 4.2.3.2 and 4.2.4.2 testing was performed on a face to face assembly such as that shown in Figure 13.

Two shock tests were performed; with the second including a modification of the seal rings with the intent to lower stress concentrations. Static air leakage rates were determined for the air side hydroload seal ring before and after each shock test. In addition to a Pre-test air leakage measurement, the oil side hydroload seal ring was tested for oil leakage before and after the shock was applied.

SHOCK TEST 1		
	Leakage Values in SCFM	
	Pre-Shock Test	Post-Shock Test
Air Side Hydroload	0.155	0.1375
Oil Side Hydroload	0.285	
	Leakage Values in Drops/Min	
	Pre-Shock Test	Post-Shock Test
Air Side Hydroload		
Oil Side Hydroload	0	0
SHOCK TEST 2		
	Leakage Values in SCFM	
	Pre-Shock Test	Post-Shock Test
Air Side Hydroload	0.155	0.105
Oil Side Hydroload	0.28	
	Leakage Values in Drops/Min	
	Pre-Shock Test	Post-Shock Test
Air Side Hydroload		
Oil Side Hydroload	0	0

Table 2 - Shock Test Results

In Table 2, the results of shock testing are shown. For each set of oil side hydroloads, no oil leakage was observed before or after the shock test. Surprisingly, the air side leakage decreased after the shock was applied. One possible reason for this is that the shock more firmly seated the rings, closing down any small gaps that were present in the per-shock arrangement. In addition to quantitative values, a visual inspection of the rings after testing revealed no apparent .

Challenges in a Marine Environment

Proper assembly of any seal is important for proper function. There is a danger of damaging the seal rings during assembly if attention and care are not used. By following a step by step procedure, assembly damage will be infrequent.

We have seen from test results that we can expect some small amounts of leakage from the seal. The addition of the housing drain hole and a collection canister are needed to capture this oil. The ability to manage this leakage into the canister has not been a major issue.

While the garter spring allows the ring to move freely and follow the shaft, a large amount of shaft movement can affect performance. Inertia effects increase as the square of shaft speed. If both of these variables are high, inertia effects must be considered to prevent unseating of the seal rings. Also a concern with high runouts is the possibility of shaft to housing rubs. While the seal can be designed to handle such cases, unexpected large shaft movements can result in a rub.

CONCLUSIONS

The selection of a suitable sealing device for a given MRG application requires analysis of a number of positives and negatives of the variety of sealing devices available today. In this paper, we discuss four types of seals, the labyrinth, lip, face, and circumferential seal. Each seal is presented, showing where its performance strengths lie, and its weaknesses. In the selection of a seal, one must weigh the advantages and disadvantages in each seal design. If a disadvantage is great, and cannot be managed, then the advantages of that seal will be negligible. The engineer, by considering the capabilities of each sealing device in the environment in question, can successfully select a seal which meets the performance requirements.

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