# Bulkhead Seals – A Sealing Solution

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ABSTRACT

*Typical Marine bulkhead seals today utilize a non-metallic seal ring of some kind to seal between the shaft and the seal housing mounted to the bulkhead. The sealing feature can occur continuously, or be actuated when needed. These arrangements can fail to work effectively in practice for a variety of reasons. This paper discusses segmented carbon seals as a viable sealing method. Test results are presented, as well as proposed mounting arrangements. A discussion of circumferential seals in general is included.*

KEY WORDS: Circumferential Seal; Bulkhead Seal; Lip Seal; Hydroload; O-ring; Mechanical Seal, Shaft Sealing, Elastomeric Seal

## INTRODUCTION

The bulkhead seal has the important task of preventing flooding from one ship compartment to another. Various methods have been employed to solve this engineering problem. As no sealing solution is perfect, one must weigh the positives and negatives of the seal selected. Particularly if the negatives cannot be managed effectively, other solutions must be sought.

A commonly used sealing method today makes use of an non-metallic seal of some kind. Typically, this seal is designed to actuate in the emergency flooding situation. During normal operation, the seal will not have contact between the rotating shaft and the stationary seal housing, and will not seal gases from passing from one compartment to another. Another method uses an inflatable elastomeric tube.

This paper discusses segmented circumferential carbon seals as alternatives to the present seals being used. Test results are presented, as well as the results of shock testing. The hydroload concept is discussed, as well as advantages and disadvantages of this kind of seal.

## SEALING REQUIREMENTS

The primary purpose of the bulkhead seal is to prevent flooding of one compartment from flooding an adjacent compartment. If the seal is ineffective, then the uncontained flooding can ultimately result in extensive damage, or the loss of the ship. In some applications, it may be necessary for possible forward motion of the in a flooding situation.

Generally the bulkhead seal will see speeds in the low to moderate range. In addition, a slight pressure differential (0.25 PSID), can be present in the normal operating condition. With the flooding condition however, pressure differentials can rise to 7-10 PSID. In addition to flooding, the seal may be required to withstand a shock load, as well as high heat. In the case of a fire, in one compartment, the seal will prevent the fire suppression agent from escaping into adjacent compartments.

One requirement of the bulkhead seal can be the sealing of gases as well as liquids (water in general). It is desirable to prevent any toxic gases from spreading throughout the ship.

The performance requirements must be satisfied even if the seal has undergone high heat, or a shock load. As the seals primary function is to be effective in an emergency condition, failure may not be apparent until a serious situation develops.

## THE LIP SEAL

The lip seal is a contacting device which prevents leakage by use of a flexible leg, held in place by a spring. In the normal condition, it can be either contacting or non-contacting. If non-contacting, the seal must actuate, and thereby contact, in the presence of flooding. One example of an actuated lip seal is one in which the seal is non-contacting, and then inflates during flooding. In this way, it prevents the leakage of water when needed.

### Design Features

Figure 1 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 1 for example, a dust prevention lip is added, possibly an important feature in a bulkhead seal 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 ensure contact, but not high enough to create high wear, and negatively affect function.

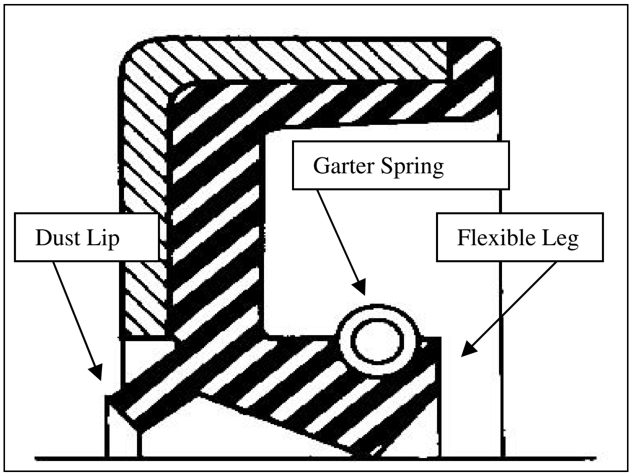


**Figure 1 – Lip Seal with Dust Prevention Lip ((add leaf spring) (Buchter 1979)**

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, $\mu$ the viscosity, $\Delta P$ 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.

The axial length required for a lip seal should be controlled to allow for assembly, considering the bulkhead seal location. For assembly and disassembly, the seal must be split. This should not affect seal performance when compared to a non-split seal assembly.

All of these design features make the lip seal a satisfactory solution to the bulkhead sealing application in certain cases.

The bulkhead seal 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.

The use of an elastomeric ring can be done also by inflation of the ring when flooding occurs. In this case, the bulkhead is not sealed during regular operation, but inflated only in an emergency flooding condition.. Of critical importance in this case is that the seal will actuate properly.

**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 low to moderate speed. In the case of the bulkhead seal, shafts speeds vary considerably. For this reason, the specific application should be considered as to whether or not a lip seal is the best solution.

Shaft orbiting can pose problems for lip seals. If orbiting conditions are such that clearances develop between the shaft and lip seal leg, water 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. Bulkhead Seals can undergone radial movement of 1.5” or greater. This poses a serious challenge to effective lip seal function. This level of misalignment can disqualify a lip seal as a viable sealing option.

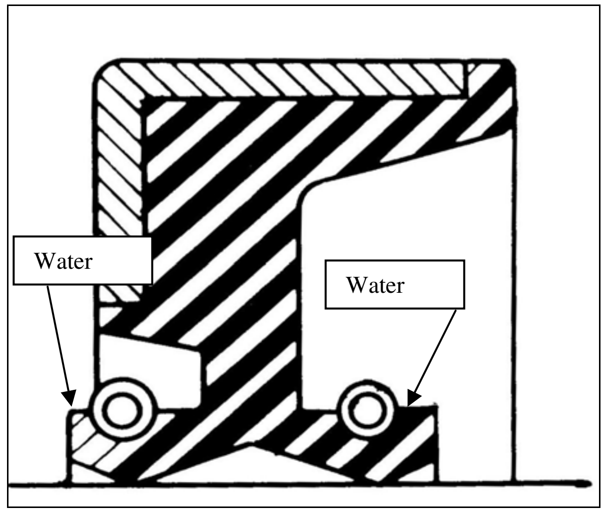


**Figure 2 – Double Lip Seal (Buchter 1979)**

## SELF ACTIVATING BULKHEAD

A commonly used bulkhead today is illustrated in Figure 1. In normal operation, the diaphragm and non-metallic seal ring spin with the shaft. Gases are allowed through the seal. When flooding occurs, the diaphragm is actuated to one side of the housing, and the non-metallic seal ring, along with the diaphragm, seals water.

**Challenges in the Marine Environment**

Being a seal which requires actuation, it is essential that the seal actuates at the correct pressure (general 7-10 PSID). If it actuates prior to this pressure, due to a pressure imbalance between compartments for example, the diaphragm may not rotate with the shaft, due to friction with the housing. In this case, the non-metallic seal ring can be damaged. Damage to the ring would not be obviously known until a failure during flooding.

In addition to the chance of non-metallic ring damage due to non rotation of the diaphragm, this seal does not seal gases. The ability to seal fire suppression agents, along with other non-desirable chemicals may be important.

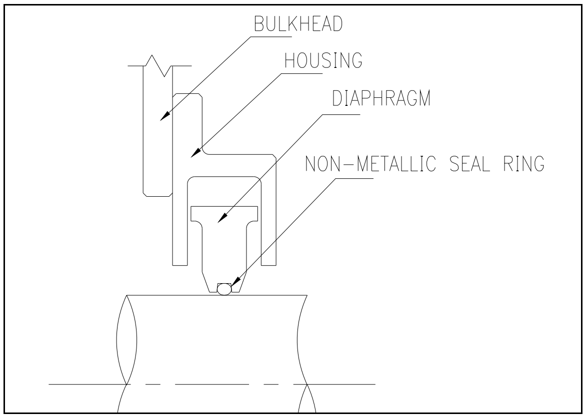


**Figure 3 - Self Activating Bulkhead Seal**

## THE CIRCUMFERENTIAL SEAL

The circumferential seal is a contacting segmented device which has been successfully employed to seal both liquids and gases. It has a wider range of applicability than the lip seal, particularly with respect to speeds, which make it suitable for use under many operating conditions.

**Design Features**

A typical circumferential seal is shown in Figure 4. 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.

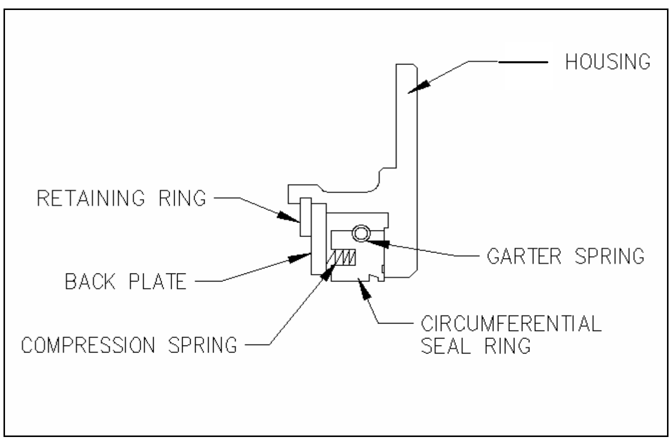


**Figure 4 - Cross Section of Circumferential Seal**

The segmented circumferential seal ring in Figure 4 has a number of features detailed in Figure 55. 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.

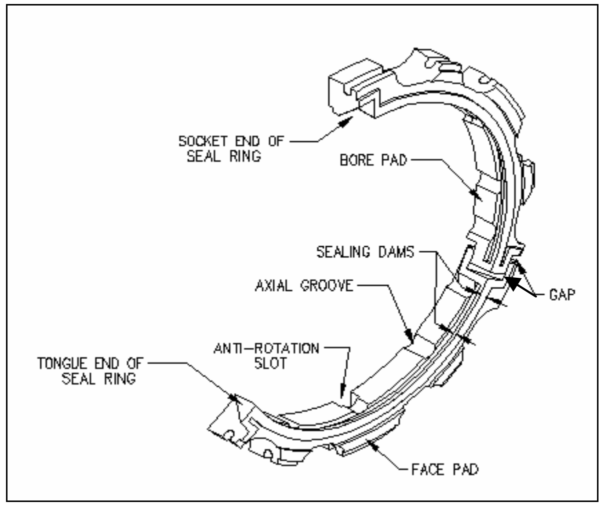


**Figure 5 - Circumferential Seal Ring**

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 increased 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 6 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, 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.

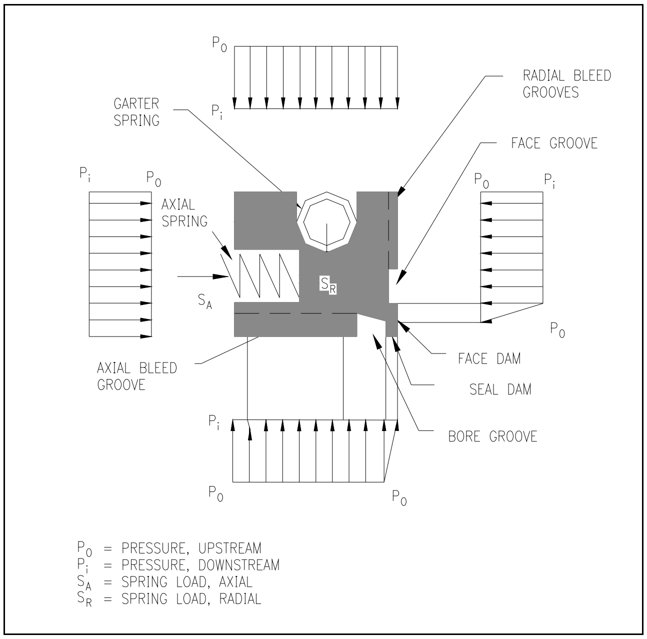


**Figure 6 - Circumferential Seal Force Diagram**

### *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, $\mu$ is the viscosity in lb-sec/in.$^{2}$ pf the liquid or gas being sealed. and h is the clearance in inches.

### *Hydroload Concept*

In a low differential pressure environment, where seating is produced mainly by the springs, "surf-boarding" of the segments on a film of liquid can result in high 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 liquid film. Without the "surf-boarding" effect, hydroload seals have performed well in incompressible fluid 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 7 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 is the ratio of the entering clearance $h_1$ to the change in clearance $\Delta h$. 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.

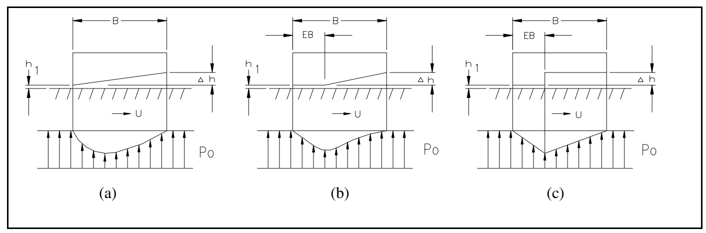


**Figure 7 - Negative Lift Geometries**

Figure 9 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 9. The pocket pumps liquid back into water flooded compartment. In addition to sealing liquids, the hydroload seal can seal gases as well. Hydroload seals 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.

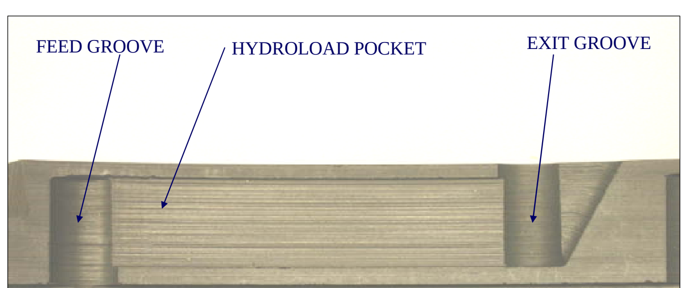

Figure 8 - Hydroload Seal Bore

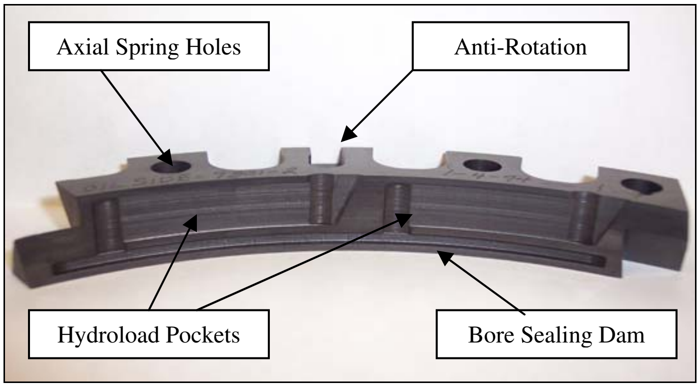


**Figure 9 - Detailed view of Hydroload Seal Ring**

The use of hydroload seals alleviate many of the concerns regarding surf-boarding associated with the standard circumferential seal. Without the surfboarding effects, the effective clearance will be on a small order, and result in minimal leakage.

**The Face to Face Bulkhead Circumferential Seal**

In Figure 10, we see a possible bulkhead seal configuration making use of the hydroload circumferential seal ring. The assembly includes debris wipers, as well as two hydroload circumferential seal rings. In addition to sealing water in a flooded environment, the circumferential seals will seal gas throughout its life. Unlike the self-activating seals discussed earlier, this seal does not require activation. It seals liquids and gases throughout its lifetime.

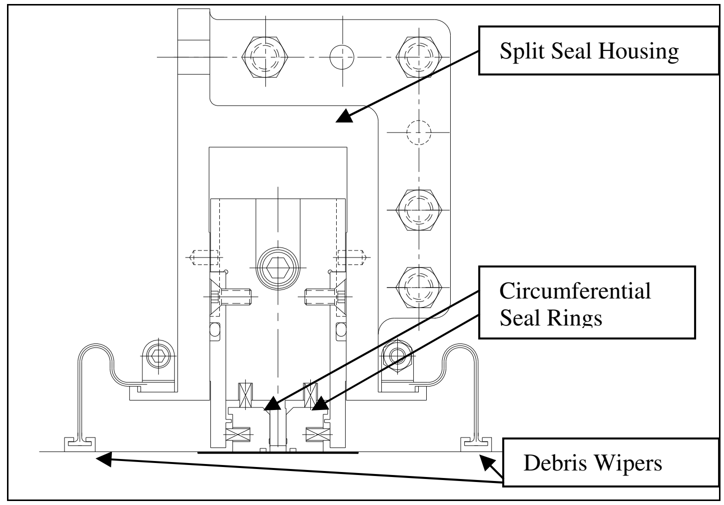


**Figure 10 –Possible Bulkhead Seal Mounting Arrangement**

## *Assembly / Disassembly*

As will all seals, proper installation ensures proper functioning of the seal. While installing the seal assembly shown in Figure 10 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 free from potentially damaging debris. Care must be used when handling the seal rings, as any damage incurred on them will affect sealing performance.

Assembly requires a number of steps due to the split nature of the seal. A service manual is provided by the seal manufacturer.

Likewise, disassembly is done following a disassembly procedure. As with assembly, preventing damage to the seal rings is important. In addition, general cleanliness should be followed by cleaning each part as it is removed.

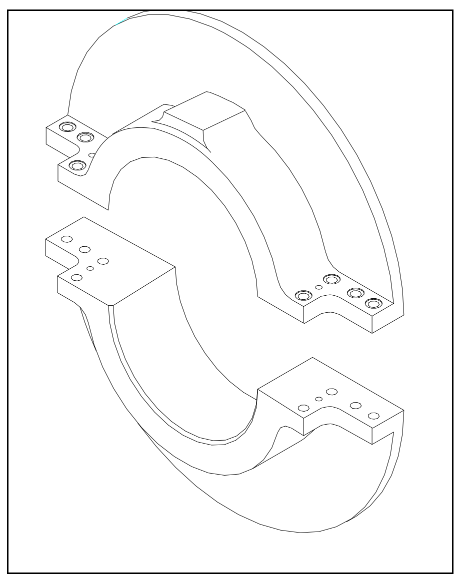

**Figure 11 – Horizontally Split Seal Housing**

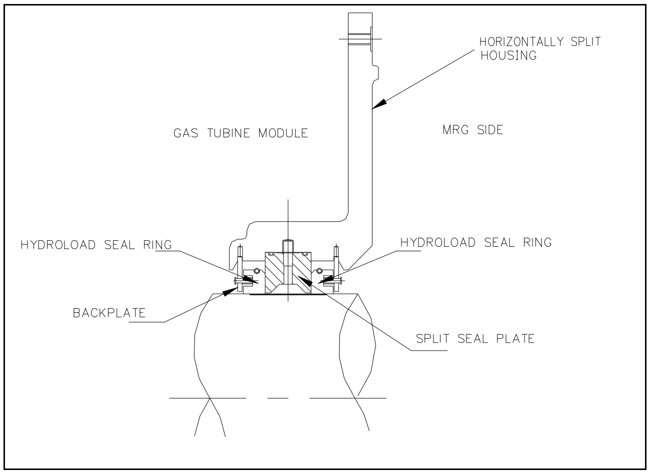


Figure 12 – Horizontally Split Seal Housing (8.7” Sealing Diameter)

**Test Rests**

A hydroload circumferential seal ring, shown in Figure 12, 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 at ~2 cc/hour, depending on the application. These quantities are usually not considered excessive.

| 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 12.

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 one hydroload seal ring, while oil leakage rates were determined for the other.

| | | |
|---|---|---|
| **SHOCK TEST 1** | | |
| | Leakage Values in SCFM | |
| | Pre-Shock Test | Post-Shock Test |
| Hydroload #1 | 0.155 | 0.1375 |
| Hydroload #2 | 0.285 | |
| | Leakage Values in Drops/Min | |
| | Pre-Shock Test | Post-Shock Test |
| Hydroload #1 | | |
| Hydroload #2 | 0 | 0 |
| **SHOCK TEST 2** | | |
| | Leakage Values in SCFM | |
| | Pre-Shock Test | Post-Shock Test |
| Hydroload #1 | 0.155 | 0.105 |
| Hydroload #2 | 0.28 | |
| | Leakage Values in Drops/Min | |
| | Pre-Shock Test | Post-Shock Test |
| Hydroload #1 | | |
| Hydroload #2 | 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 damage to the seal parts.

**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 od leakage from the seal. Leakage of this magnitude is usually not considered excessive, and can be readily handled.

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 shaft movement can results in a rub.

## CONCLUSIONS

The selection of a suitable sealing device for a given bulkhead seal application requires analysis of a number of positives and negatives of the variety of sealing devices available today. In this paper, we discuss various seals for use. 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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