Radial Segmented Main Shaft Seal For Hydro Turbines

by Guojun Ren, P. Eng and Keith Laskey, P. Eng, Thordon Bearings Inc., Ontario, Canada

Abstract

Reliable shaft sealing is an important operator concern for large hydro turbine shafts from 400 to 2000+ mm (16 to 80 in.). Typical turbine seals are constructed of segmented carbon rings stacked in stages, usually three. Carbon seal rings are formed of very brittle segments and are subject to relatively short life if misaligned or subjected to abrasives.

Thordon Bearings Inc. has embarked on a program to optimise designs of such seals utilizing its proprietary SXL elastomeric material. SXL is formulated from thermosetting resins which form a very hard tough synthetic polymer alloy. This polymer has performance characteristics superior to most other typical bearing and seal materials. SXL has high natural abrasion resistance, and is well suited for such sealing applications.

Typical radial segmented seals comprise three stages of interlocking segments. Each ring has both dynamic (against the shaft) and static (against the housing) sealing sections. First stage seals are normally reversed compared with the two upper rings, to allow introduction of a higher-pressure injection flow between the first and second rings. This higher pressure flow (1.10 to 1.15 turbine pressure) functions as a seal lubricant, coolant and a barrier preventing abrasives from entering the seal faces. A garter spring functions to hold the segments with a nominal light force against the shaft during periods of shutdown and low pressure and also to maintain the integrity of the seal ring within the housing cavity.

A dedicated test rig has been constructed for this development. The rig has an effective shaft diameter of 400 mm (16 in.) and is capable of operating at shaft peripheral speeds to 25 m/s (82 ft/s).

This paper outlines the basic design concepts of radial segmented main shaft seals for hydro turbines. It forms a discussion document capturing the state of the art as applied to Thordon materials in use as radial segmented seals in hydro turbine applications. It calls on the current engineering practice and field experience of Thordon Bearings Inc. in the application and use of such seals.

Introduction

This paper outlines the basic design concepts of radial segmented main shaft seals for hydro turbines. The material is based on Thordon Bearings (TBI) experience, current engineering practice and in-house test results, albeit as yet limited. The paper is an attempt to capture the state of the art as it applies to Thordon material. It is not intended to be restrictive of any new ideas or improvements by users; in fact such constructive critiques are welcomed. The paper deals with only radial segmented seals. Axial seals will likely be the subject of a future paper for presentation in a similar forum.

A seal for hydro turbine shafting is in general a misnomer. The "seal" is really a controlled leakage device. These devices may be applied with the dynamic sealing face either parallel to the shaft (radial) or perpendicular to the shaft (axial). The need to control leakage at the point of pressure boundary penetration by the shaft is axiomatic.

Two basic designs are presently in use and are being evaluated at the TBI test facility. The physical difference can only be observed when the ring is assembled. In the first case, contact type, there is a circumferential gap between the segments when the pieces contact the shaft. Higher turbine pressures mean higher contact pressures. This type allows the segments to adjust as the contact surface wears.

Alternatively, for the floating ring type of seal, the ID will match the shaft OD and the segment ends will contact allowing no radial adjustment for wear. One clear advantage of this type of seal is that higher turbine pressures do not increase the shaft contact pressures. It is surmised that some initial wear will occur until a stable clearance and leakage rate is established. At this point, the seal should provide many years of satisfactory service providing the status quo is not significantly disturbed by outside influences (i.e. excessive vibration, temperature, or abrasives).

1.0 Application Information and Design Parameters

The information and design parameters listed below are essential and have to be taken into consideration before starting design of the system.

1.1 Information required from site:

- Water pressure being sealed (water pressure inside turbine)
- Shaft diameter
- Shaft rotating speed (rpm)
- Water quality (clean or abrasive)
- Environmental (water) temperature
- Maximum Leakage flow allowed
- Turbine Arrangement (Vertical or Horizontal)

1.2 Design parameters to be determined:

- Interface pressure between shaft and seal segment face (contact type)
- Garter spring force
- Injection water flow rate
- Injection water pressure
- Maximum allowed leakage

1.3 Mating shaft and housing requirements

- Shaft (sleeve) surface finish: 0.4 to 0.8 μm (16 to 32 μin)
- Housing cover finish: 0.8 to 1.6 μm (32 to 63 μin)
- Radial shaft dislocation (vibration amplitude) $\pm$1.5 mm ($\pm$0.06 in)

2.0 General Arrangement

2.1 Basic Elements of system

Figure-1 represents a typical segmented main shaft seal system. It consists of

- Thordon SXL segmented seal rings
- Press plates (Spring Adapters)
- Garter springs
- Injection water inlet
- Anti-rotation pins
- Drainage
- Housing covers

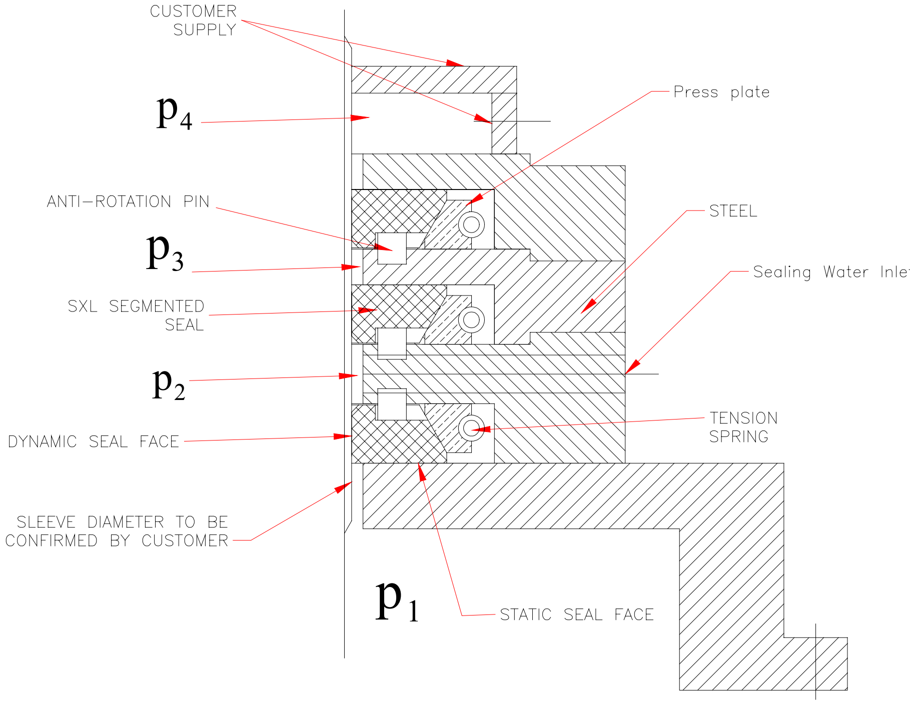

Figure-1

Above assembly provides the following functioning elements (ref Figure-1):

- 5 chambers with differing pressures p_1, p_2, p_3, p_4. Note p_1 is turbine internal pressure, and p_4 is drainage pressure.
- 3 dynamic sealing faces (1 per ring). These are the main sealing faces between shaft and segmented seal rings.
- 3 static sealing faces, which are a static contact interface between housing (covers) and segmented seal rings.

2.2 Arrangement

In accordance with the turbine design, the seal ring may be arranged vertically or horizontally and typically may be constructed of a 2 or 3 ring arrangement based on the pressure breakdown required.

2.2.1 Vertical Arrangement

The standard radial segmented seal ring is well suited to the typical vertical turbine shaft. Figure-1 presents an example of vertical turbine shaft arrangement.

2.2.2 Horizontal Arrangement

Currently, there is not much experience available on horizontal applications. However, the basic seal design can be readily adapted for use in horizontal turbines. Figure-2 shows a typical example of a horizontal arrangement.

2.2.3 3-ring Arrangement

In a 3-ring arrangement, the first ring provides pressure breakdown between injection pressure (p_2) and turbine pressure (p_1) where $p_2 > p_1$. The second and third seal rings provide pressure breakdown between injection and drain pressures.

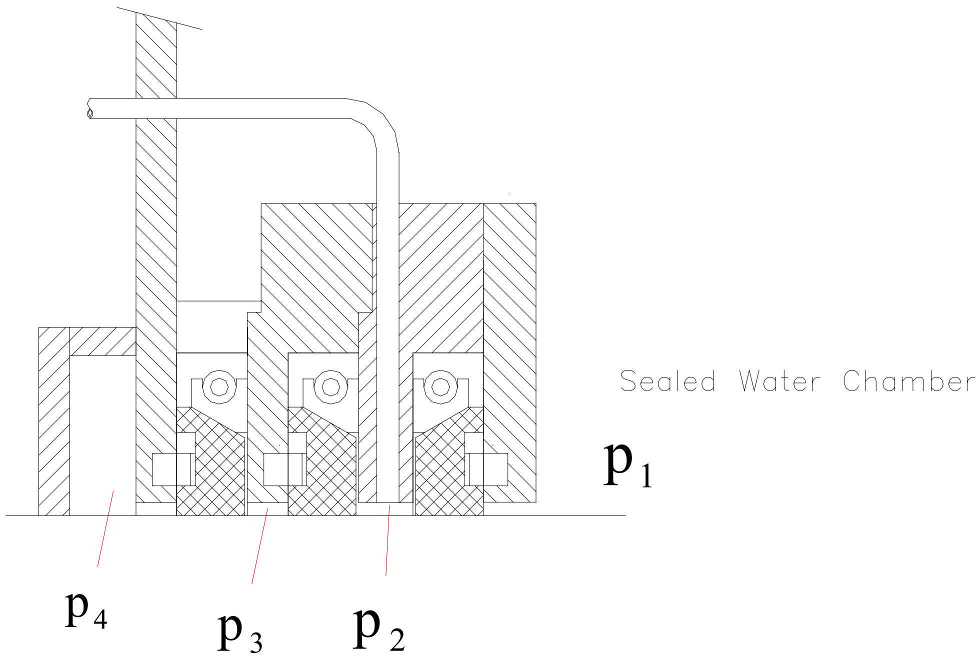

Figure-2

2.2.4 2-ring Arrangement

For a low pressure application, it is possible that two rings may be adequate for the sealing function.

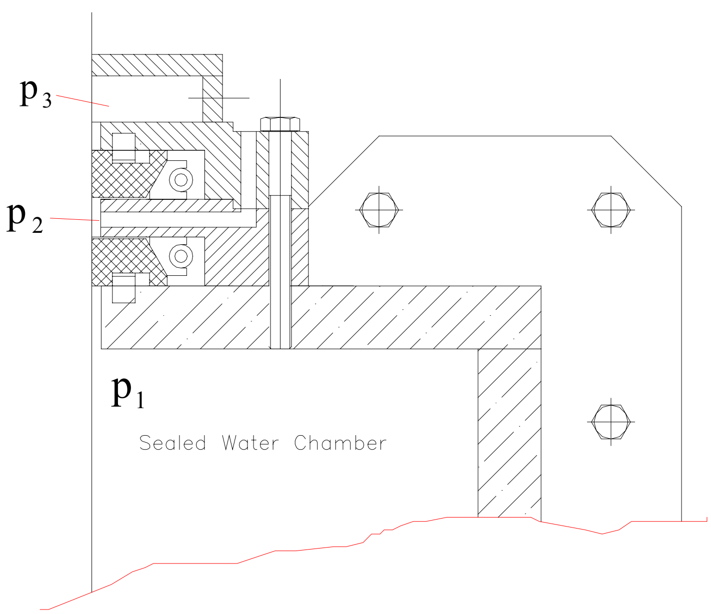

Figure-3

3.0 Design Parameters

The design parameters will be determined based on calculation and/or experience. These are the interface pressure between shaft and seal segment face, garter spring force, injection (cooling) water flow rate, injection water chamber pressure, and maximum allowed leakage.

3.1 Contact Pressure between shaft and seal segment face

A lower contact interface pressure will produce less heat leading to longer seal life in terms of diametric wear. The static calculation method is included in Appendix I. When the shaft is turning, a hydrodynamic film pressure is expected to be developed in the interface. This pressure will counter the resultant static pressure loads applied by the injection water.

3.2 Garter Spring Force

The function of the garter spring is to hold the segments against the shaft in the ring formation for assembly purposes and during idle periods when effective pressures may be reduced. As the surrounding pressures essentially activate the seal, spring tension can be quite low, in the order of 50 to100 N (11 to 23 lb.). This will allow the garter spring to perform its function and have little detrimental effect on the performance of the seal. A calculation method for establishing the garter spring cut length and stiffness is given in Appendix II.

3.3 Injection Water Flow Rate

The required flow rate for injected cooling water can be estimated based on the generated heat on the dynamic sealing face. However, experience and experiment shows that even though there is sufficient cooling water supply, it may still burnish the seal segments. This is considered normal provided temperature control is maintained and the effective wear rate of the material is acceptable. Improved designs may include grooving on the seal face. Currently, for seal face height greater than 25 mm (1 in), a circumferential groove is included in the contact face. The cooling water may be directly connected to these grooves.

An estimate of the required water flow for the seal to work properly is estimated using the following formula for adiabatic conditions:

$$Q_w = \frac{P_f}{c_w \cdot \rho \cdot (t - t_0)} \qquad (1)$$

Where:

P_f- Friction power loss.

$$P_f = n \cdot p_{ICF} \cdot h \cdot D \cdot \pi \cdot V_{surface} \times CoF \qquad (2)$$

$V_{surface}$-Shaft surface speed (m/s)

$$V_{surface} = \frac{\pi D \cdot RPM}{60}$$

D-shaft diameter (m)

CoF = coefficient of friction, use 0.01 (conservative for hydrodynamic conditions)

h-Height of the seal segment face (mm),

p_{ICF}-Interface pressure on dynamic sealing face (MPa)

n- The number of sealing rings.

c_w- Specific heat of water (4180 Nm / kg-°C)

ρ - Density of water, ρ = 1000 kg / m^3.

$t - t_o$ Temperature rise allowance (10 degrees).

3.4 Injection Water Pressure

The injected cooling water (or sealing water) provides a triple function for the system. Initially, it cools the segment and protects it from overheating. Secondly, it provides lubrication of the dynamic sealing face and a tertiary function is the provision of an anti-water flow to the first sealing ring to preclude abrasives within the turbine water from entering sealing chambers. The recommended injected water pressure is 10 to 15% higher than the internal water pressure of the turbine.

3.5 Injection Water Quality

It is strongly recommended that the cooling water be clean. In case there is no clean water available, filtration to approximately 100 microns is suggested. Lower particle size is better, however the practicalities of providing extra filtration may be prohibitive.

3.6 Maximum Allowed Leakage

The most difficult parameter to set is the leakage. There is little information available as yet from our test program for estimating leakage. As the sealing ring is segmented, there are a number of leakage paths available although the segment tolerancing is set to minimize this effect. In Appendix III attached, further discussion is provided regarding available leakage paths.

4.0 Detail Element Design

4.1 The Seal Segment

The standard design of a seal segment takes the shape shown in Figure-4.

4.1.1 Length, Height, Width

The number of segments that make up one seal ring depends on the shaft diameter and segment length. Dimensional integrity is increased with shorter segments but also increases the number of potential leakage paths. In general, longer segments will be subject to larger distortion under load. As an approach to standardization for new turbines, the recommended segment length is 500 mm (20 in) as maximum. Shorter segments are preferable. In addition, experience has shown that a good working height for segments is 25 mm (1 in) with section width of about 40 mm (1.5 in). Historically, sizing of segments have been dictated by retrofit requirements. However, there are successful turbine applications using larger segments than that suggested above. Therefore, Thordon continues to be flexible in the sizing requirements for each application. Segment height is typically about 1.0 mm (0.04 in) narrower than the chamber height. Based on the above recommended section sizes, a thermal and water swell of 0.25 mm (0.01 in) is anticipated leaving 0.75 mm (0.03 in) as the effective clearance. This allows water to flow freely behind the seal to maintain the water pressure in the chamber.

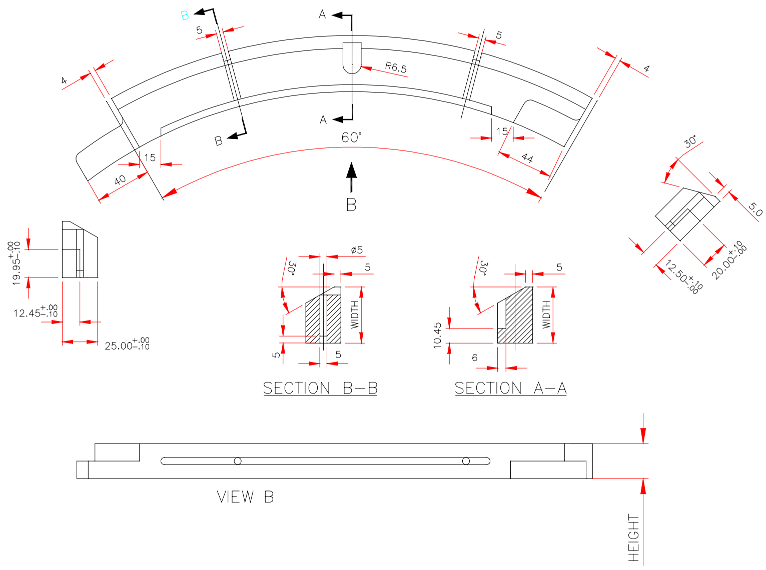

Figure-4

4.1.2 Taper on Segment OD
The taper on outside diameter is to accommodate the press plate (or spring adapter). A 30° angle measured from the shaft line is recommended.

4.1.3 Groove on interface
As noted previously, interface grooving is recommended for all seal face heights greater than 25 mm (1 in). Field experience has indicated that for face height greater that 25 mm without grooving the face area has been overheated and damaged. The typical width of groove is 5 mm (0.2 in) and the depth should be 5 mm minimum. The length of groove will depend on the segment length but will typically end prior the end of the segment. The number of water supply holes is determined according to segment length.

4.1.4 Joints of the segment
The overlapped convex and concave portion of the tongue and groove at each end does not provide a direct leakage path from bottom to top when the segments joint together. The leakage path is the side gap between the convex and concave surfaces. For the contact design, a radial movement of segment must be allowed to compensate for the wear down amount at the sealing interface during service. To do this, the segment ends must leave the appropriate gap to allow radial movement of the segment toward the shaft.

For this seal design, the size of the gaps between the ends of seal segments at new installation will depend on three factors: wear down allowance, thermal expansion of seal segments, and water swell of seal segments.

- A minimum 5 mm (0.2 in) diameter wear down allowance should be taken into consideration at the design stage.
- Thermal expansion should be based on maximum possible operating temperature of water around the seal segments.
- Water swell should be based on maximum operating temperature of water around the seal segments.

For the non-contact or floating ring type design, the water swell characteristics will allow slightly greater leakage over time but should not change substantially after the first 6 to12 months of operation. Thermal expansion will be self-compensating. If temperature rises due to friction, more leakage will pass tending to lower the seal face temperature keeping the flow in check.

4.1.5 Anti-rotation Pin

The position for anti-rotation pins may be arranged on either bottom or top sides depending on seal housing design.

4.1.6 Production Tolerance

Good conformance between the seal ring face and shaft is a fundamental requirement for sealing. Also, the production tolerance used by Thordon for the tongue and groove joints between the segments have been selected to ensure controlled leakage paths.

4.2 Anti-rotation Pin

A ϕ12 mm (½ in) stainless steel anti-rotation pin is recommended. A suggested grade is 17-4 PH with minimum hardness of 40 Rc.

4.3 Press Plate (Garter Spring adapter)

The press plate should be well conformed to the cone-form surface of the seal ring. The gap between segments of the press plate should allow sufficient room for the seal ring to move radially. The radial groove on the press plate face provides a channel for water entering the chamber at the back of seal for balancing the internal pressure of turbine. A lightweight material is strongly recommended (Thordon SXL or Regular).

4.4 Injection Water Pipe

When external injection water is supplied, it is recommended to supply one pipe of size 25 mm (1 in) per segment.

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APPENDIX I
Calculation of the seal interface pressure

A1-1.0 Force on seal segment

The pressure distributions on the first (lowest) seal segment from Figure-1 is presented in Figure-A1-1.

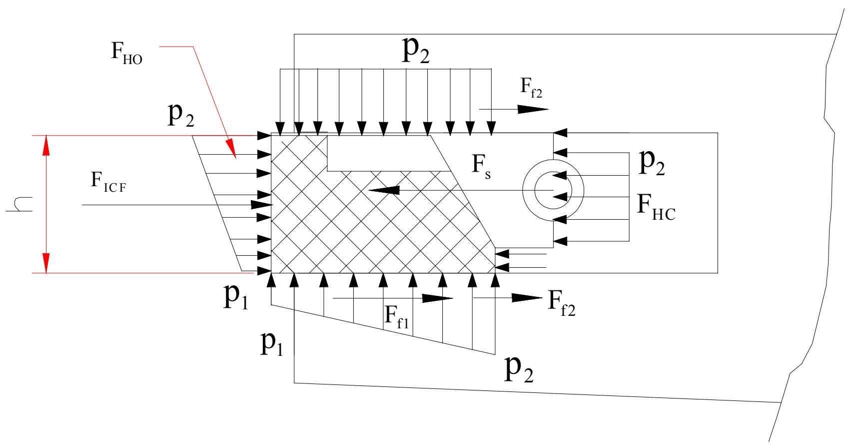

Figure-A1-1

Since the cooling water pressure is in general 10-15% higher than the water pressure inside the turbine, the bottom ring is subjected to a static hydraulic opening pressure from the higher pressure p_2 to lower pressure p_1. In addition, because of the pressure gradients on both the static and dynamic surfaces, there will be opposing moments applied to the segment.

Viewing the segment from top, the force distribution is presented in Figure-A1-2, which shows the forces on one of the segments.

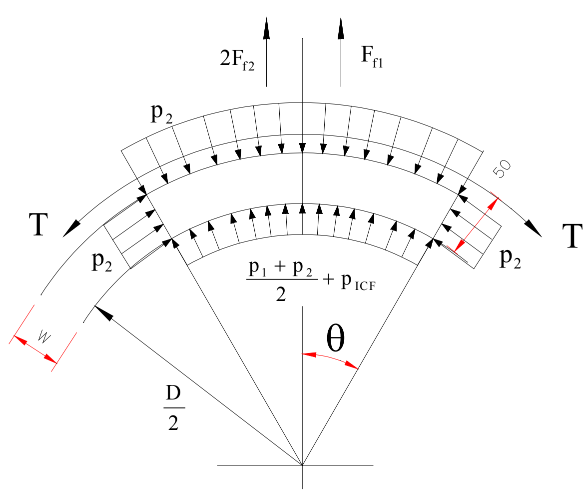

Figure-A1-2

In Figure-A1-1 & Figure-A1-2, the notations are:
F_{HC} - Hydraulic closing force, which tends to close the dynamic face gap. Considering one seal segment, this force is given by (ref Figure-A1-2):

$$F_{HC} = \left(\frac{D}{2} + W\right)h \cdot p_2 \cdot 2\sin\theta \qquad \text{(A1-1)}$$

F_{HO} - Hydraulic opening force. This force tends to open the gap between shaft and seal segment. It can be calculated with:

$$F_{HO} = \frac{p_2 + p_1}{2} \cdot D \cdot h \cdot \sin\theta + F_{End} \qquad \text{(A1-2)}$$

F_{End} is the end force which provides an additional opening effect acting against the projected area of the end face. Specifically, P_1 acts against the projected tongue area (A_T) and P_2 acts against the projected end area less the tongue area ($A_{End} - A_T$).

$$F_{End} = P_1 \cdot A_T \cdot 2\sin\theta + P_2(A_{End} - A_T) \cdot 2\sin\theta \qquad \text{(A1-3)}$$

In general, the end force is small since the projected area is small and can be neglected.

F_{ICf} -Seal Interface contact force. Since the hydraulic closing force (A1-1) and opening force (A1-2) do not balance, the shaft must impose an additional force to the segments in order to maintain force equilibrium. The seal interface contact force will be the difference between the closing and opening forces. The total opening force is $F_{HO} + F_{ICF}$.

$$F_{HO} + F_{ICF} = \left(\frac{p_2 + p_1}{2} + p_{ICF}\right) \cdot D \cdot h \cdot \sin\theta + F_{End} \qquad \text{(A1-4)}$$

Here, p_{ICF} is the interface pressure between shaft and seal segment (solid contact, static conditions). For dynamic conditions, this pressure will be a hydrodynamic film pressure developed in the interface between shaft and seal face. The interface pressure is important when designing segmented sealing rings because it is the force which determines the heat generation and diametric wear of the ring.

F_{f1} is the friction force developed at the static seal interface between the seal ring and the housing. The effective force on the seal is due to the pressure unbalance in the vertical direction as shown in Fig A1-1. F_{f2} is the friction force caused by the axial component of garter spring force F_s. As the tension in the garter spring is minimal, the resultant friction forces developed are small and will be neglected here. The friction force will always oppose motion of the part. The friction force can be calculated by

$$F_f = 0.3 \cdot \theta \cdot (D + W)W \frac{p_2 - p_1}{2} \qquad \text{(A1-5)}$$

where the static friction co-efficient has been taken as 0.3.

A force balance gives

$$F_{ICF} + F_{HO} \pm F_f = F_{HC} + F_s \qquad \text{(A1-6)}$$

The interface pressure can be expressed as :

$$p_{ICF} = \frac{(F_{HC} - F_{HO} \mp F_f)}{h \cdot D \cdot \sin\theta} \qquad \text{(A1-7)}$$

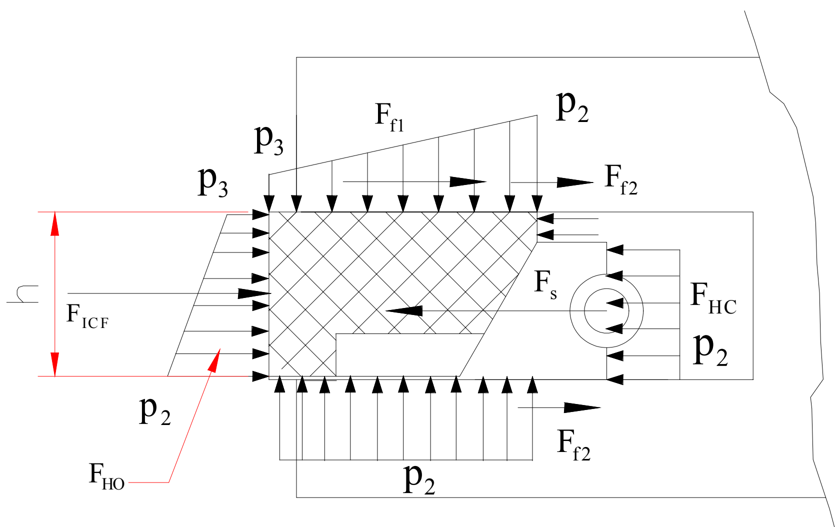

Figure-A1-4

The section view of second sealing ring is a mirror image of the first ring as illustrated in Figure-A1-4.

The calculation can be simplified by means of replacing the pressure p_1 with p_3 in previous equations. As before, the hydraulic closing force is

$$F_{HC} = \left(\frac{D}{2} + W\right) h \cdot p_2 \cdot 2\sin\theta \qquad \text{(A1-9)}$$

and the total opening force is

$$F_{HO} + F_{ICF} = \left(\frac{p_2 + p_3}{2} + p_{ICF}\right) \cdot D \cdot h \cdot \sin\theta + F_{End} \qquad \text{(A1-10)}$$

Similarly, the friction force is given by

$$F_f = 0.3 \cdot \theta \cdot (D + W) W \cdot \frac{p_2 - p_3}{2} \qquad \text{(A1-11)}$$

APPENDIX II

A2-1.0 Determination of spring length

A2-1.1 Material and size of spring, spring rate

In the general case, it is recommended to choose 50 to 100 N (11 to 23 lb.) as the desired spring tension. The typical selection of spring is type 302 stainless steel material with spring section of 12.7 mm (½ in) diameter. There are two sizes of spring wire diameter available, 1.37 mm (0.054 in) and 1.83 mm (0.072 in). For continuous-length extension spring, the spring rate is calculated as below.

The spring rate defined as

$$Spring\,Rate = \frac{Force(N)}{Spring\,stretch(mm)}$$

Since so defined spring rate will depend on the actual length of spring used, product catalogue provides a so-called spring constant that is independent on the spring length.
The spring constants for the 12.7 mm spring diameter for the two available wire sizes are given in the following table:

Spring Diameter	Spring Constant		Wire Diameter		Coils / mm	Coils / in
	lbs. x coils / inch	N x coils/ mm	mm	in		
12.7 mm (0.5 in)	119.8	21	1.37	0.054	0.73	18.5
	428.5	75	1.83	0.072	0.55	13.9

The spring rate then is calculated by:

$$spring\,rate = \frac{Spring\,Constant}{Number\,of\,Coils} = \frac{Spring\,Constant}{Coils/mm \times Length}$$

A2-1.2 Spring Length

The minimum spring length after stretching is calculated with

$$\text{Stretched Spring Length} = \pi \cdot (\text{D} + 2 \times \text{segment width})$$

Using the following example data:

spring force = 100 N (22.4 lbs.)
shaft diameter = 600 mm (23.6 in)
segment width = 40 mm (1.57 in)

The stretched spring length = π (600 +80) = 2135 mm (84 in).

$$Cut\,Length = \frac{Spring\,Constant}{Spring\,Constant + coils/mm \times 100N} \times stretched\,length$$

$$CutLength = \frac{75}{75 + 0.55 \times 100} \times 2135$$ = 1232 mm (48.5 in)

And the spring rate:

$$springrate = \frac{75}{0.55 \times 1232} = 0.11 N / mm$$ (0.63 lb./in)

APPENDIX III
Possible Leakage Sources

Being able to predict the amount of leakage is very important for a turbine designer to determine the capacity of required drainage systems (pump, piping etc). Unfortunately, there are no experimental reports or well-known estimating methods available at the current time. However, customers are usually very interested in knowing the amount of leakage and it may sometimes be set as a basic requirement of seal specification.

For estimating leakage, the conservative approach will assume maximum gap size for the specific leak paths. For this seal design, there are three possible paths, which the water may pass through and cause leakage.

1. <u>The tongue and groove areas between seal segments</u>.
   This is the first source of leakage. The geometry of this path is somewhat complex, see Photo 1. The water path consists mainly of 3 sections, "A", "B", and "C" (Photo 1). The gap areas "A" and "C" provide the circumferential spacing to allow movement of the segments in operation. These are relatively large but will be reduced in size over the life of the seal.

   Section "B" provides the flow control gap between the convex and concave surfaces of the tongue and groove. This will be closely controlled in manufacturing of the segments. In this case, leakage estimation will be based only on section "B" being the controlling gap.

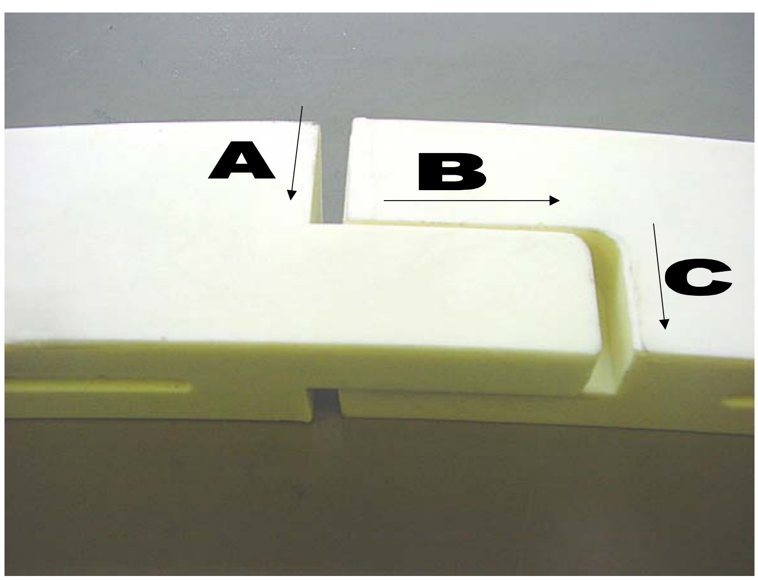

Photo 1

2. Gap between the shaft and seal segments
Since the hydraulic closing force and garter spring keeps the segments close to shaft, this gap will be essentially closed (or very small) keeping this leakage small.

3. Static seal interface with housing
In addition to the above two leakage paths, there can be leakage through the static seal contact area of the segments against the housing. Because of the differential pressures involved, the seal ring will be pushed against either the upper or lower boundary surface of the seal cavity. However a perfect seal will not be formed at this interface and some leakage will be evidenced.

Photo 2

Authors

Guojun Ren, PhD P. Eng. has been a Senior Application Engineer with Thordon Bearings Inc for 4 years. He has been directly involved with several seal projects for Hydro Turbines in South America, New Zealand, and China. Dr. Ren received his PhD from University of Stuttgart, Germany.

Keith Laskey, PhD P. Eng. has been Chief Design Engineer with Thordon Bearings Inc for 5 years and is intimately involved with new product development and testing. Dr. Laskey received his PhD from University of Waterloo, Ontario, Canada.