Design Aspects for Axial Segmented Main Shaft Seal For Hydro Turbines

by Keith Laskey, PhD P. Eng, Thordon Bearings Inc., Ontario, Canada

Abstract

Reliable shaft sealing is an important concern for large hydro turbine operators for shaft dia. 400 to 2000+ mm (16 to 80 in). Typical turbine axial seals are constructed of a non-rotating segmented carbon face ring operating against a rotating steel collar. Carbon seal rings comprise of very brittle segments which present significant handling and installation difficulties and may have a somewhat foreshortened life span if misaligned or subjected to abrasives.

Thordon Bearings Inc. has embarked on a program to optimize designs of such seals utilizing its proprietary SXL elastomeric material. SXL is formulated from thermosetting resins forming 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 axial seal segments comprise two annular lands formed via a central groove. The individual segments are bonded into place to form the ring and secured with bolting. The ends of the segments are typically sealed with adhesive. Each of the circumferential lands will perform a pressure breakdown function between a centrally fed injection pressure and the turbine pressure on the one side and atmospheric pressure on the other. The arrangement will depend on whether the seal is outside or inside pressurized. The higher pressure injection flow (10 to 15 % above turbine pressure) functions as a seal lubricant, coolant and a barrier to abrasives entering the seal faces.

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

The paper presents design aspects of typical axial seals for hydro turbine and large pump applications. Test results for radial seals from the development program to date covering operational pressures to 1.0 MPa (150 psi) are also given.

Some operational history for both radial and axial seals is included.

Author

Keith Laskey, PhD P. Eng. has been Chief Design Engineer with Thordon Bearings Inc for 7 years and is intimately involved with new product development and testing. Dr. Laskey received his PhD from University of Waterloo, Ontario, Canada.

Introduction

This paper is a continuation from the previous paper presented in 2005 at the Waterpower XIV conference in Austin, TX. At that time, our test results were rather sparse and even now the new experimental results are limited. However, Thordon has some test data and field experience data to report on for the radial seals and an exciting field success for a large axial seal application.

Lubrication

Some form of lubrication at the dynamic faces of the seals will be required to facilitate the removal of heat and to control wear of the faces. Three possible lubrication modes are described as follows:

1. hydrodynamic
2. hydrostatic
3. boundary

Hydrodynamic lubrication is a self-generated pressure film in the interface between relatively sliding surfaces.

Hydrostatic lubrication is normally developed through the use of an externally supplied high pressure source available to the interface. In this case the high pressure fluid to be sealed seeps into the interface with the tendency of separating the faces.

Boundary lubrication occurs when the film thickness from either hydrostatic or hydrodynamic lubrication does not develop a supporting film thick enough to allow clearance of surface asperities. In this case high friction and reduced leakage may result in premature damage of the faces from overheating.

For the pressure fed axial seal, there will be the obvious hydrostatic effect in combination with some hydrodynamic and likely areas of boundary lubrication near the atmospheric pressure edge.

The contact type radial seal depends on a combination of hydrostatic and hydrodynamic principles to lift the seal away from the shaft during operation. Test experience shows some burnishing of the dynamic surface indicating that this type seal is subject to some boundary lubrication conditions.

Radial seals

(a) Design

The development of Thordon materials for use as radial seals was well covered in the previous paper. Presently we have two basic design concepts for the radial seal:

Hydrodynamic or Contact style of seal

At the time of writing, this is the only seal we have tested. This design of seal is installed with auto adjustment gap between segments when the seal rings are assembled. Reference Fig 1, 2. As the pressure is applied, without shaft rotation, the segments are pushed against the shaft. As the shaft rotates, the amount of power required to overcome friction suggests that the hydrodynamic effects may not be quite as effective as desired.

Floating Ring Seal

This style of seal is installed essentially without any gap allowance between segments. The ID of the dynamic face will be the same as the shaft size. As wear and fluid absorption occur this seal ring will develop an optimum running clearance where the clearance is no longer increasing with run time and the leakage has stabilized at an acceptable value.

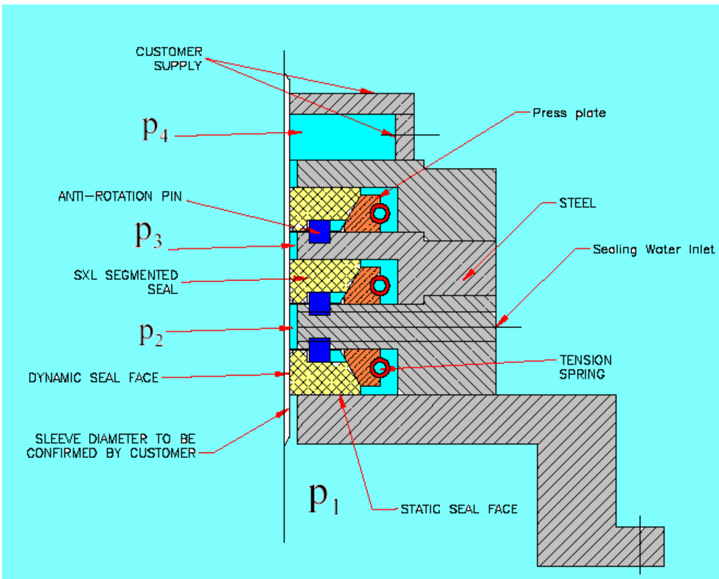

Fig 1: Typical Radial Seal Arrangement

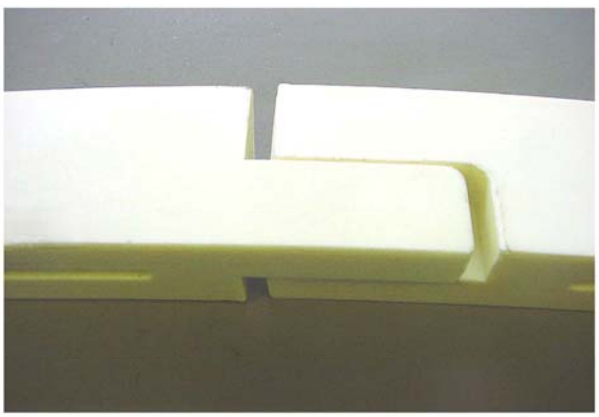

Fig 2: Radial Seal Segment Overlap

(b) Leakage

Seal leakage has been monitored for these radial seal from the beginning of the test program. There is a fairly wide scattering of values but the general trend appears to be close to a straight line and only moderately increased with pressure.
A plot of the varied leakage measures for the duration of our testing is given in Fig 3.

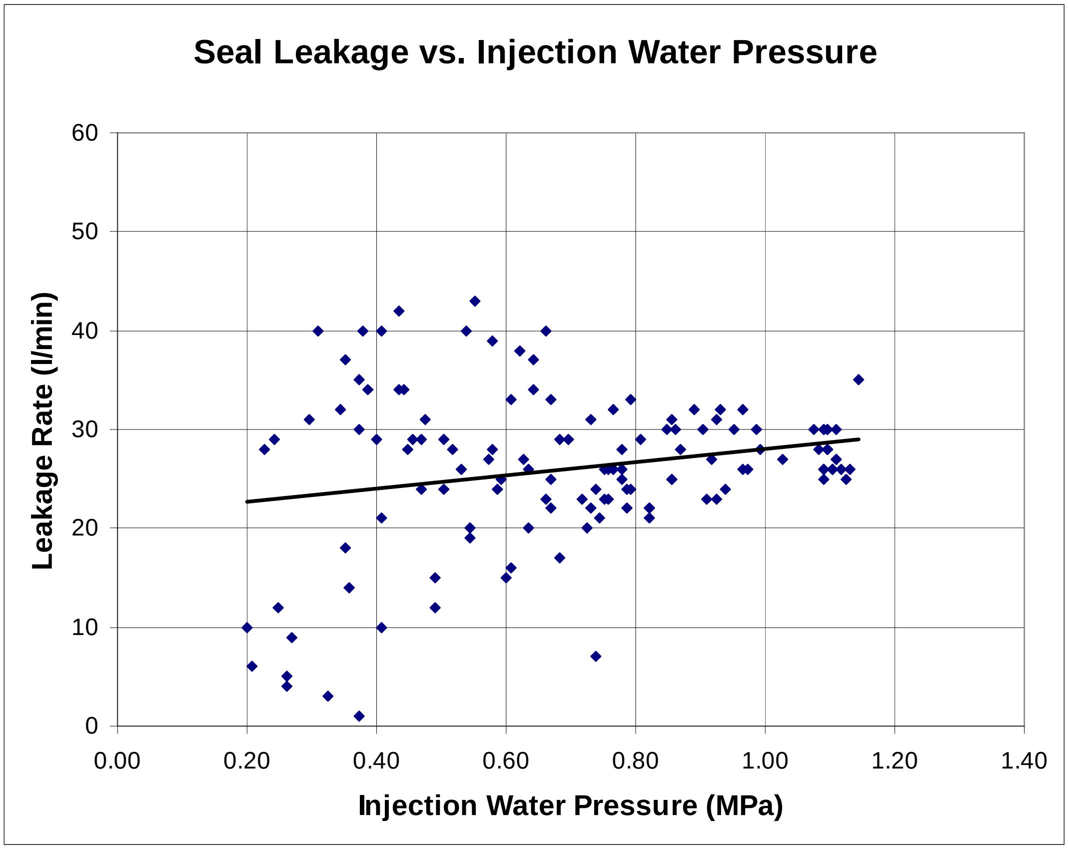

Fig 3: Radial Seal Leakage flow from test rig (1MPa = 145 psi)

(c) Test Observation

The dynamic seal face of one of the sets of seals has been modified to incorporate partial axial slots which appears to be somewhat more helpful in achieving hydrodynamic action than the plain surface, ref Fig 4. On two occasions when cooling was inadvertently lost and the temperature increased above 70^{o}C (158^{o}F), the plain lands were contacted more heavily resulting in some material removal. The seal faces with the axial slots did not sustain any damage. The amount of leakage under normal operation demonstrated no significant difference between these seal designs.

Fig 4: Axial slots in dynamic seal face

(d) Field Experience

(i) The contact design of seal has been installed in a recent high pressure (1 MPa) application in China and to date the customer is quite pleased with the performance. Special leakage handling systems which were originally installed have not been activated as yet, so normal leakage is well controlled. The design of seal is as generally described in the previous paper presented by the author.

(ii) Another application at Hydro Quebec, Canada also using this style of seal has been performing satisfactorily for many years. This seal operates under ~ 0.2 MPa (30 psi) pressure and the wear rate has been equal to or better than the carbon equivalent operating side by side. In this case the shape, features and face design was defined by the original carbon seals.

(iii) A large set of pumps (60,000 HP/unit) at Lake Havasu, AZ, were originally installed with carbon radial seals. These were later modified from the original carbon to Thordon SXL material. Seal design essentially follows the general carbon seal face detail format for a replacement component. These seals performed successfully for about 5 years prior to being replaced with an axial seal design (again using Thordon SXL) in an attempt to reduce the leakage even further.

(iv) An additional large pump application by the California Dept of Water Resources in Los Angeles water uses the hydrodynamic or contact style of seal. This application has operated successfully now for approximately 10 years at stuffing box pressures of ~ 0.17 MPa (25 psi). Normal leakage for this application is approximately 35 l/min. (10 gpm)

(v) One more significant success story with a floating ring design seal is at the Manapouri Station in New Zealand. This seal is made with ID matching the shaft diameter and installed with a small circumferential gap (~ 2 mm or 0.08 in) for bedding-in of the seal. The customer has been very happy with this seal now for more than twenty years. Over time, the seal wears and allows increased leakage. The situation is remedied by removing one segment from each seal ring and shortening the length slightly to return the leakage control to original condition.

Vertical Axial Seal Design Aspects

Typical axial seal segments comprise two annular lands formed via a central groove, see segment photo Fig 5. The individual segments are bonded into place to form the ring and secured with bolting. The ends of the segments are typically sealed with adhesive. Each of the circumferential lands will perform a pressure breakdown function between a centrally fed injection pressure and the turbine pressure on the one side and atmospheric pressure on the other. The specific arrangement will depend on whether the seal is outside or inside pressurized. The higher pressure injection flow (10 to 15 % above turbine pressure) functions as a seal lubricant, coolant and a barrier to abrasives entering the seal faces.

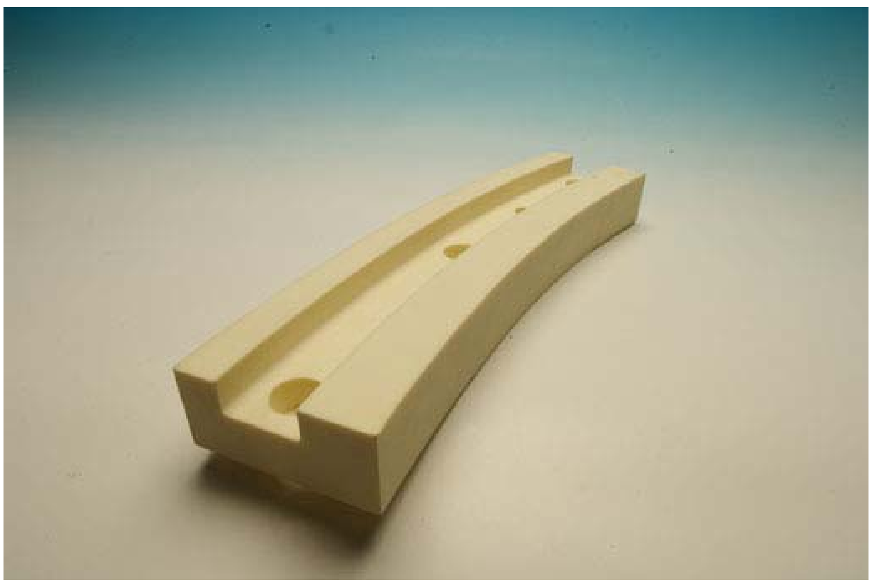

Fig 5: Typical Axial Segment

The basic construction of the vertical axial seal usually consists of a rotating collar attached to the shaft with a static secondary seal (rotating). On the upper face of the collar, a non-rotating floating cylindrical element with a facing of suitable material to form the seal is mated with the rotating collar, see Fig 6. The floating element is designed to move vertically while sealed against the casing structure (secondary seal, static). This element may or may not be provided with a spring adjustable loading element. For internal pressurized units, with the floating element mounted below the rotating element, springs would be a necessary component of the design.

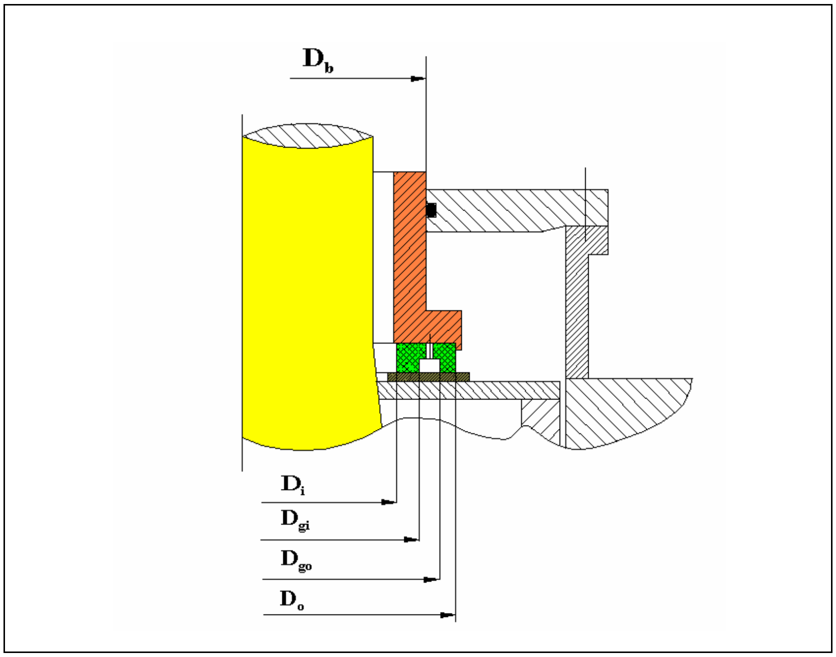

Fig 6: Typical Axial seal arrangement (outside pressurized)

Seal closing force

The total closing force will be the sum of the floating element weight, any applicable spring force and the hydraulic load force on the seal faces. The total closing force is

$$\mathbf{F_{tc} = F_{sp} + F_{el} + F_{hy}} \quad \ldots\ldots\ldots\ldots\ldots.. \text{(1)}$$

where: F_{sp} is the spring force
F_{el} is the floating element weight
F_{hy} is the hydraulic load

The closing pressure on the seal face is

$$\mathbf{P_{cl} = F_t / A_f} \quad \ldots\ldots\ldots\ldots\ldots.. \text{(2)}$$

$$\mathbf{A_f = \text{seal face area} = \frac{1}{4}\pi(D_o^2 - D_i^2)} \quad \ldots\ldots\ldots\ldots\ldots.. \text{(3)}$$

Spring force: F_{sp}

Because of the large diameters involved with these applications the spring load will be realized through a number of individual springs with some method of stiffness adjustment provided. Because of the difficulty of achieving equalized pressures using springs, the outside pressurized arrangement with stationary element above the rotating collar is the preferred orientation.

Hydraulic force: F_{hy}

The hydraulic closing force will depend on the balance ratio of the seal, B. This ratio compares the hydraulic load area to the seal face area, A_f. For B>1, the seal is considered unbalanced. As B falls below 1, the seal is termed balanced with varying degrees of balancing based on the actual value of the ratio fraction. The balance ratio for typical mechanical face seals falls between 0.65 and 0.85. For values less than 0.65, the clearance gap may be come unstable resulting in possible face damage, premature wearing and excessive leakage.

Hydraulic force

$$\mathbf{F_{hy} = \tfrac{1}{4}\,\pi\,(D_o^2 - D_b^2)\,P_u + \tfrac{1}{4}\,\pi\,(D_b^2 - D_i^2)\,P_d} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

where: P_u is the upstream pressure or pressure in the turbine
P_d is the down stream pressure outside of the seal
D_o is the seal outside diameter
D_i is the seal inside diameter
D_b is the seal balance diameter

For the general case of hydro turbine, P_d will be atmospheric pressure, P_a, and will have equal effect on the opening force and the closing force. Therefore in general the second term may be eliminated and the formula becomes

$$\mathbf{F_{hy} = \tfrac{1}{4}\,\pi\,(D_o^2 - D_b^2)\,P_u} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Seal balance ratio

$$\mathbf{B = \tfrac{1}{4}\,\pi\,(D_o^2 - D_b^2) / \tfrac{1}{4}\,\pi\,(D_o^2 - D_i^2)} \qquad \text{(outside pressurized)} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

or

$$\mathbf{B = \tfrac{1}{4}\,\pi\,(D_b^2 - D_i^2) / \tfrac{1}{4}\,\pi\,(D_o^2 - D_i^2)} \qquad \text{(for inside pressurized)} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

Reference Fig 6 for definition of the various diameters.

For the outside pressurized case, a rearrangement of terms results in the following for the hydraulic closing force

$$\mathbf{F_{hy} = A_f * B * P_u} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (8)$$

And the total closing force can now be written as

$$\mathbf{F_{tc} = F_{sp} + F_{el} + A_f * B * P_u} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

If springs are not used, $F_{sp} = 0$

Seal Opening force

The resisting support load pressure distribution at the seal face will be a function of the seal material stiffness. For Thordon SXL material, the face deflection is anticipated to deflect in similar fashion to the seal cross-section shown in Fig 7. This assumes a central feed pressure at ~10% higher than the turbine pressure. Alternately, if no central hydrostatic lift pressure was introduced and the seal was to operate with the pressure across the full face, the material deflection would be as indicated in Fig 8. The latter

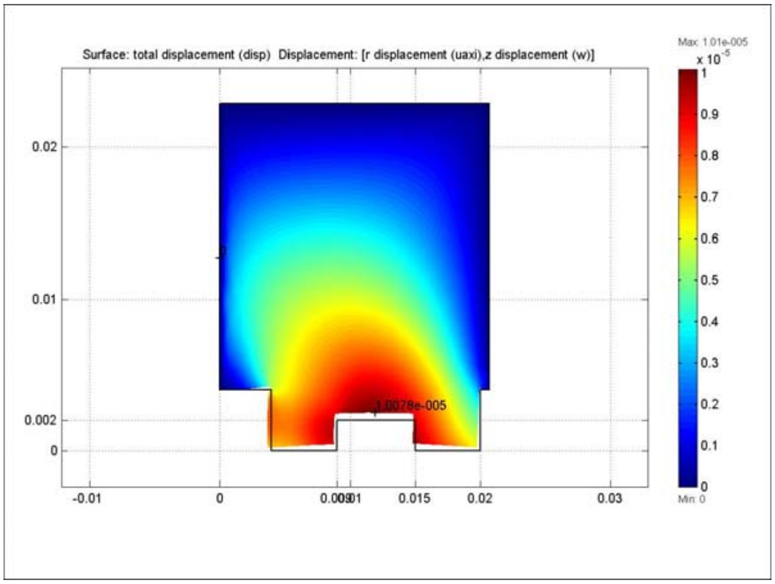

Fig 7: Typical Axial Seal with central feed pressure

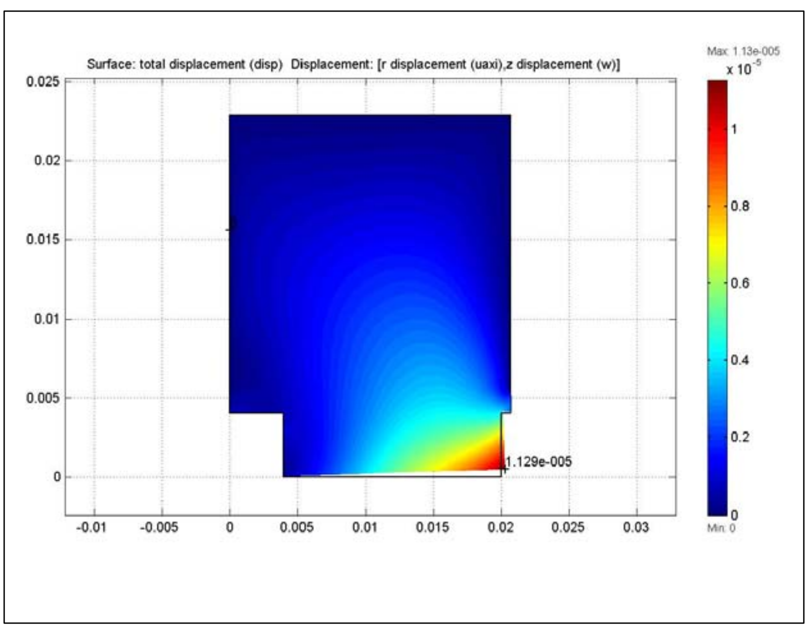

Fig 8: Axial seal with No central feed

approach may result in premature wear of the ID edge area, particularly if abrasives are introduced into the normal turbine flow. With the higher pressure central feed to the seal, abrasives may be controlled via treatment of the feedwater (filtration or separation). This approach also virtually ensures the development of convergent surfaces from the higher pressure region to the lower pressure regions. The annular groove position will be slightly offset toward the high pressure side which is the OD for outside pressurized arrangements. The ΔP to the OD side will be smaller (P_i-P_u: where P_i is the injection pressure) compared with the pressure drop to the ID side (P_i- P_a).

Opening force

The total opening force will be the summation of three components

$$\mathbf{F_{to} = F_{hs} + F_{hd} + F_{fc}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

where: F_{hs} is the hydrostatic opening force
F_{hd} is the hydrodynamic opening force
F_{fc} is the face contact force

The hydrostatic opening force is developed according to the pressure pattern indicated in Fig 9.

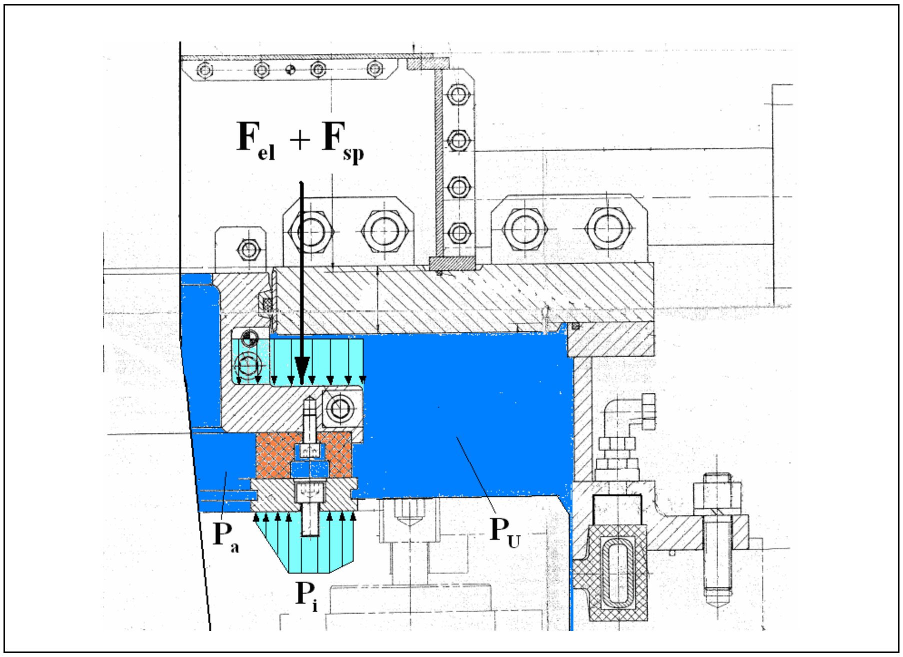

Fig 9: Force pattern on seal face

$$Fo = \tfrac{1}{4}\pi (D_{gi}^2-D_i^2) * \beta_1 * P_i + \tfrac{1}{4}\pi (D_{go}^2-D_{gi}^2) P_i + \tfrac{1}{4}\pi (D_o^2-D_{go}^2) * \beta_2 * P_u$$

$$= \tfrac{1}{4}\pi\{(D_{gi}^2-D_i^2) * \beta_1 * P_i + (D_{go}^2-D_{gi}^2) P_i + (D_o^2-D_{go}^2) * \beta_2 * P_u\} \quad \ldots\ldots (11)$$

where: D_{gi} = the diameter of the inside edge of the annular groove
D_{go}= the diameter of the outside edge of the annular groove
P_i = injection pressure
$\beta_1 =$ the pressure distribution factor applicable to the pressure drop from P_i to P_a
$\beta_2 =$ the pressure distribution factor applicable to the pressure drop from P_i to P_u

The pressure distribution factor β will depend on the shape of the mating faces. The value will normally vary from 0.5 to 1 for faces converging in the direction of pressure drop and from 0 to 0.5 for diverging faces. For parallel faces the value of β will be 0.5. The actual value for Thordon SXL is anticipated to be 0.7 to 0.8 because of the resulting surface shape due to material flexibility.

For the axial seal situation, the value of injection pressure will be selected so that the face contact force, F_{fc}, is effectively zero. If the injection pressure is too high, excessive leakage will result. The value should be set to minimize leakage which will depend on a small hydrodynamic film being developed between the surfaces which effectively minimizes seal damage and wear at the low pressure edge. The lubrication condition at this location may well transition between boundary (or contact) to full hydrodynamic regime during operation. The correct injection pressure will be determined based on several trial runs while monitoring leakage. Adequate leakage will be required to allow proper lubrication and cooling of the seal while controlling the leakage flow to a manageable quantity. This may vary from 20 to 100 l/min (5 to 26 gpm) depending on the application pressures involved and the diameter of seal.

Field Experience at Ataipu, Brazil

An axial seal was installed in a large (715 MW) turbine at Ataipu Hydro Power Plant, Brazil during late 2005. This seal has an upstream or turbine pressure of P_u = 1.16 MPa (168 psi). The injection pressure is approximately 10% greater than P_u. The seal was manufactured and installed in segments which were bonded and mechanically fastened to the floating element. The intermediate gaps between segments were filled with adhesive.

Normal leakage from this seal has been monitored consistently from start up and has typically varied in the range of 20 to 50 l/min (5 to 13 gpm) with one excursion to as high as 96 l/min. This spurious excursion may have been a result of slight build up of materials on the seal face, see Fig 10. The seal has recently been examined, cleaned and found to be in excellent shape with only very minor wear of ~ 0.1 mm (0.004 in) in one localized area. This seal continues to operate successfully in this application.

Fig 10: Itaipu axial seal after one year of operation

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