Refurbishment of Axial & Radial Type Turbine Shaft Seals Using Elastomeric Polymers to Extend Wear Life and Reduce Leakage Rates

Rénovation des joints d'étanchéité de type radiaux et axiaux des arbres de turbines au moyen de polymères élastomères pour augmenter la durée de vie et réduire les taux de fuites

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Mots clés: Rénovation, joints d'étanchéité, élastomères, lubrification à l'eau, fuites.

ABSTRACT

Reliable shaft sealing is an important concern for large hydro turbine operators. Large diameter turbine shaft seals (400 – 2000mm shaft diameters) have traditionally used either axial or radial type seals, constructed with a non-rotating housing containing sealing face elements most often made of carbon-graphite material, and a rotating metallic steel sleeve or collar. Shaft seals in this size range function primarily to control the leakage to an acceptable and stable level, since some leakage is essential for cooling and lubricating the seal faces.

For over five years, Thordon Bearings embarked on a comprehensive test and development program to optimize the design of shaft seals, utilizing an elastomeric polymer material as the sealing face instead of carbon-graphite. This paper will discuss the basic working principles of axial and radial type shaft seals, with an emphasis on design considerations when utilizing modern polymer materials to reduce leakage and extend seal face wear life in existing seal assemblies. Two simple case studies will be shared where real world success was achieved by refurbishing the shaft seals following the design approach developed through Thordon's R&D program.

With an emphasis on "green" technologies, the paper will also present a brief introduction to the use of water lubricated turbine guide bearings as an effective solution to eliminate potential sources of oil contamination in close proximity to the turbine shaft seals. Using water lubrication for the turbine guide bearings rather than traditional oil lubricated bearings can also simplify shaft seal accessibility and reduce maintenance time by allowing the seal to be relocated to a position above the turbine guide bearing.

La fiabilité des joints d'étanchéité d'arbres est une préoccupation importante pour les opérateurs des grandes turbines hydrauliques. Les joints d'étanchéité d'arbres de grands diamètres (400 – 2000 mm) ont traditionnellement utilisé des solutions de joints axiaux ou radiaux, constitués d'un logement non-tournant contenant des éléments de joints de contact habituellement faits de carbone-graphite et d'une chemise métallique ou d'un collier tournant. La fonction des joints d'arbres dans cette gamme de tailles est principalement de maîtriser la fuite à un niveau acceptable et stable, car un peu de fuite est essentiel pour refroidir et lubrifier les faces de contact des joints.

Thordon Bearings s'est lancé dans un programme complet de 5 ans de tests et développements afin d'optimiser la conception de tels joints, en utilisant un matériau polymère élastomère comme joint de contact à la place du carbone-graphite. Cet article traitera des principes basiques du travail des joints d'arbres radiaux et axiaux, en soulignant les considérations de conception lors de l'utilisation de matériaux polymères modernes pour réduire les fuites et augmenter la durée de vie du joint de contact dans les assemblages de joints existants. Deux études de cas de turbines hydrauliques seront

partagées, pour lesquelles le succès « dans le monde réel » a été obtenu, en rénovant les joints d'arbres selon l'approche de conception développée à partir du programme R&D de Thordon.

En mettant l'accent sur les technologies « vertes », l'article présentera également une brève introduction sur l'utilisation des paliers guides de turbines lubrifiés à l'eau comme une solution efficace pour éliminer les sources potentielles de contamination par huile dans l'environnement immédiat des joints d'arbres de turbines. Utiliser un palier guide de turbine en polymère élastomère lubrifié à l'eau plutôt qu'un palier traditionnel lubrifié à l'huile peut également simplifier l'accessibilité au joint d'arbre et réduire la durée de maintenance en permettant au joint d'être relocalisé en position au-dessus du palier guide de turbine.

1. Background on Elastomeric Radial & Axial Shaft Seals

Perhaps the simplest and probably the oldest sealing solution is the labyrinth. The seal rings are often lined with white metal and will have small gaps between the rotating shaft and the stationary seal rings. True sealing is not achieved in this non-contact design, but for small to medium sized machines operating at low pressures, acceptable leakage rates may be achieved.

Another fairly simple sealing technology is the "stuffing" or "packing" box. In these designs, a soft packing material expands within the housing when compressed, and thus closes the clearance between the shaft and housing. Initial leakage rates are usually low after installation of new packing, however there are two main shortcomings for this solution. First, abrasive silt in the water can become imbedded into the packing material and damage the mating shaft surface. Second, frequent manual adjustments are required to maintain the correct level of compression of the packing to ensure low leakage rates. This adjustment can be quite difficult due to accessibility, especially when the seal is located underneath an oil lubricated turbine guide bearing.

A commonly used solution in service on many turbines around the world is the carbon ring radial seal. This is a "segmented" shaft seal with carbon rings comprised of individual arc segments. There is a variety of designs for this type of seal and the general concept is to achieve some balance of water pressure between the rings to ensure low leakage rates. One of the main leakage paths in this type of seal is at the joints between carbon segments. The leakage for a segmented radial seal can vary from low to high depending on shaft size, number of rings per seal set, number of segments per ring, sealed water pressure and vibration. The disadvantages of segmented carbon seals are that they are susceptible to abrasive wear and the material can be brittle and often damaged during installation.

For shafts greater than ~1000 mm, or when the sealed water pressure exceeds ~1.0MPa, an axial type of seal arrangement is often used instead of the radial type. These are sometimes referred to as mechanical seals or face seals. The traditional face material for this type of seal is carbon with stainless steel or bronze as the mating surface. Carbon sizes are limited however and for large shaft diameters the carbon ring needs to be segmented.

A direct improvement on the conventional radial and axial shaft seals can be achieved by using an elastomeric polymer as the segment or face material instead of carbon. The Thordon SXL elastomer can be easily machined to directly replace carbon segments in existing seal assemblies, and exhibits high resistance to abrasive wear and toughness against cracking or chipping during installation. Thordon Bearings Inc. has successfully designed radial segmented seals that have been installed on shafts ranging from 300 mm to 2000 mm. Axial seals for sealing diameters up to 4000mm have been supplied using this elastomer face material in.

2. Working Principles & Design Considerations

2.1 Radial Sealing

Typical radial segmented seals comprise two or three stages of interlocking segmented sealing rings. Each ring has both dynamic (against the shaft) and static (against the housing) sealing faces. Most radial seal assemblies will have a supply of higher-pressure filtered water introduced between the sealing rings. This higher pressure flow functions as a seal lubricant & coolant, as well as a barrier preventing abrasive water 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 also to maintain the integrity of the seal ring within the housing cavity. Two of the most common radial seal configurations are shown in Figure 1 and Figure 2 below.

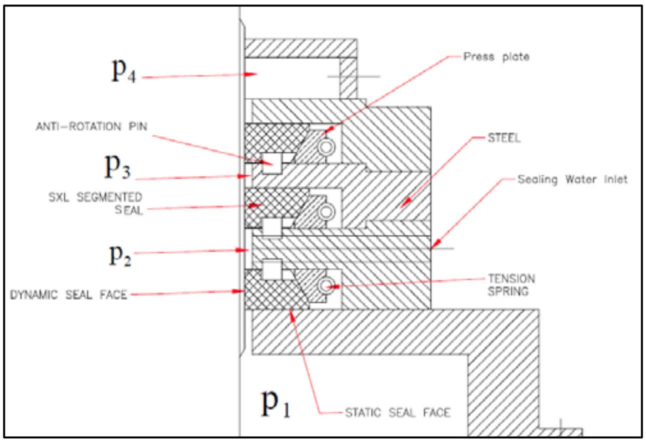

Fig 1 - Typical 3-Ring Radial Seal

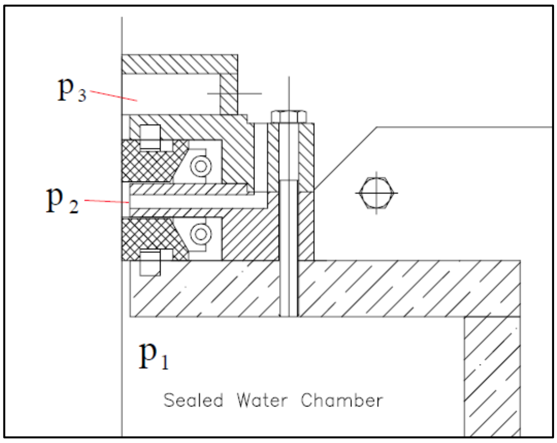

Fig 2 - Typical 2-Ring Radial Seal

When considering the design of a radial type of shaft seal, some information is required from the site in order to perform a comprehensive design review. The data required is summarized in Table 1.

"P1" – Sealed water pressure (bar)	Shaft diameter (mm)
"P2" – Injection water pressure (bar)	Shaft rotating speed (rpm)
Existing flow rate of injection water (L/min)	Filtration level of injection water (micron)
Quality of sealed water (relatively clean, or high abrasive/silt loading)	Maximum acceptable leakage rate (L/min) – usually limited by drain or pump capacity

Tab 1 - Data Required From Site – Radial Shaft Seals

Once the site data is gathered, several more design parameters are determined by analysis and calculation. The critical parameters for a radial seal design are summarized in Table 2.

Seal segment / shaft interface pressure	Tension spring force
"P2" – Optimal injection water pressure	Tension spring dimensions
Required flow rate of injection water	Optimal Filtration level of injection water
Seal segment dimensions & quantity	Approximate leakage rate expected (L/min)

Tab 2 - Design Data Determined by Calculation – Radial Shaft Seals

A full calculation is not presented here, as this is typically done for each specific installation by the supplier of the seal face material. Each sealing face material will have different performance characteristics that may affect the life and durability of the seal face material.

2.2 Axial Sealing

The operating principles of a hydro-turbine axial seal may appear similar to those of smaller traditional mechanical face seals typically fitted to pumps and other small industrial applications, but the shaft size disparity between the two applications causes the practical operation of the seals to differ considerably.

Due to the much larger size of hydro-turbine axial seals, a precise sealing surface flatness cannot be economically achieved. Rather than microscopic "light band" flatness as found on small mechanical seals, the sealing surface flatness on a large axial turbine seal exhibits as macroscopic unevenness. While this unevenness is beneficial for hydrodynamic film formation and consequent low operating friction, it is also the cause of much of the leakage past the seal.

Observations show that even when there is significant water flow across the sealing face there is still a significant amount of heat created by friction within the seal. Some leakage must occur to remove this heat and prevent seal from overheating. While it may be desirable to have "zero" leakage" from a large turbine, practical economics require that the seal should just be capable of maintaining leakage at a consistent and acceptable level for the life of the seal face. A typical axial seal assembly design is shown in Figure 3 below, with Figure 4 identifying the area ratio that must be considered carefully when calculating the force balance on an axial seal face.

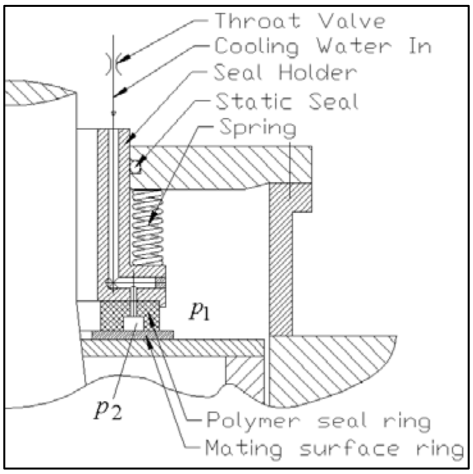

Fig 3 - Typical Axial Seal Assembly

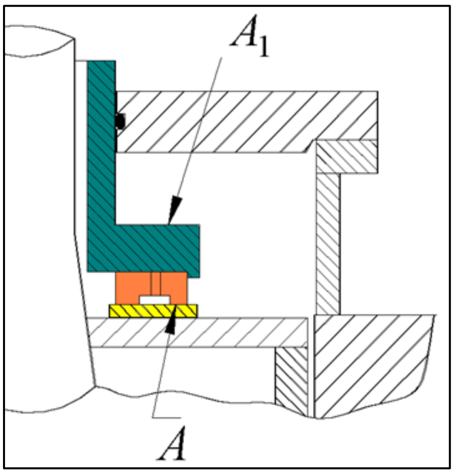

Fig 4 - Axial Seal Face Balance Ratio (A1/A)

As with the radial seals, information from the site is required in order to analyse the design of an axial seal assembly. The data required for an axial seal is summarized in Table 3.

"P1" – Sealed water pressure (bar)	Shaft diameter (mm)
"P2" – Injection water pressure (bar)	Shaft rotating speed (rpm)
Existing compression spring details (quantity, size, stiffness, etc.)	Existing seal face dimensions, mass and dimensions of seal holder
Existing flow rate of injection water (L/min)	Filtration level of injection water (micron)
Quality of sealed water (relatively clean, or high abrasive/silt loading)	Maximum acceptable leakage rate (L/min) – usually limited by drain or pump capacity

Tab 3 - Data Required From Site – Axial Shaft Seals

The calculated design parameters for an axial seal design are summarized in Table 4.

Seal face / mating surface interface pressure	Compression spring force
"P2" – Optimal injection water pressure	Compression spring dimensions & stiffness
Recommended seal face balancing ratio (A1/A)	Approximate leakage rate expected (L/min)
Required flow rate of injection water	Optimal filtration level of injection water
Sealing face geometry and material	Suggested mating surface finish & hardness

Tab 4 - Design Data Determined by Calculation – Axial Shaft Seals

The typical segment design of a turbine axial seal is quite simple. The cross section of the segments is rectangular in shape with a circumferential water groove offset from the mid-diameter. Holes for the fastening bolts and injection water inlets bring water to a groove at the sealing interface. To limit the amount of abrasives introduced to the seal, typically filtered injection water is suppliedvia the water groove at a pressure higher than the turbine water. With consideration of continuously varying friction between machine parts, fluctuating temperatures and pressures of both injected and sealed water, the precise calculations of force balancing for an axial seal quickly become quite complicated. Figure 5 below illustrates the major forces acting on the free-body of the seal (seal holder and elastic polymer seal ring together).

- F_S is the total spring force tending to close the seal
- F_f is the net friction force from all interactive parts
- W is the total gravitational weight acting on the movable seal holder component
- F_C is the solid interactive contact force between the seal mating surfaces
- p_1 is the turbine pressure to be sealed (note that p_1 is usually not equal to the turbine head, since the head pressure is usually already reduced by a labyrinth type seal at the runner)
- p_2 is hydraulic pressure in the seal water groove and as a rule of thumb, $p_2 \approx (1.1-1.15) \cdot p_1$.
- A_1 is the area of the seal holder exposed to sealed water pressure contributing to closing force
- A_2, A_3, and A_4 represent the different surface areas on the sealing ring contributing to opening force

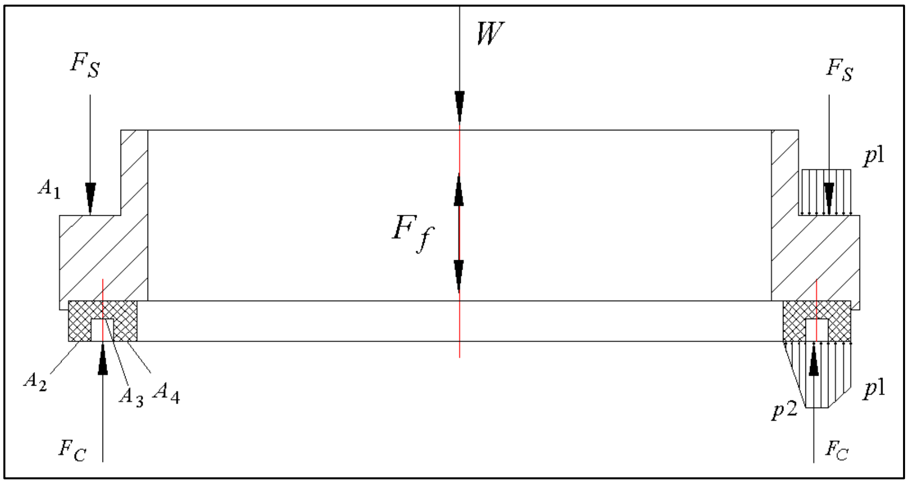

Fig 5 - Forces Acting on Axial Shaft Seal

In order for the axial type shaft seal to function correctly, an optimal force balance must be achieved by selecting spring forces, seal face geometry and injection water pressure carefully in order to maintain sufficient leakage through the seal for cooling purposes through the whole turbine operating range, but also holding leakage at an acceptable level. This analysis is typically done on a case-by-case basis for each axial seal design.

3. Case Study - Carbon Radial Shaft Seal Retrofitted with Thordon SXL Elastomer

In 2016 Thordon Bearings began discussion with a customer operating a power station in New Zealand to discuss possible refurbishment of a radial shaft seal that was fitted with carbon segments. A total of six identical vertical Francis turbines were installed at the station, with each unit rated at 93MW at 92m of head, rotating at 166 rpm (sliding speed of 7.4 m/s at the seal face). The units were originally built and installed by General Electric in 1965, and have a shaft diameter of 850mm over the stainless steel sleeve fitted in way of the shaft seal.

Fig 6 - Original Carbon Seal Segment

Fig 7 - Assembled Shaft Seal

The original seal assembly was quite complex, comprised of four rings of carbon segments, with eight segments per sealing ring as shown in Figure 8 below. Leakage was fairly high at about 40 – 50 L/min, and the carbon segments were replaced about every five years due to wear. The existing seal housing was built with large metal castings, but overall was still in good condition.

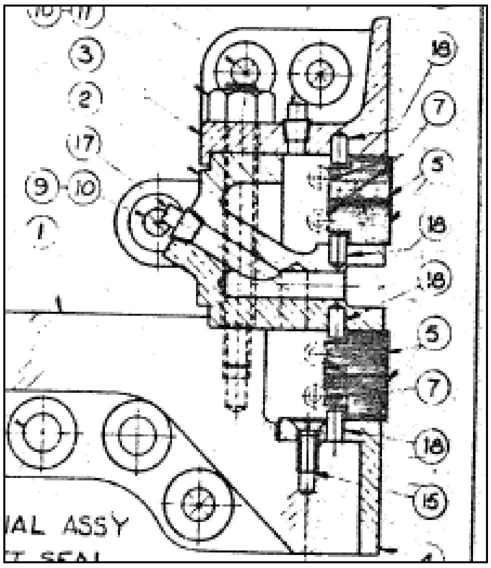

Fig 8 - Original Seal Assembly Design (4x carbon seal rings, 32 segments)

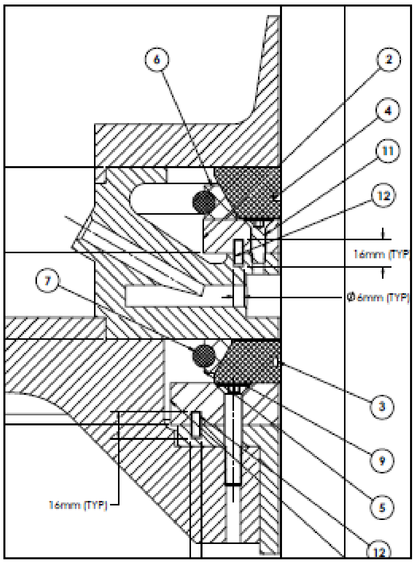

Fig 9 - Thordon Refurbished Design (2x SXL seal rings, 12 segments)

A design concept as shown in Figure 9 was submitted to the customer. This concept proposed rebuilding the internal components of the seal assembly with two new rings of elastomer seal segments, and two new bronze spacer plates (Figure 10) in place of two of the original carbon seal rings. This solution would allow the large cast metal housing to be retained to minimize cost, but provide the required geometry and housing support for the new elastomer seal rings. The number of individual seal segments per ring was also reduced from eight pieces to six pieces, and together with a unique interlocking end-joint detail, the Thordon SXL segments would help to reduce the potential leakage pathways and reduce the total leakage from the assembly.

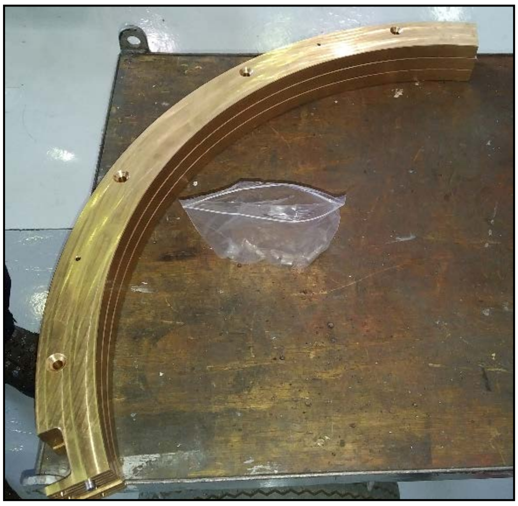

Fig 10 - New Bronze Spacer Plates

Fig 11 - New Garter Springs & Spring Holder (yellow)

A stainless steel shaft sleeve (Duplex 2205 grade) was already fitted to the shaft in way of the seal rings, so no further modification to the turbine shaft was required.

After six months, the first seal was disassembled and inspected with no observable wear on the SXL segments, or shaft sleeve. Leakage was reduced to <10 L/min. Upon completion of the successful trial at this hydro station on the first unit, the remaining five turbines will be converted from carbon segments to the SXL elastomer in the same manner.

4. Case Study - Carbon Axial Shaft Seal Retrofitted with Thordon SXL Elastomer

Towards the end of 2003 Thordon Bearings were contacted by the operators of one of the largest hydro-electric power plants in the world, in connection with retrofitting the seal on one of their operating turbines. The power station has a total of 18 large vertical Francis turbines, originally built from 1984 - 1993 by Voith & Alstom. Each turbine is rated at 715MW, rotating at 93 rpm with a head of 110m.

The existing seal was an axial face type using a segmented carbon ring with an average diameter of 4000mm. The seal leakage was unacceptably high (140 L/min) and the carbon segments were suffering from severe abrasive wear from sand in the water. The segmented carbon seal was very brittle and there was concern about the extreme care required when performing maintenance work.

In Figure 12, the original seal arrangement is shown. Two carbon rings, each comprised of many small segments, were installed into two separate grooves of the steel carrier ring. Figure 13 shows the simplified arrangement as suggested by Thordon. The two carbon rings and their steel carrier were replaced by a single ring of profiled SXL elastomer seals, comprised of only six segments.

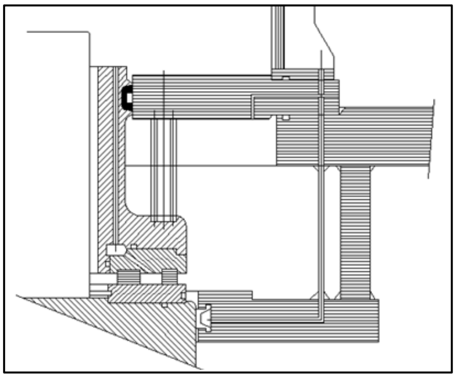

Fig 12 - Original Carbon Seal Arrangement

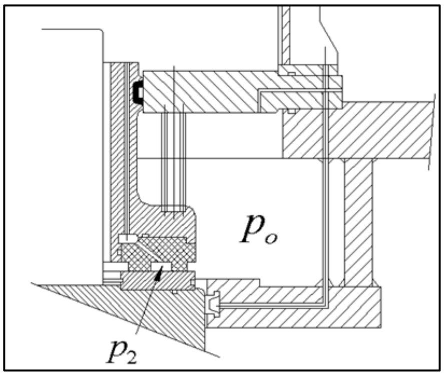

Fig 13 - Thordon Refurbished Design

A major consideration of the proposed change was that the original steel ring was 400kg while the SXL sealing ring was less than 100kg, a difference of over 300kg to the balancing conditions of the seal. Thordon engineers carefully examined the force balance conditions. The technical review indicated that changes to the original springs would be unnecessary, as calculations indicated the existing spring force would be sufficient to keep the sealing face closed.

The first seal was installed by the end of 2005 and the plant engineers had created a detailed plan to test the seal's performance during re-start of the turbine, and in service. Accordingly, the seal was tested under different conditions. As the turbine was re-started, the leakage rate and temperature increase of leakage water were carefully recorded for stand still condition, runner operating at 15 rpm and at full speed (90 rpm); all results were satisfactory. The cooling water pressure, water supply flow, leakage rate and temperature increase were all carefully monitored for the next year. During the first two months the readings were taken daily and for the months thereafter, the reading interval was extended to twice a week. Initial feedback from the plant engineers were that conditions all seemed satisfactory and the leakage was less than that of the previous carbon seal.

After some time, the operators again contacted Thordon with concerns that the leakage rate had further reduced. While delighted that the leakage was small, the operators were concerned that leakage was perhaps too low and would compromise the longevity of the seal as they thought perhaps it would overheat. With a review of the temperature rise within the injection water and its outlet temperature, Thordon advised that the seal was not overheating and operation of the machine should continue. Leakage rates continued to fluctuate somewhat during initial commissioning, but eventually stabilized at approximately 60 L/min - considerably less than that of the previous carbon seal.

The total injection cooling water flow rate was held at 240 L/min, with a supply pressure at 3.0 bar. The external leakage rate of 60 L/min represents about 25% of this injection water, which means that the remaining 75% (180 L/min) of injection water is flowing back into the turbine and providing an effective barrier to prevent abrasive laden river water from entering into the sealing interface.

After one year of operation, the turbine was taken apart for inspection. After opening the seal, the sealing face on the turbine water side was clean and since the machining marks were still visible there had been negligible wear if any. The sealing face on the leakage side however had several regions that appeared discolored (dark) and there were some film like material partially adhered / detached from the seal face. It was immediately suspected (incorrectly) that these were indications of overheating and sealing problems. Most of the discoloration and the adhering films however were removed with simple cleaning and the sealing surface then appeared very smooth. The brown coloured film substance taken from the sealing surface was sent to Thordon for composition study. Thordon tested the content of substance and it was determined 60% of it was organic from the process water and 40% was SXL material removed from sealing surface. Precise dimensional measurements of the seal were recorded and only one small section had worn by 0.1mm; wear in other areas was negligible. Accordingly, it was predicted that the seal should work for the 20 year requirement.

Fig 14 - New Thordon SXL Seal Segment

Fig 15 - Sealing Face - 1 year Inspection

In light of the successful installation, the operators similarly changed the seal of a second turbine in 2007, and have since replaced the carbon seals in six of the eighteen turbines at the power station. It is the intention of the operator to continue upgrading the carbon seal faces for the remaining twelve turbines, as each unit is taken out of service for refurbishment according to the pre-defined schedule.

The original seal installed on the first unit in 2005 is still working perfectly and should not require replacement until the next maintenance overhaul in 2025. This achievement represents a substantial increase in service life and increased generation potential by reducing the downtime associated with replacing the previous carbon seal faces every five years.

5. Case Study – Improved Access to Shaft Seal With Water Lubricated Guide Bearing

In a turbine operating with an oil lubricated guide bearing, the shaft seal must necessarily be located below the bearing assembly (Figure 16). In the case of a water lubricated turbine guide bearing, the shaft seal may now be fitted on the top end of the guide bearing (Figure 17). This case study presents an example where a turbine with an oil lubricated bearing had a very difficult to service shaft seal, and was upgraded to use a water lubricated turbine guide bearing, primarily to improve access to the shaft seal assembly.

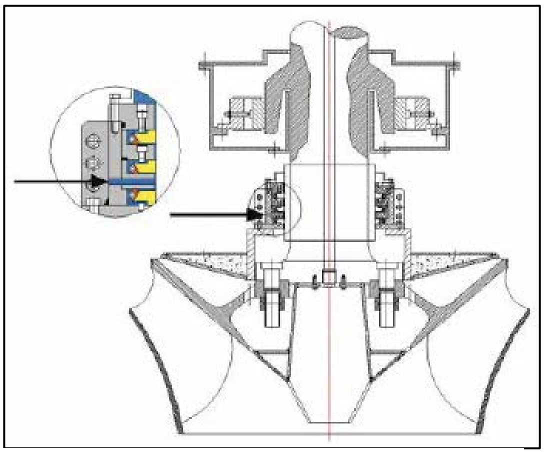

Fig 16 - Oil Lubricated Guide Bearing (Seal Assembly is Below Bearing)

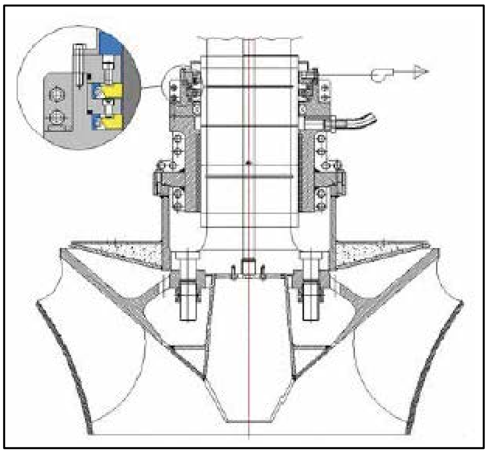

Fig 17 - Water Lubricated Guide Bearing (Seal Assembly is Above Bearing)

The Centrale di Fabbriche power station is owned by a large utility in Italy, and was originally built in 1955 with one 15MW vertical Kaplan turbine supplied by RIVA, operating at 430 rpm and 30m head. The original turbine guide bearing was an oil lubricated Babbitt design, with a conventional radial shaft seal using multiple rings of carbon graphite segments, located below the oil bearing.

Discussion with the customer began in 2014, and was mainly focused on finding a way to improve access to the shaft seal. In the original design, any problems encountered with the shaft seal were very difficult and time consuming to resolve since it would require a complete disassembly of the oil lubricated turbine guide bearing. It was decided that a water lubricated bearing could be the solution to this problem, so several design concepts were developed for review and discussion. The most suitable concept required complete elimination of the existing oil bearing and shaft seal, being replaced by a new water lubricated bearing and shaft seal mounted above the bearing assembly.

Figure 18 shows the configuration of the original turbine guide bearing and shaft seal, with the major components identified.

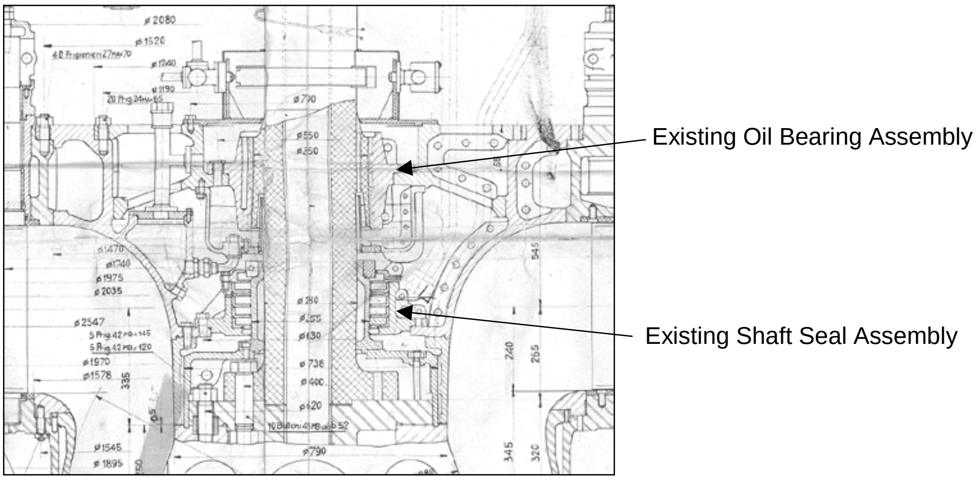

Fig 18 - Original Oil Bearing & Shaft Seal

The implemented solution to improve access to the shaft seal is outlined in Figure 19.

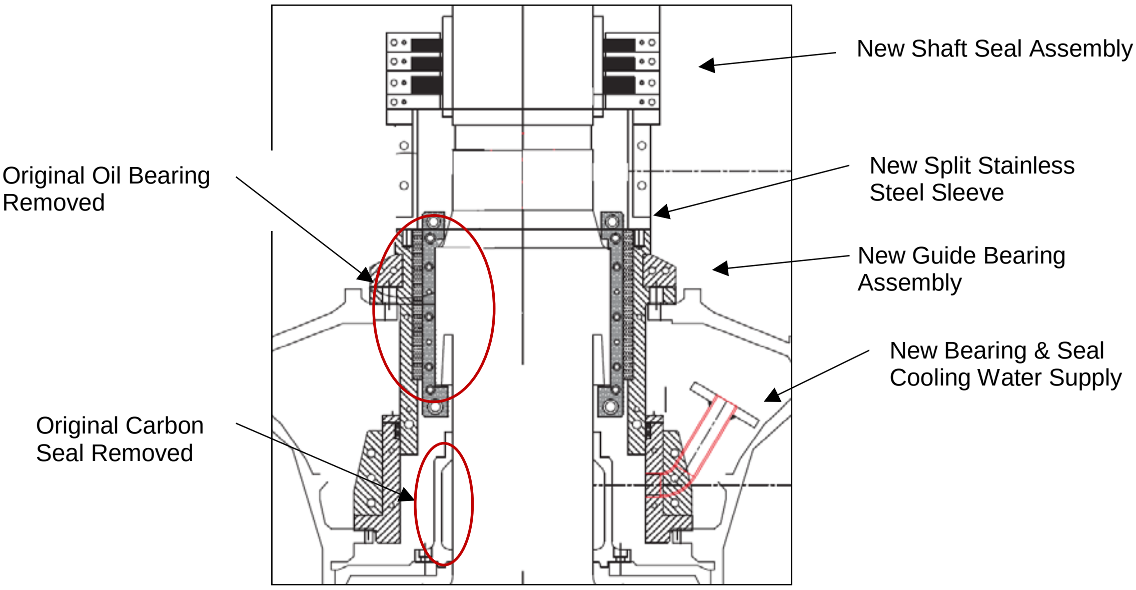

Fig 19 - New Water Lubricated Bearing & Shaft Seal

The elastomer bearing material was secured into the new steel bearing housing using a tapered keyset design to create an interference fit between the bearing and the steel housing. This configuration allows for easy servicing or inspection of the water lubricated bearing inserts, reducing downtime associated with any bearing maintenance that may be encountered in the future.

The filtered water system configuration has water entering the assembly below the bearing and moving upwards through the bearing to the shaft seal.

Fig 20 - Thordon SXL Bearing Insert

Some minor leakage was expected through the runner gap and also through the shaft seal, but this configuration created a semi-closed loop system that reduced the total consumption of clean water while still maintaining the minimum required flow rate through the bearing space to cool and lubricate the bearing.

The shaft was protected using a split and bolted stainless steel shaft sleeve that could be removed in the future if needed, rather than the permanent metallization approach that was utilized in the first case study. The disadvantage of using a bolted sleeve was that it increased the diameter over the shaft and therefore the bearing and housing had to be correspondingly made larger in diameter.

After completing the fabrication of all components in late 2015, the new water lubricated bearing and shaft seal were installed successfully in March 2016.

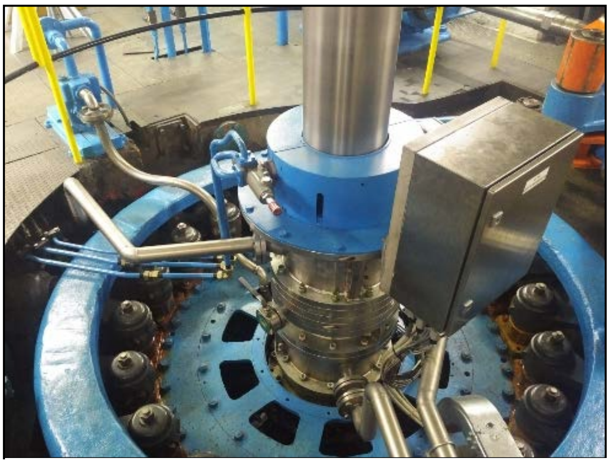

Fig 21 - Installed Bearing & Shaft Seal

Fig 22 - Bearing & Seal Water Filtration System

Initial trials went well, with a planned stop for inspection several weeks after the initial start-up. The unit was stopped to inspect the bearing and shaft seal. There was some observation that the initial water leakage from the shaft seal was quite high, but after investigation it was discovered that there was a problem with the orientation of the seal segments and after correcting this, the leakage stabilized at an acceptable rate of <10 L/min. The shaft seal and water lubricated bearing are still working well in service now for just over two years.

6. Conclusions

Arriving at the optimum shaft seal design is not a simple task, and the designer must consider all components of the overall arrangement, force balancing, spring selection, and use creative design ideas to fit new materials into existing assemblies. The case studies presented have demonstrated that using an elastomeric polymer seal during refurbishment of large axial and radial type shaft seals offers great advantages to the turbine operator. A well-engineered seal refurbishment solution will provide:

1. Robust construction, easier handling & installation of seal face elements
2. Fewer segments, smaller leakage pathways, reduced overall seal leakage
3. Reliable operation and long wear life, with exceptional resistance to abrasive wear from dirty river water conditions.
4. Cost savings, by re-using large metal components where possible

When faced with difficult access to a problematic shaft seal, converting the turbine to use a water lubricated guide bearing can also be considered as a solution. Water lubricated bearings have been used successfully in practice around the world since the earliest installations of hydro turbines, utilizing wooden blocks or staves to support the turbine shafts.

Modern water lubricated bearing technology, using elastomeric bearing materials, is suitable for Francis or Kaplan turbines in vertical or horizontal configurations. The oil-water bearing conversion example outlined in this paper demonstrates the integration of a modern solution into an older turbine to solve a challenging seal problem.

With careful planning and review of the condition of existing equipment, the end-user can integrate the conversion work into a planned outage to minimize disruption to power production, with most fabrication and component manufacturing work completed ahead of time.

Thordon Bearings has long been at the forefront of providing premium bearing and seal systems for water lubricated applications. With the seal upgrades in New Zealand and Brazil, as well as the oil to water turbine guide bearing conversion done in Italy, Thordon has continued setting the example globally and demonstrating that well-engineered water lubricated designs can improve the performance of existing shaft seals, as well as solve ongoing maintenance and repair problems associated with older equipment.

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