Reducing Maintenance with Water Lubricated Turbine Guide Bearings - Design Principles & Case Studies

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Abstract

Water lubricated turbine guide bearings can provide an effective solution for reducing maintenance costs and downtime associated with keeping older turbines running reliably. Significant operating and economic benefits can be realized by replacing a poorly functioning or damaged oil lubricated turbine guide bearing with a water lubricated design, as part of a well-planned refurbishment or overhaul project. Future maintenance will be simplified since the water lubricated bearing allows the shaft seal assembly and the turbine guide bearing to be arranged in an uncomplicated and compact way. The seal assembly can be easily accessed, the oil sump and lubrication system will be replaced with a simpler water supply system, and the bearing itself can be quickly inspected or replaced, in case of damage, by renewing only the non-metallic bearing insert in a matter of hours. Moreover, there will be no danger of leaking oil into the river, eliminating pollution risk at its source.

This paper will briefly review general design guidelines, principles of operation and answers to commonly asked questions related to using water lubricated non-metallic bearings in vertical Francis or Kaplan turbines. An extended discussion will then continue, showing two successful examples of rehabilitation projects where water lubricated bearings were used to achieve the operational and maintenance benefits described previously. Before and after diagrams of each implementation will be presented, with specific attention to the critical design parameters considered for each project.

The first case study presented is a Francis turbine owned by a large Spanish utility, originally built in 1958 and upgraded with a water lubricated turbine guide bearing in 2015, currently fully operational. The second case study comes from a large utility in Italy where a similar upgrade to a water lubricated turbine guide bearing was carried out in 2016 on a Kaplan turbine originally built in 1956, also now fully operational. There are some notable differences in the design approach between the two cases, and the reasons why certain features were selected for each project will be explained.

The paper will conclude with a step by step approach for maintenance and operational personnel, so they can identify problematic oil lubricated turbine guide bearings for potential conversion to a complete water lubricated bearing solution. Following the suggested approach, the paper will provide the end-user with factual project scoping and budgeting support when planning future turbine overhauls and major refurbishments.

1 History of the Water Lubricated Turbine Guide Bearing

The history of water lubricated bearings must have started with the idea of using water as a lubricant. This came about with the introduction of the screw propeller as a means of ship propulsion in the 1840's [2]. The development of the water lubricated bearings relates to the story of bearing materials evolution, as stated by Mr. John M. Foulk [3]. Brass was the first bearing material used for supporting propeller shafts, followed by white-metal bearings. The high profile failure of the white metal bearing during two transatlantic trips of the vessel *Great East* marked the emergence of *lignum vitae* wood as a suitable material for water lubricated bearings [2]. Since those early days, this species of wood in stave format continues to be used as a bearing material, but has been replaced in most water lubricated applications by modern synthetic materials.

Since the advent of using water as a lubricant for rotating underwater machinery, modern bearing material development has seemingly been following two main directions. One approach has been the use of soft elastomers that are resistant to abrasive wear, while the other approach has been to use hard ceramics. Both solutions are intended to handle some moderate level of abrasives and increase wear life when compared to earlier materials. A variety of other materials fall in between with different hardness, mechanical and tribological properties, such as lignum vitae, phenolic laminates, plastics [5] and other composite materials. The accidental discovery of rubber as water lubricated bearing material by a California mining engineer, Charles Frederic Sherwood, is a very fascinating and legendary story [2]. It is said that this has triggered the use of rubber and other elastomeric materials as bearings.

Indeed, using a soft elastomer as a water lubricated bearing material provides a number of advantages. Elastomeric bearings are inherently better able to handle the abrasives that are often present in the river water that is used to lubricate the turbine guide bearing. The grit particles are elastically pressed onto the bearing surface, rolling along with the shaft with minimum damage to the bearing itself, until they are flushed away by the water flowing through the bearing grooves. This behaviour has been confirmed both through experiments and in practice. The deflectable surface of a soft elastomeric bearing is able to form the lubricating water film at lower shaft speeds in comparison to the rigid surface of metal or ceramic bearings because the deflecting surface spreads pressure over a large surface and the peak pressure is reduced. This provides an early formation of lubrication film, but also enhances the ability of the bearing to accommodate shaft misalignment. The use of elastomers as water lubricated bearing materials for propeller shafts, pumps and hydro turbines became popular in the USA and Canada since the initial introduction of rubber bearings in December of 1922. In ~1935, rubber was introduced as water lubricated bearing material in Russia where it then found success in applications for hydro turbines [1].

2 Implementing a Water Lubricated Turbine Guide Bearing – Design Considerations

2.1 Load Capacity, Stiffness and Damping

From the practice point of view, the load capacity of a bearing is given by the load (pressure) and shaft velocity combination under which the bearing relies on a fully developed hydrodynamic lubrication film to support the load. The load capacity and friction characteristics – coefficient of friction change with shaft velocity (Stribeck Curves) are based on experiments [2-4]. Thordon Bearings Inc. has performed many load and friction tests on different bearing configurations. Since elastomeric materials typically possess a much higher E-modulus, the surface deflection under typical load conditions is at the same order of magnitude as the minimum lubrication film thickness. This allows the bearing load capacity to be calculated based on a rigid bearing approach for design optimization purposes. Design engineers are able to enhance bearing load capacity under given load conditions by changing the design parameters, such as running clearance, number of grooves as well as the size of the bearings [7]. Experience confirms that a reasonable guideline for maximum bearing design pressure shall be 0.27 MPa (40psi) for water lubricated vertical turbine and pump bearings [5]. Bearing pressures may be higher with careful consideration of loading and speed of operation to ensure a stable hydrodynamic water film through the normal operating range of the turbine.

While load capacity is typically well understood, bearing stiffness sometimes causes confusion among design engineers. Even many experienced hydro engineers argue that the stiffness of water lubricated bearings is low due to the low hardness of the materials. However, there is a big difference between the hardness of the materials and the stiffness of the bearings. As defined, the hardness of a material is its property against local deformation. In the elastomer world, hardness is correlated to the elastic modulus of the material. It is true that the stiffness of a bearing is a direct function of the elastic modulus of the material; however, the high Poisson ratio makes the stiffness of the bearing higher than that of the material. As example, the Poisson ratios for the Thordon elastomer family range from 0.450 to 0.499 depending on the bearing grade used. The high Poisson ratio makes an elastomer almost incompressible. The elastomer behaves similar to the oil in hydraulic cylinders when subject to compression, especially when the shape factor is high.

The stiffness of the hydrodynamic water film can be calculated separately based on theoretical analysis. Thordon Bearings Inc. has developed a procedure specifically designed for water lubricated turbine guide bearings with axial grooves [7]. This procedure is suitable for all bearings made of hard elastomers. The combined stiffness of water film and bearing material is usually higher than that of the supporting structure of the bearing housing. Since water lubricated bearings are running under higher eccentricity ratios in comparison to their oil lubricated counterparts, the stiffness of the water film will usually be higher than that of the oil film in oil lubricated bearings. Damping of the bearing doesn't change the critical speed of the rotating system, but will strongly influence the level of vibration. A higher level of damping is always beneficial. Again, the damping coefficient of the water film and the bearing material can be determined separately. While the damping of material can be obtained from experimentation, the damping of the water film can be calculated with the same procedure as stiffness.

2.2 Running Clearance

Running clearance is one of the most critical design parameters for any water lubricated bearing. When it comes to bearing design for vertical hydro turbine guide bearings, a small running clearance is beneficial. A smaller running clearance will increase the eccentricity ratio during operation for any given pressure and shaft speed which will essentially increase the water film thickness. The installed clearance, as the name indicates, is the total clearance of a bearing after installation. Since all non-metallic bearing materials will absorb a certain amount of water, this must be accounted for and a certain amount of allowance on clearance must be considered during the bearing design. Another influence on clearance is the operating temperature. The bearing design must ensure that the bearing never reaches a zero clearance condition through the entire operational temperature range of the turbine. For this reason, an allowance on clearance for maximum operation temperature must also be considered.

2.3 Operating with Abrasive Water Supply

In earlier years, abrasives were perhaps the biggest concern for designs using water lubricated bearings. With improved bearing materials and water filtration/conditioning technology, the presence of abrasives in river water is no longer a major issue for water lubricated turbine guide bearings. There are two typical solutions to handle abrasives in the lubricating water. The first solution is to separate the solids from the water. This may be done using conventional mesh or basket type filters, or by using centrifugal separators. The second solution to minimize abrasive wear is to use softer elastomers as the bearing material. Applying these two solutions together (water filtration & optimal material selection) allows the water lubricated bearing to achieve a similar life to oil lubricated bearings.

2.4 Material Selection

As stated in previous section, it is critical for design engineers to select a bearing material that is able to withstand the presence of some level of abrasives as well as a material that is favourable for developing a stable hydrodynamic water film. For example, Thordon produces two grades of elastomeric materials that can be applied depending on the quality of the water that can be delivered to the bearing. Thordon SXL (off-white colour) is normally used when water filtration can remove particles larger than ~80 micron; if this is not possible, the softer and more abrasive resistant GM2401 (black colour) grade can be used for installation in smaller units where there is no dedicated water filtration available.

2.5 Bearing Load and Length to Diameter (L/D) Ratio

With the development of computational fluid dynamics (CFD) software, bearing loads can be determined with reasonable accuracy as a result of hydraulic loads exerted on the runner. These loads may be used to determine optimal bearing sizing as well as calculation of stiffness, damping, and L/D ratio for the bearing. Several factors will affect the optimal bearing L/D ratio. The main factors are the bearing pressure, shaft alignment conditions, bearing performance, and space available. Sometimes, a longer bearing is required in order to achieve a safe bearing pressure with a fixed shaft diameter and load. Since turbine guide bearings are designed with a tight running clearance, shaft alignment becomes critical as the bearing length increases. Since the viscosity of water is low, a longer bearing is preferable in terms of water film formation; however, there may not be enough space for the longer bearing. Past experience suggests that an L/D ratio of 0.5 to 1.5 is suitable for a water lubricated turbine guide bearing. If there are no special requirements, an L/D = 1:1 is recommended.

2.6 Water Flow Requirements and Heat Removal

In practice, a water lubricated turbine guide bearing is generally implemented with an open-loop system. This allows the bearing to be located as close to the runner as possible. For a vertical turbine, this also means there is no need to fit a seal between the guide bearing and the runner. The main shaft seal is located on top of the guide bearing. Lubrication water provides three different purposes: it functions as a lubricant, as a cooling agent, and also flushes away the abrasives. The lubrication water is delivered into the space below the main shaft seal and above the turbine guide bearing so that it can be directed downward to lubricate the bearing surface, removing the heat and finally flows through the space above the runner exiting into the draft tube through balance holes of the runner (Francis), or at the hub gap just above the runner (Kaplan).

Ideally, a bearing operating under a fully hydrodynamic lubrication condition should achieve a very low friction and have no wear since there is no direct contact between the shaft and the bearing surface. However, the water film thickness is typically 20 – 50 microns. Installation misalignment, manufacturing inaccuracy, and frequent starts/stops or other transient conditions may interrupt the water film thickness with direct sliding contact between surfaces which can create a slight temperature increase in the bearing. Therefore, a sufficient flow of cooling water is always essential to the success of the water lubricated hydro turbine guide bearings. Although it is always recommended to perform a detailed heat load analysis to calculate minimum water flow rates, as a general guideline a minimum flow rate of 0.3 L/min per mm of shaft diameter will ensure sufficient cooling and lubrication for most installations.

2.7 Vacuum Below Bearings & Restrictor Rings

If implemented with an open-loop system as described above, there is no shaft seal between the turbine guide bearing and the runner. Practical experience confirms that there are situations where the water pressure below the guide bearing may have a large variation during turbine start/stop or transient operation. It is possible for the pressure below the guide bearing to become lower than the atmospheric pressure. If a vacuum is created below the bearing, it will result in a large pressure drop of the lubrication water across the bearing length which can cause the water to be drawn preferentially through the grooves and leave the working bearing surface un-lubricated. To solve this problem, turbine guide bearings can be designed with blind grooves. This means the bearings are facilitated with an annular ring at the bottom end of the bearing instead of letting the longitudinal grooves go through the entire length. Alternately a separate annular restrictor ring may be fitted below the bearing. The clearance between the annular ring and the shaft, as well as the length of the annular ring can be designed precisely based on the level of vacuum expected and water flow rate.

2.8 Bearing Construction and Fitting

Several factors determine the bearing construction. The most important are: ease of installation, retention and maintenance. The shaft supporting the runner typically will have a flange on both ends for connecting the runner and generator shafts, which means that all bearings must be fully split. A rigid steel bearing housing is required to hold the elastomeric bearing shell in place and support the bearing loads, and also to mount the whole assembly to the head cover. The housings are normally recommended to be made in stainless steel but may also be produced using painted structural carbon steel grades.

There are several options to hold the elastomeric bearing shell into the housing. The preferred method uses a tapered keyset design to create the required interference fit to hold the bearing in place. With the tapered keyset design, the bearing can be quickly inspected or replaced, in case of damage, by renewing only the non-metallic bearing insert in a matter of hours. Another elegant design is to use double keys incorporating two metal bars welded to the side edge of each half of the housing. When the two halves of the housing are put together, it forms two separate spaces for elastomer bearing shells. The third solution is to directly bond and/or mechanically fasten the bearing shell into the housing.

2.9 Main Shaft Seal

When the turbine guide bearing has an open-loop water circulation system, the shaft seal can be put on the top end of the guide bearing. The selection of seal type primarily depends on shaft diameter and sealed water pressure. For small to medium size shaft diameters from 100 mm to 300 mm, a good quality packing or mechanical seal can be used effectively. For larger shaft diameters, from 300 mm up to the largest sizes practical, radial or axial type shaft seals incorporating a long wearing elastomer seal face are a reliable and affordable solution.

3 Case Study #1 – C.H. de Prada – 36MW Francis Turbine

3.1 Project Background

The C.H. de Prada power station is owned by a large Spanish utility, and was originally built in 1958 with two 36MW vertical Francis turbines supplied by Neyrpic, operating at 600 rpm and 308m head. The original turbine guide bearing was an oil lubricated Babbitt design, with a conventional radial shaft seal using carbon graphite segments located below the oil lubricated bearing.

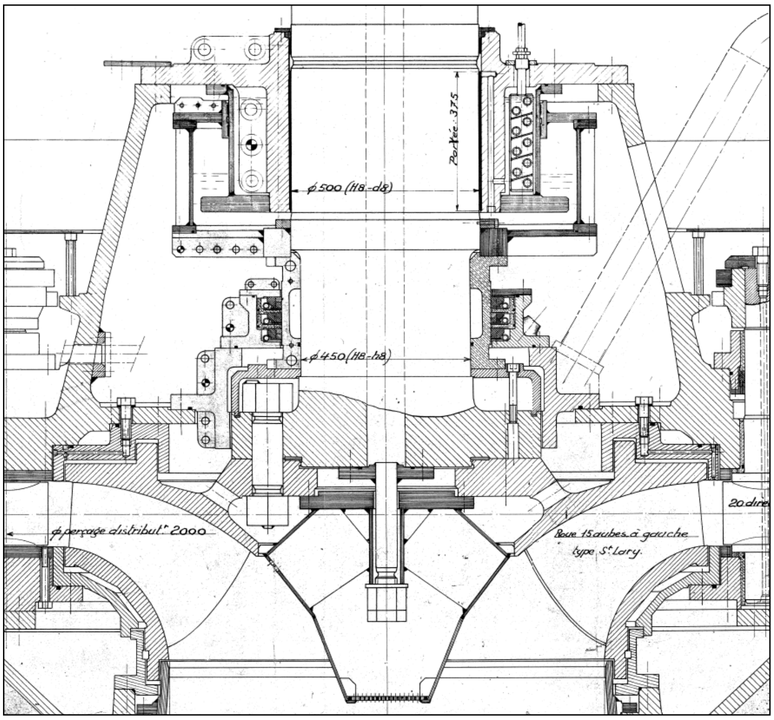

Figure 1 - Original Bearing & Shaft Seal – C.H. de Prada

Discussion was initiated with the customer in June 2015 about the possibility to convert the oil lubricated bearing on Unit 1 to an environmentally friendly water lubricated design. The conversion to water lubrication had the added benefit of improving access to the shaft seal, which would no longer be located below the oil bath and bearing assembly.

Thordon worked closely with a local workshop in Spain who were able to perform all of the metal fabrication and installation services for this project. Based on initial technical discussions and review of the turbine design, two basic concept sketches were prepared for consideration. The first concept maintained the existing carbon shaft seal, since it was uncertain how high the sealed water pressure below the seal could be during all operating modes. The second concept eliminated the carbon seal assembly entirely and placed the shaft seal at the top of the bearing assembly where it could be more easily accessed.

In order to select the most suitable design option, further investigation at the site was carried out to determine the real sealed water pressure during normal turbine operation. From these measurements it was determined that the existing runner labyrinth was effective at reducing the sealed water pressure so that the pressure below the existing carbon seal could be expected to range from a slight vacuum during unit stop (-0.1 bar) up to a maximum of 1.5 bar at full power.

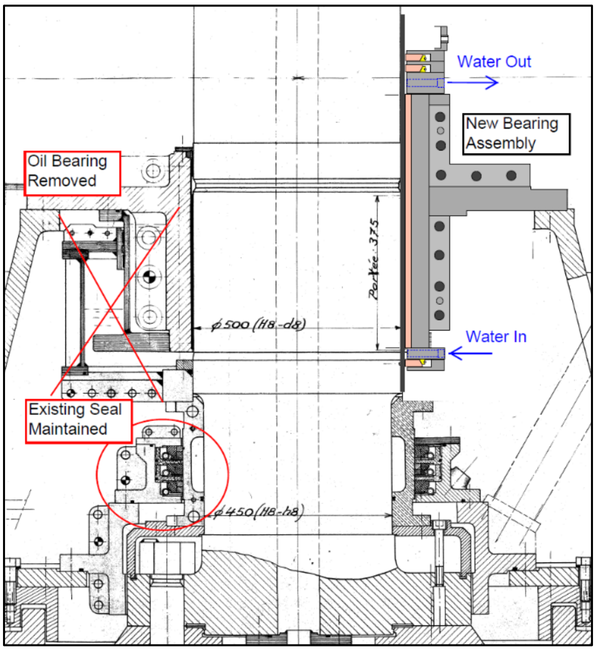

Figure 3 - Concept 1 – Maintain Original Seal

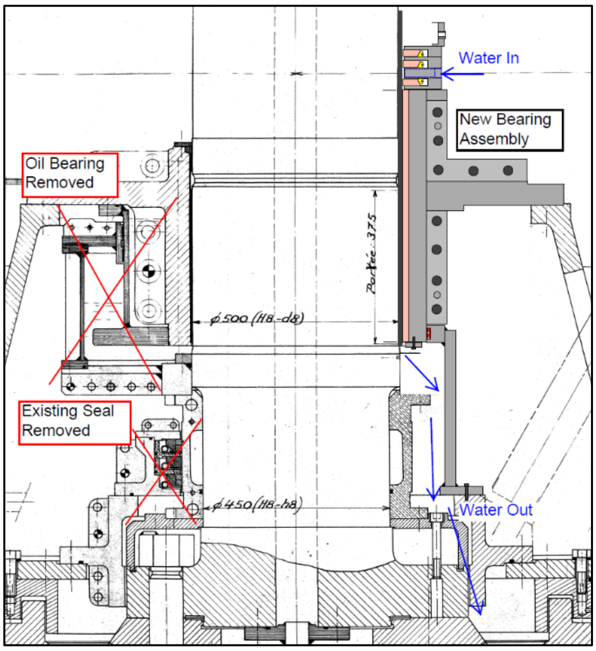

Figure 2 - Concept 2 – Remove Original Seal

3.2 Implemented Solution – Design Details

After reviewing the data that was measured to understand the real water pressure below the seal, it was decided to proceed with Concept 2. Working together with the local workshop, the final assembly design was developed and a full detailed bearing and shaft seal drawing set was prepared.

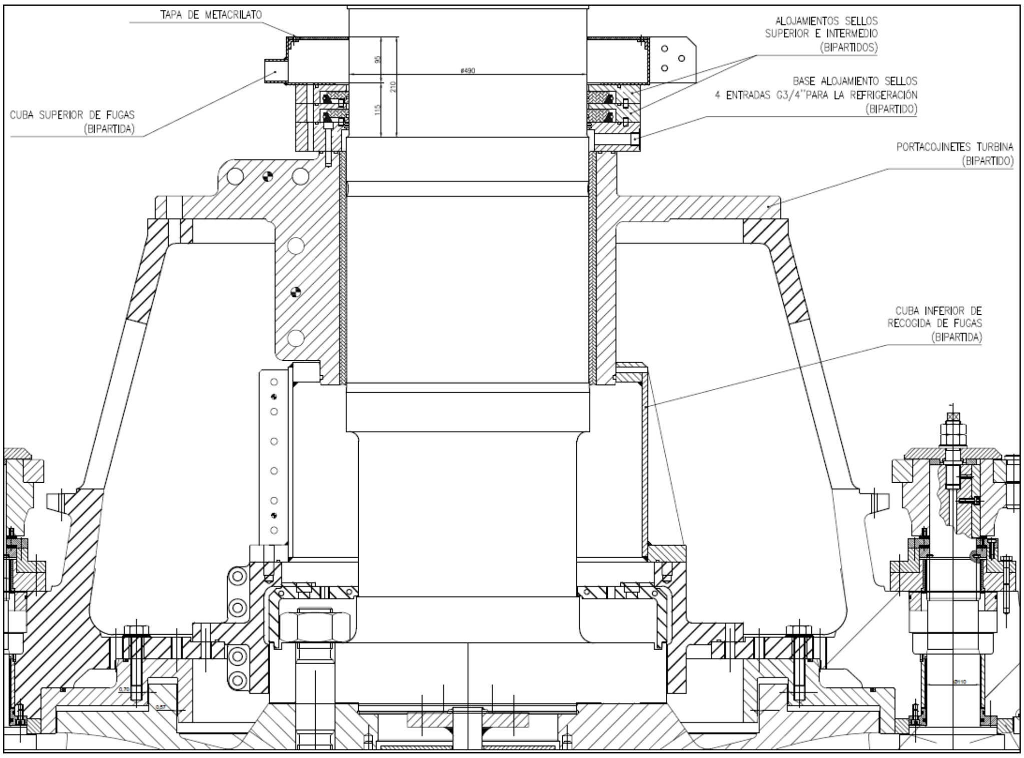

Figure 4 - Implemented Design (Drawing provided courtesy of local workshop - Milsa Trillo S.A.)

The Thordon SXL elastomer bearing was installed inside of a newly fabricated steel housing made of AISI304 stainless steel. A bonded fit design was used with careful preparation of the housing and use of an epoxy adhesive.

In order to protect the existing turbine shaft against corrosion after eliminating the oil from the system, shaft metallization was done in several areas along the shaft. A high hardness HVOF coating was applied in the active seal area on the shaft to eliminate risk of wearing of the shaft from seal contact, with a slightly softer 420 grade stainless steel metallization applied in the bearing area. The shaft was removed from the unit and taken to a workshop for this work to be carried out.

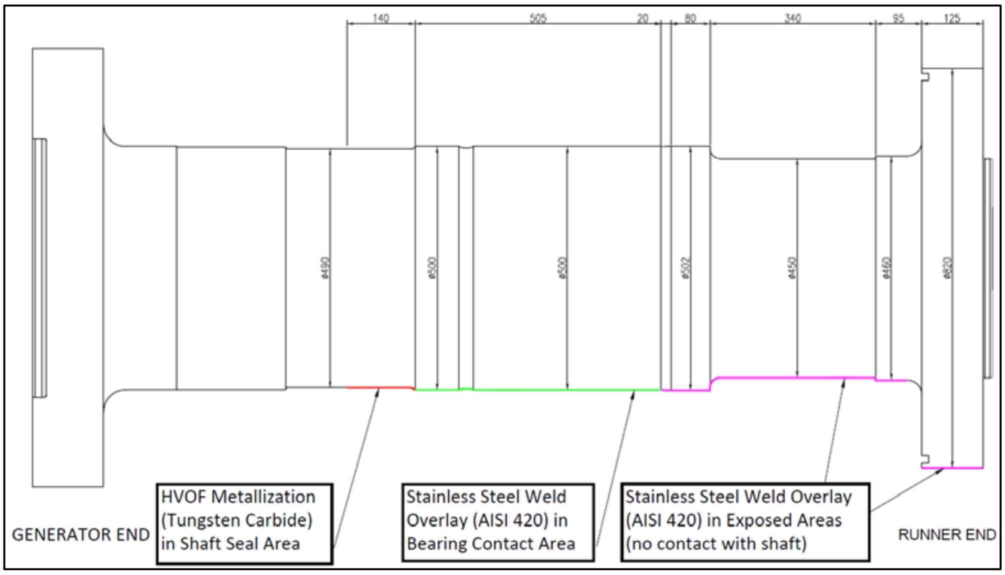

Figure 5 - Shaft Corrosion Protection (Drawing provided courtesy of local workshop - Milsa Trillo S.A.)

Water for bearing cooling & lubrication was supplied above the bearing and below the shaft seal at a flow rate of 150 – 180 L/min, with a minimum pressure of 2 bar in order to ensure positive water flow direction through the bearing at all times. In this installation, water was taken from the high pressure supply available from the penstock, which was then filtered to remove large abrasive particles that could shorten the bearing life. The water system utilized a self-cleaning cyclone type pre-filter, followed by a double inline filter of 60 micron. A pressure regulating valve was incorporated to reduce and control water supply pressure, and a flow control valve was included to allow the correct flow rate to be achieved. An ultrasonic flow meter, as well as pressure gauges and switches were provided to connect with the turbine control and alarm system. The control system was configured to confirm correct water flow & pressure as a pre-condition to allow unit start-up.

Figure 6 - Water Filtration System

3.3 Final Result & Current Status

After completing the fabrication of all components, the bearing and shaft seal was installed successfully at the site in November of 2015.

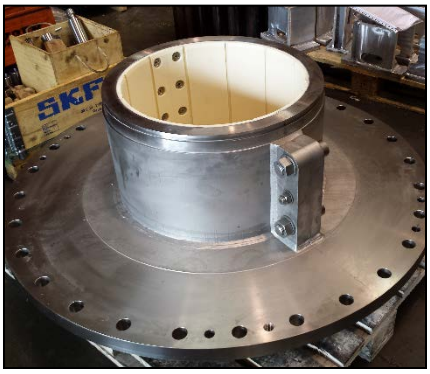

Figure 7 - Water Lubricated Bearing

Figure 8 - Bearing Installation

Figure 9 - Completed Assembly

In December 2015, the unit was re-started with the new water lubricated turbine bearing. It continues to run well today. In May 2016, Thordon worked again with the same customer and the local workshop to perform a similar conversion on the second unit at the same power station, so that now both turbines are working entirely with an environmentally friendly water lubricated bearing solution with easy maintenance accessibility and zero risk of bearing related oil leakage.

4 Case Study #2 – Centrale di Fabbriche – 15MW Kaplan Turbine

4.1 Project Background

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 was similar to the previous case study, with 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.

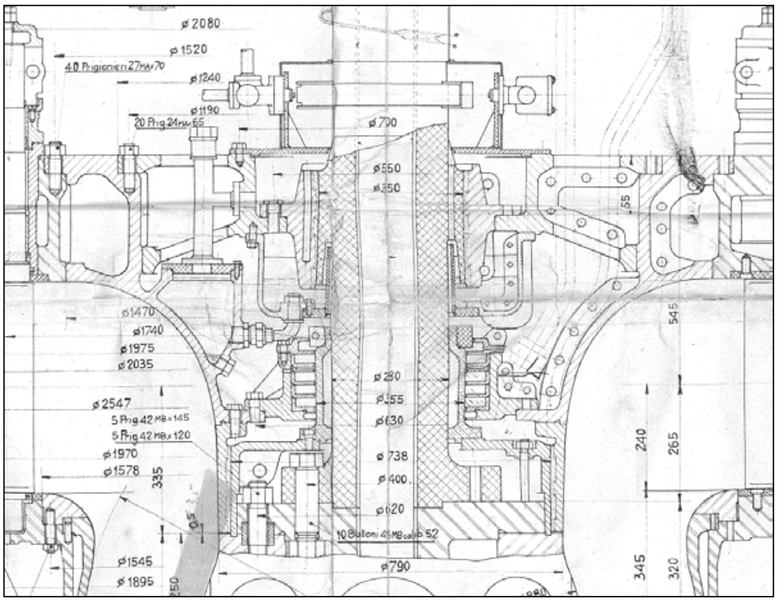

Figure 10 - Original Bearing & Shaft Seal - Fabbriche

4.2 Implemented Solution – Design Details

Although the general concept was similar to the previous case study, there were a few notable differences in this example.

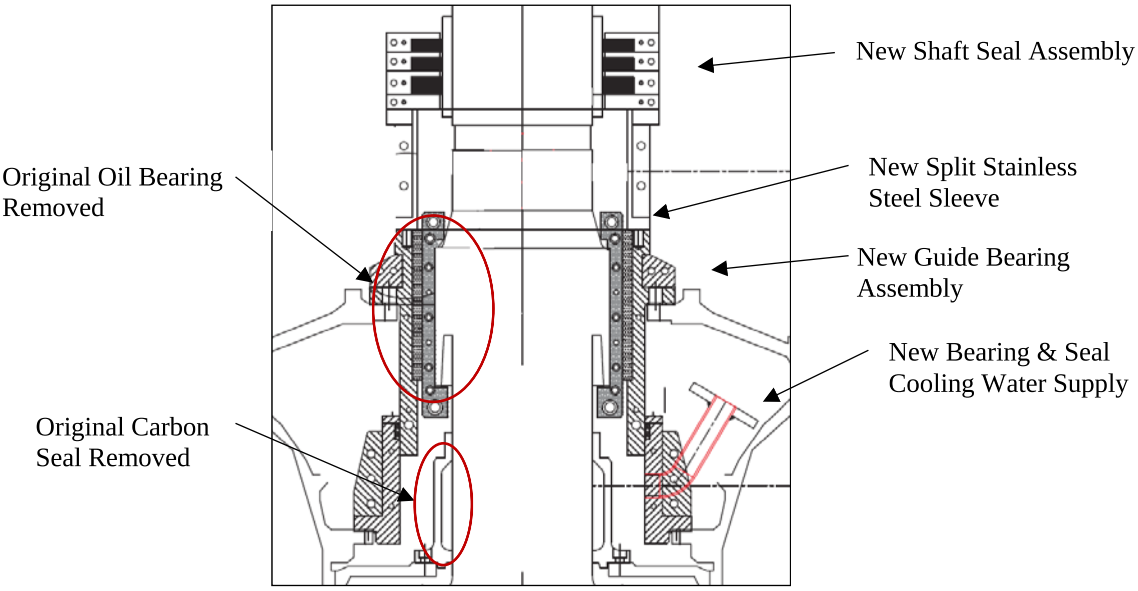

Figure 11 - Water Lubricated Bearing Design

Rather than securing the elastomer bearing material by bonding, in this project the bearing was installed 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 problems that may be encountered in the future.

The filtered water system configuration was also slightly different, with water entering the assembly below the bearing and moving upwards through the bearing to the shaft seal. 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 required through the bearing space to cool and lubricate the bearing.

Figure 12 - Thordon SXL with Tapered Keyset

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 must be correspondingly made larger in diameter.

4.3 Final Result & Current Status

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

Figure 13 - Complete Assembly

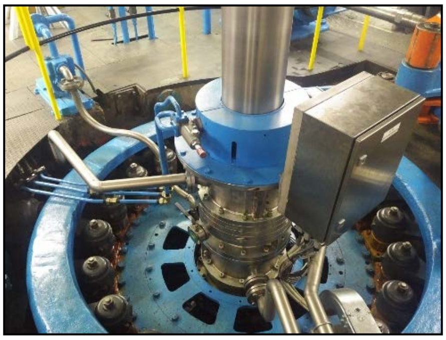

Figure 14 - Installed Bearing & Seal

Figure 15 - 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.

5 Recommended Approach for Considering a Water Lubricated Bearing Conversion

5.1 Feasibility Review

Many turbines are suitable candidates for conversion to water lubrication, but older turbines that are encountering on-going problems with the oil lubricated Babbitt bearings and/or shaft seals are probably the best choices for conversion to water lubrication as part of a planned overhaul or refurbishment project. The general approach to review the feasibility can be summarized as follows:

a) Identify units with planned refurbishments occurring within a 1 – 5 year time frame
b) Identify units encountering problems associated with turbine guide bearing and/or shaft seals
c) Prioritize units based on severity of problems and closest to planned refurbishment/overhaul
d) Gather drawing sets for candidate units (general arrangement drawing, shaft drawing, turbine guide bearing and shaft seal detail drawings)
e) Prepare technical data for each candidate unit (MW output, turbine head, rpm, load data if available, existing bearing clearance, runner clearance, sealed pressure below shaft seal)
f) Initiate discussion with the bearing supplier(s) and/or workshop(s) with previous oil to water bearing conversion experience and request concept proposals and budgetary pricing

This process should be able to be completed in 3 – 4 weeks, assuming that good operational and maintenance data is available for the power stations being reviewed.

5.2 Data Collection & Rough Scoping

Once discussion is initiated with possible supplier partners, there will often be missing information that needs to be gathered. After selecting several candidate turbines and a basic design concept is agreed on, the end-user should work on gathering any additional documentation and data from the operational units that may be required in order to allow the suppliers to refine their technical proposals. At the same time, the bearing supplier will work on preparing budgetary costing estimate, defining the project scope, and advising lead time required to manufacture and supply the required components. This step could normally be completed in 3 – 4 weeks.

5.3 Preparing Specifications, Gathering Formal Quotations & Initiating the Work

After the concept is agreed upon and the technical solution is refined to a point where it can be estimated accurately, then the end-user will need to prepare specification documents that can be used for seeking bids for the work from any qualified supplier partners. This process could normally take from 1 – 3 months, depending on the value of the project and how it may fit within the scope of a larger refurbishment project.

5.4 Implementation

Once the contract is awarded, the supplier will finalize the component design and begin manufacturing the required items. The basic scope of supply for converting an oil lubricated bearing to a water lubricated design will typically include:

- fabricated steel bearing housing
- non-metallic water lubricated bearing
- shaft seal assembly
- corrosion resistant shaft sleeve (or metallization)
- water filtration/conditioning system
- temperature and shaft position/vibration sensors (allows for tracking of bearing clearance)
- additional hardware or instrumentation requested by the end-user

Supply of these components would typically take 2 – 4 months depending on the size and complexity of the design. The actual downtime to perform the conversion on the unit depends on other work that may be on-going, and whether the shaft is being modified in place or at a workshop. Installation of the bearing and shaft seal assembly can be done in 2 – 3 days, once other major components are in place. Commissioning of the water filtration/conditioning system and shaft seal is normally done within several days after unit start-up.

6 Conclusion

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, then evolving toward rubber, and eventually utilizing advanced synthetic elastomer materials. The water lubricated bearing technology is suitable for Francis or Kaplan turbines in vertical or horizontal configurations, assuming that careful thought is given to the critical design considerations presented here.

As illustrated by the two case study projects, it is clear that conversion of turbine guide bearings from oil to water lubrication is a technically and economically feasible solution to reduce maintenance and solve on-going problems with older oil bearings or shaft seals. In both installations, the work was coordinated primarily by the end user with the input and guidance from supplier partners including Thordon Bearings Inc. and local workshops with expertise working on hydro turbines.

With careful planning and review of problems with 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 these recent conversions in Italy and Spain, Thordon has continued setting the example globally and demonstrating that a well-engineered water lubricated design can eliminate risk of turbine oil leakage, as well as solve ongoing maintenance and repair problems associated with older equipment.

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