A Case Study of an S-Turbine Seal Failure

By: Michael Detenbeck, P. Eng., Hydro Tech Inc.

Site: Energy Ottawa New York, Dolgeville Generating Station

EONY Manager: Murray Hall

Headwaters of the Dolgeville Power Station

Looking down penstock towards powerhouse

Powerhouse

ABSTRACT

Dominion Bridge and Sulzer partnered to introduce a low head horizontal Kaplan “S” type turbine into the North American market in the 1970’s. These units were intended for generating capacities up to about 10 MW. It appears that there are somewhere in the vicinity of 150 units in service in Canada and the United States of America.

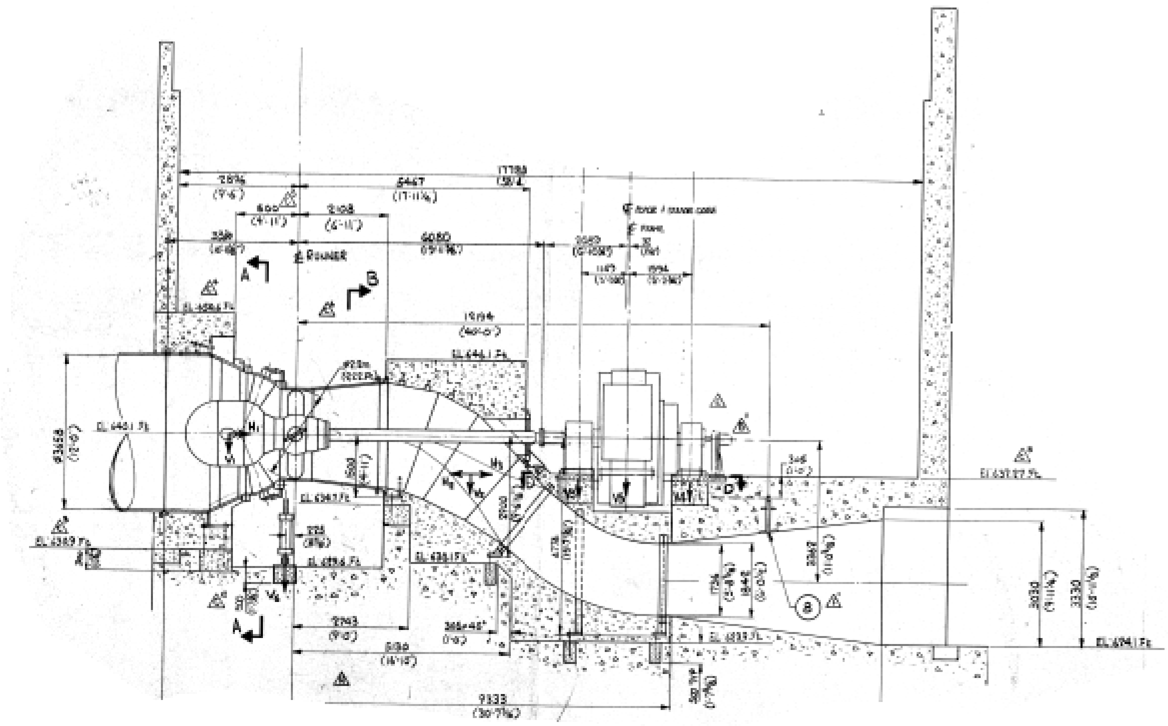

Figure 1 – Elevation Plan

These units perform well generally, but many owners have experienced short life spans and unreliable performance of the turbine runner water seal. This is a costly problem as the unit needs to be de-watered in order to change out the seal. As well, some of the components are long lead-time items creating extra costs associated with inventory of on-hand spare parts.

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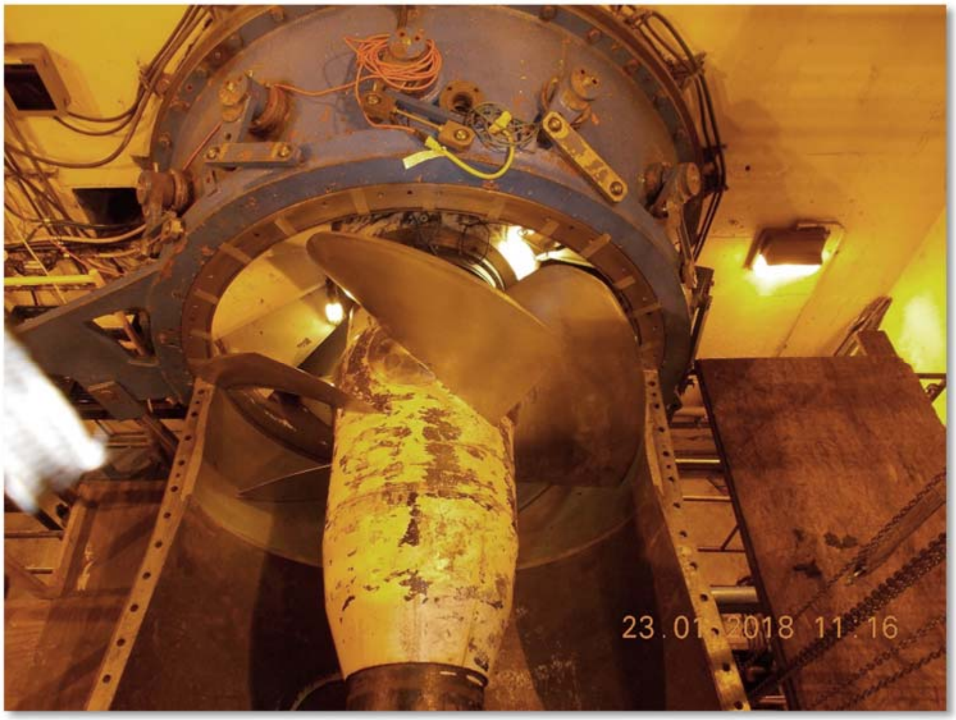


Photo 1: Turbine with discharge ring removed for access to seal area

Hydro Tech was called to the Dolgeville Generating Station in January 2018 to review a history of seal failures that occurred from 1987 through 2017. The objective was to see if there was an engineering solution available to keep seals functioning for a reasonable time frame before needing to be replaced.

The S-Turbine turbine currently operating at the Dolgeville site was selected in 1984 and commissioned in 1987. The sacrificial seal element for the runner hub seal was asbestos based. The original seal material had a modest life expectancy that needed improvements. When health and environmental concerns that were followed by strict government regulations surrounding use of asbestos were introduced, it was easy to make the decision to change materials to see if life expectancy could be improved. This horizontal water S-t Turbine has seen numerous resin-composite style seals installed only to fail prematurely due to overheating and extensive pre-mature wear. Asbestos has high temperature resistance and on start-up it is tolerant of the lubrication system breaking down temporarily, so it was not unreasonable to expect a reduced life when introducing alternative composite products. These compound changes to address the asbestos issue did not provide the desired improvement on life expectancy.

ORIGINAL DESIGN

Underlying Issues

The designers relied upon lubricating and cooling the seal face by extracting penstock water. While economical in the first instance, any blockage of flow would cause the seal element to contact the thrust face, heat up and wear prematurely. River silt would create grinding action between the contact faces and could plug up the passageways.

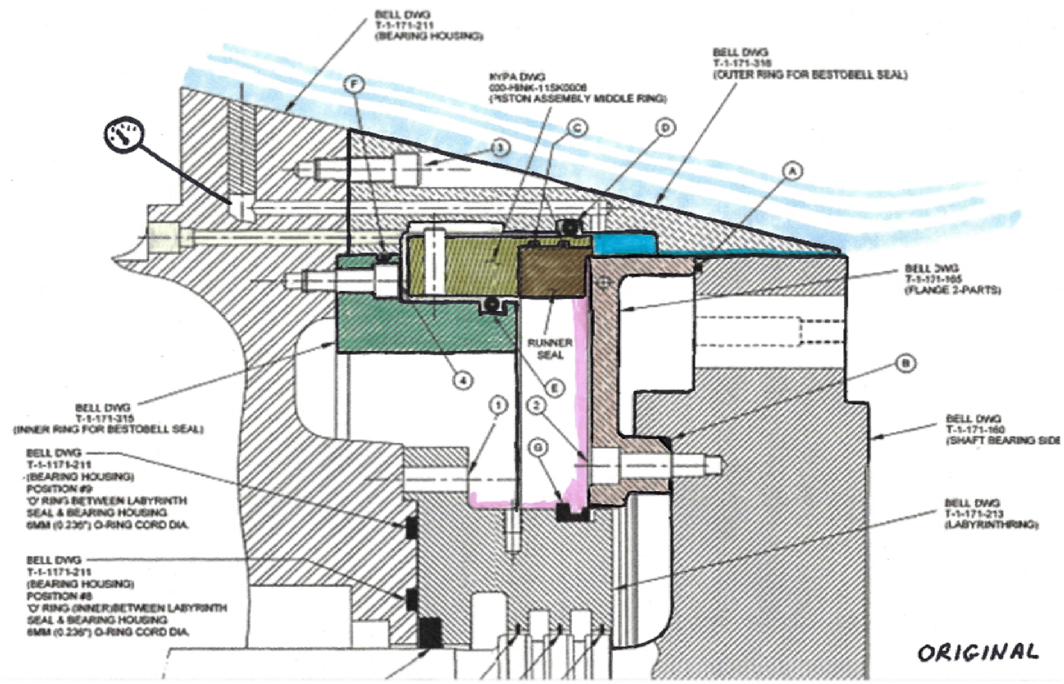


Figure 2 - Original

Water ports to positively lubricate the seal were not designed into the stationary bearing housing as space for this was not readily available. The original engineering process relied on large clearances between the runner shaft and the outer seal ring, thus allowing river water to pass over the runner shaft flange, under the seal ring, across the seal face, and into the drain. Under start-up conditions, and during episodes with high particle counts in the water, one can see how quickly the seal contact face can lose lubrication and cooling.

Another factor in designing ***hydraulically*** actuated seals of this nature is the maintenance of consistent contact pressure at the seal face. Seal pressure in this design was achieved by tapping into the penstock to utilize the river head pressure. This pressurized water activates the piston that applies pressure to the seal element This penstock water was passing through a small duplex strainer. It is worth noting that air entrained water is a compressible fluid and is normally not used for ***hydraulic*** applications without some rigorous thought put into it. Particulate laden water is definitely not a good thing.

There is another major mechanical design issue that reduces the ability to maintain pressure in the piston chamber. Since all the seal components had to be split this proved to be a weakness. These components had to be split so they could be installed around the shaft after the runner and runner shaft were installed. Not only was this a problem to keep pressure in the chamber, it was also a mechanical nightmare to assemble successfully.

Construction Faults

Inner Seal Ring:

The inner seal ring was made from cast G25 carbon steel and painted on the exposed surfaces but not in the piston chamber area and O-ring seats. One pair of bolts and one pair of locating pins keeps this part in alignment. Note the position of the O-ring groove.

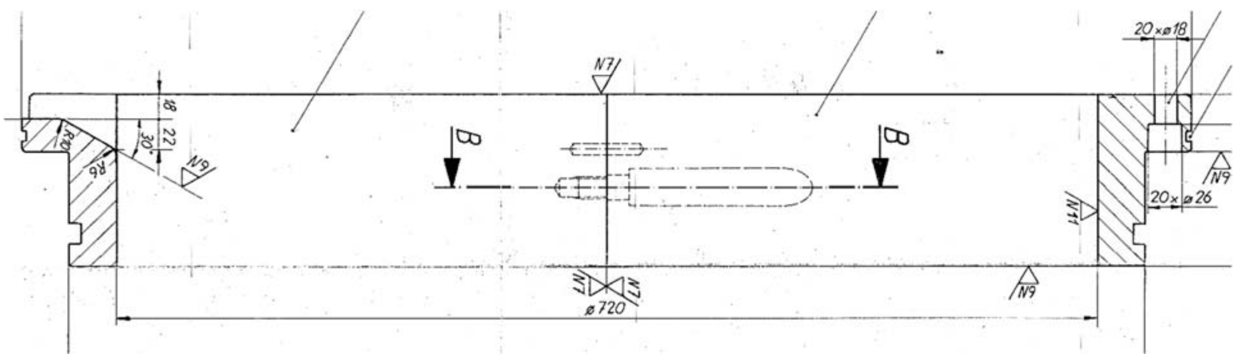
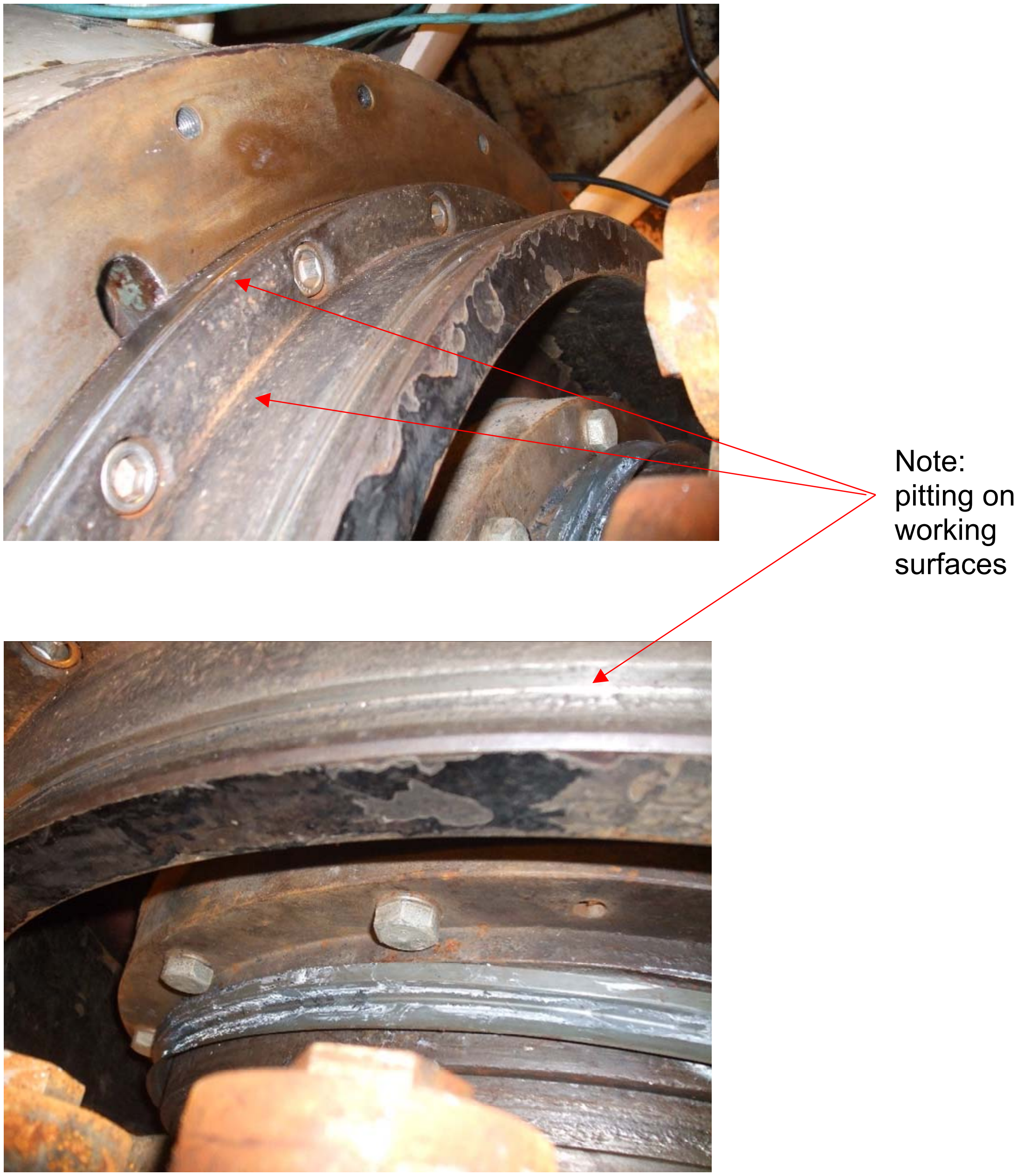


Figure 3 – Inner Seal Ring



Photos 2a & 2b: Inner seal ring cleaned and installed

Imagine this part having water sit stagnant in the O-ring groove and on the outer surfaces where the piston slides. Corrosion will always be the enemy here, eating away at the O-ring lands and binding up the movement of the piston.

Piston:

The piston was also of split construction and it relied on an outer O-ring and an inner O-ring to maintain seal pressure. Unfortunately, the O-rings are offset and create a cantilever beam that allows the piston to bend slightly, enough to relieve the pressure on the outer O-ring which, in turn allows water to escape and pressure to drop. The split made it hard to install the O-rings without creating a leakage path or pinching. This combined with trying to glue the O-rings and not get lubricating grease in the bonding area, was extremely difficult.

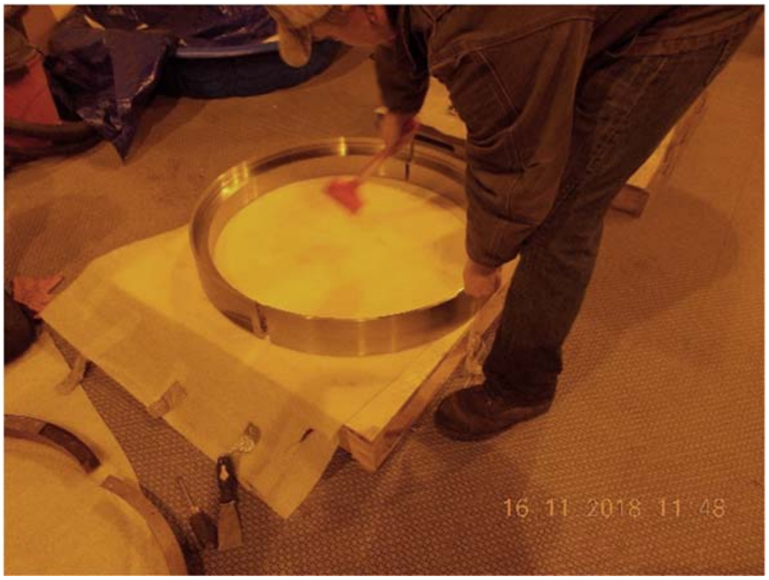


Photo 3: Trial fitting piston before installing

Outer Seal Cover:

The outer seal cover was split and tapered in a shape that eliminated the possibility of equalized clamping across the face. There was only one pair of bolts holding the halves together and it was near the mounting flange at the bearing housing. On our first visit, it was noted that this cone was twisted on the centreline axis. Inspections later verified it was out of alignment front and back by 0.6 mm. This was caused by the seal grabbing the outer seal housing and causing it to open and distort at the split line.

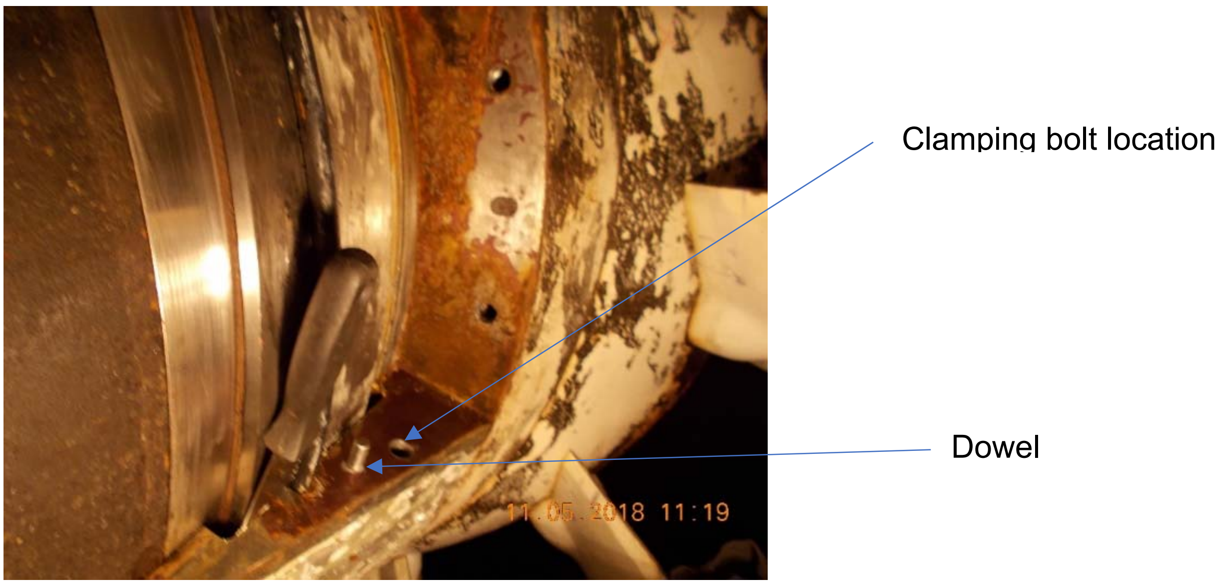


Photo 4: Lower half of outer seal cover

Flange Ring:

The flange ring (2 part) was split and relied on dowel pins to maintain alignment of the two halves, with flange bolts to hold it true to the shaft. This one dowel could not be an interference fit due to the method of installation. Also, with only one dowel, the flange split could rotate on the dowel creating a step on both the inner and outer edges. Also, there was no provision to prevent the split line from opening due to seal friction pressure.

Also, noteworthy, once the seal material started to wear, the material started to gall the seal steel face material. One should note that the damaged originally polished steel is now acting as a grinder on the seal.

Photo 5 – Gap and sharp edges act as a scraper on the seal element

Photo 6: Burn marks on thrust flange.

Metallurgical Issues:

The unit had been overhauled recently and experienced two seal failures during commissioning. The metal seal components showed evidence of damage sufficiently great enough that repairs and replacement of these components were in order.
The original flange was made from a German stainless steel. The flange material had been changed over time and was found to be 316L stainless steel. The piston was originally carbon steel and had been changed to 316 stainless steel. The outer seal cover was a cast steel of an unknown European grade.

SEAL ELEMENT INVESTIGATION

To obtain a full understanding of the mechanisms of failure, a complete set of operating logs, working drawings and quality assurance documents were collected and reviewed for significant events, fits and tolerancing issues. This process was hindered by flood damage in 2005 where most of the manuals and documents had been lost.

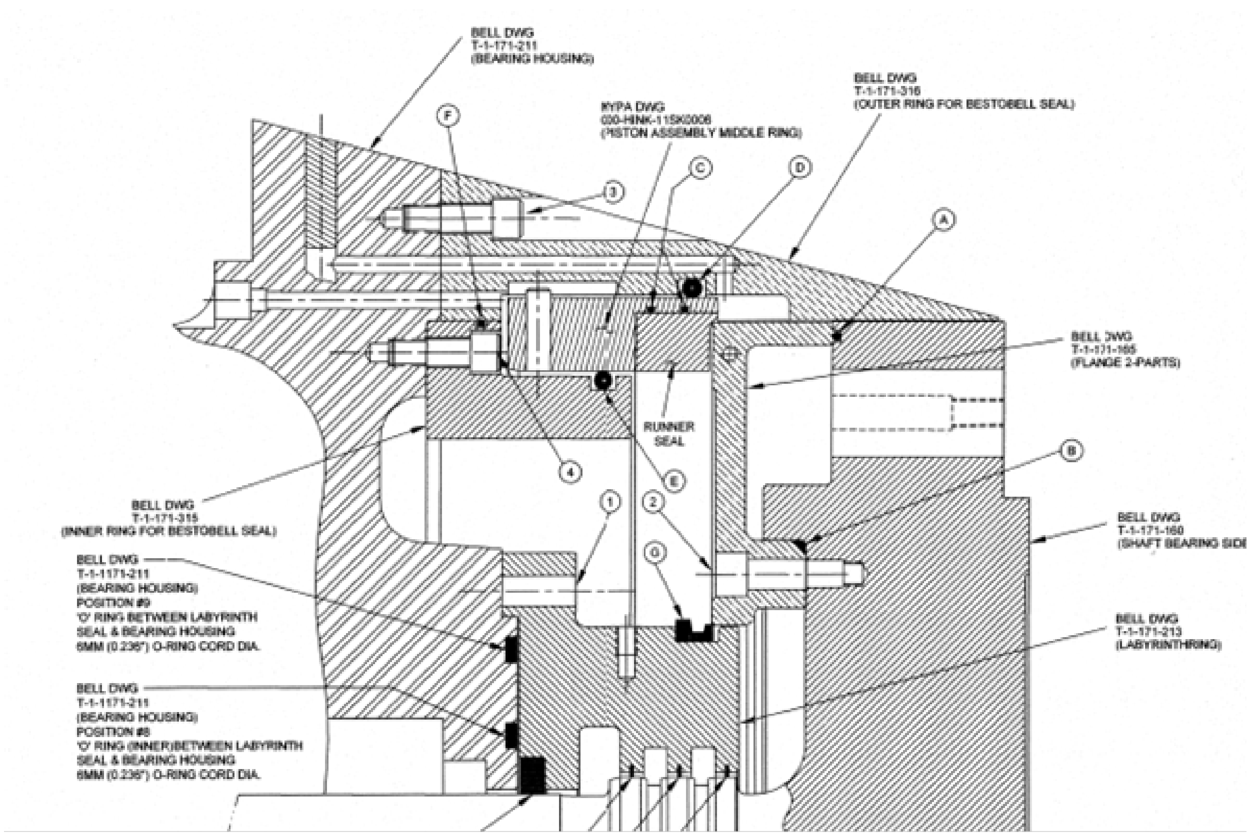


Figure 4: Section View of Bestobell Seal

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Seal replacements based on the purchase records from the OEM were ten (10) seal elements over a (31) thirty-one year period. The operating time on these seal elements was less than 25 years. The average operating life was approximately 2.5 years to failure, and in the last three years the life was down to days:

OEM Sales Record:

- 24358/L; containing asbestos – initial seal element 1987
- 24358/L: replacement 1994– obsolete material in 2001
- 24358/L/C: replacement 1999, 2001, 2003 – carbon fiber/yarn, graphite impregnated & resin
- 24358/L/H :(2) replacements bought 2005 – Kevlar component added
- 24358/L/D: replacements 2009, 2015, (2) in 2017, (2) in 2018

L/D material is basically a cotton type yarn passed through resin and wound on a mandrel.

In addition, in 2017 another manufacturer provided one or two seals made from a polymer base substance that failed during commissioning.

Records of commissioning commencing from 20 June 2017 were available. A turbine specialist company was hired to do the work and provided an aftermarket seal element. This seal element was installed and immediately began to overheat and smoke. The unit was shut down on 21 June 2017 without going on-line. The unit was re-started at a restricted speed of 50% during no-load conditions and put on-line. On June 25 a leak arose in the lower stay vane area (*Figure 5*). It turned out that the seal had fragmented and was clogging the water drain (*Photo* 7).

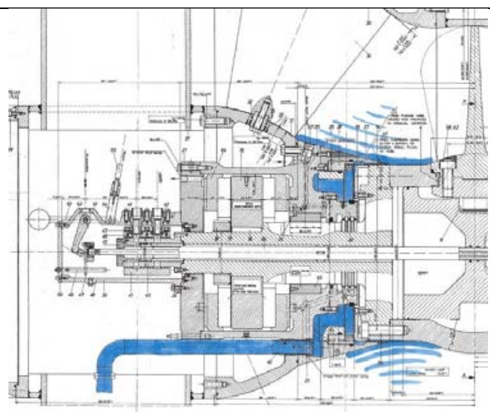

Fig 5: Drain path

Photo 7: Seal water flow indicators

In October 2017 another attempt to start the turbine/generator unit occurred. The first seal melted at 25% loading of the generator. A second seal was installed but lasted only 2 minutes and 57 seconds.

Photos 8 & 9: Failed aftermarket seal 2017

CUSTOMER REQUIREMENTS

As the repairs to this power plant were being closely monitored, some restrictions were placed on Hydro Tech in coming up with a solution. Also, some historical documentation from the seal manufacturer, the previous repair team and an alternate seal manufacturer were presented to aid us in coming up with a workable solution.

- The supplied solution shall be reversible so the unit can be put back to original.
- A vacuum seems to be present on start-up and shut down causing the seal to run dry.
- Work with Bestobell to come up with a solution they agree with.
- Inspect and witness the repairs being attempted on the piston, inner and outer seal covers and 2-part flange with the onus on re-using these parts if possible.
- Minimize any changes to the bearing housing.
- Keep the chamber inside the seal from being flooded as this compromises the oil seal on the runner stub shaft.
- Use the existing control system, modifying it as needed.

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ENGINEERING REVIEW

Best industrial practices outside of the hydro power industry use centripetal forces to send lubricating fluids across the seal face from the inside of the seal to the outside. Positive spring pressure keeps the moving element in enough contact with the mating surface to prevent excessive flow, while keeping boundary lubrication present. In the original design this does not happen; the water comes from the outside to the inside and relies on water contained in a hydraulic cylinder to pressurize the seal for reduction of water flow.

Seal face theory suggests a hardness difference between materials, with the harder material surface finish polished to work successfully. Usually the hard surface is fixed, and the soft surface is rotating. There are exceptions to this and conditions on these exceptions. The split line must be invisible to the seal element or the edges dig in, shearing the seal material and pushing away the lubrication.

The cooling and lubricating fluids should not contain anything that can act as a grinding compound as this affects the performance of the seal boundary. Silty river water can act like sandpaper, quickly compromising the performance of the seal face.

These engineering principles were not accounted for in the original design of the seal, and retrofits around the original assumptions became hard to accomplish.

ENGINEERED SOLUTIONS

The necessity to achieve water flow across the seal became the priority we needed to address first. Bestobell provided the flow rate and we tapped into a city water line to achieve this rate. We set up pressure and flow monitors on this for verification and tied it into the original control system.

We connected the city water supply to the pressure sensing port on the bearing housing. This port previously monitored the draft tube water pressure in the distributor housing.

We installed a turbovent to provide air into the distributor housing which would relieve any vacuum pressure that could occur during start-up or shut down. We set up the draft tube pressure gauge on this piping.

The turbovent was designed as a 3” vent line. As this line may be subjected to a full vacuum (30” HG or up to 30 psi), it was fitted with a shut-off ball valve, a sinking ball check valve, a silencer and a strainer basket. Its purpose was to feed station air into the water stream during start-up automatically and on demand.

Photo 10: Turbovent piping

Several claims during the overhaul in 2017 were made that the seal failures were caused by a perceived vacuum condition occurring during start-up. The evidence to support this was a pressure gauge that the needle was pegged past zero psi. This condition was thought to diminish the water flow across the seal face.

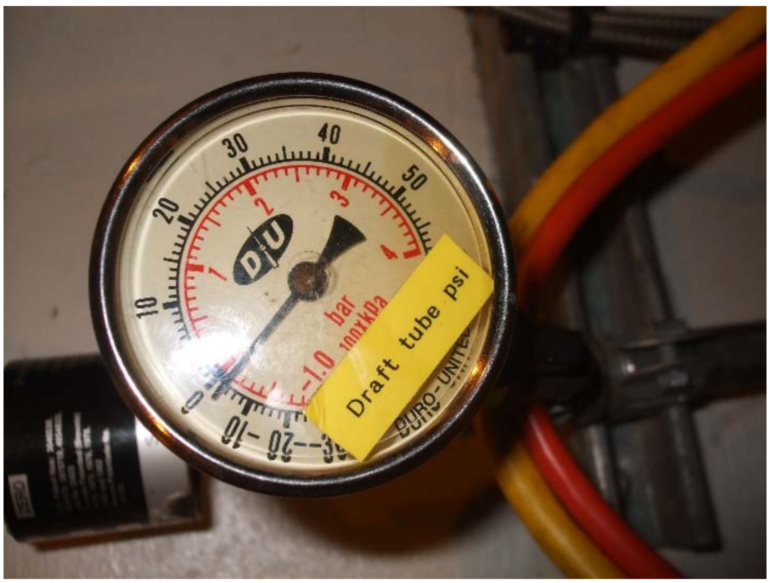

Photo 11: Draft Tube PSI

It was noted that when the seal wears and the piston is extended, it closes off the opening to the old pressure port. As we were using this port to provide water, we modified the outer seal ring by cutting in a slot and closing the clearances by adding a bronze band. In hindsight, it turns out that when the piston closed over this port, when in original factory condition, a vacuum was occurring that was being read as negative penstock pressure.

We added a bronze band to the outer seal cover to close the clearances around the runner hub.

Photo 12: Slot in outer seal cover and bronze band

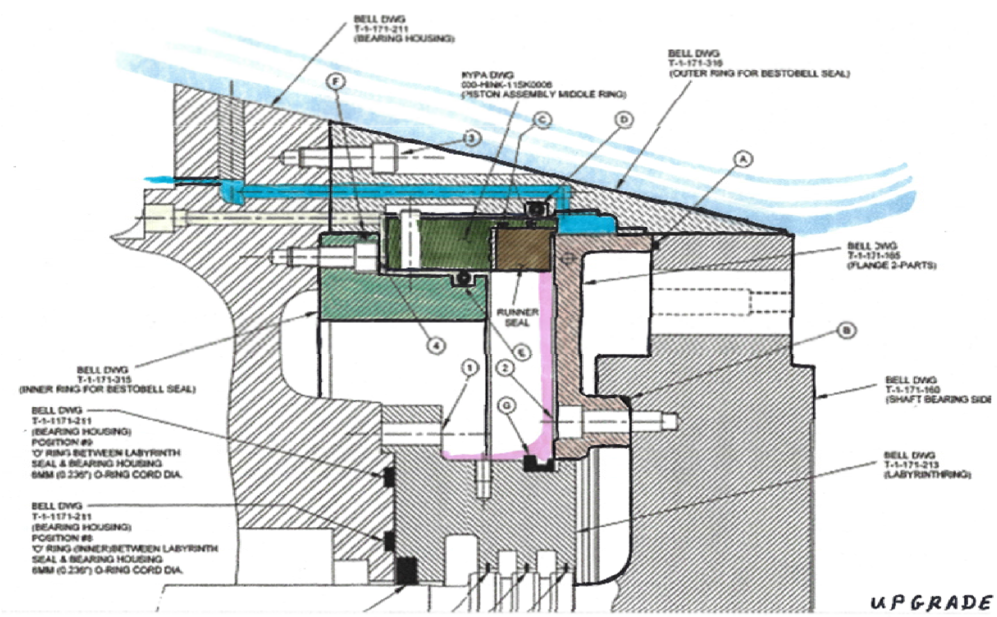

Figure 6: Upgrade of seal lubrication

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FIRST TEST MARCH 2018
Putting the equipment back together as close to original as possible.

It was proposed that the original manufacturer's seal element would be installed. The entire set of seal components was measured. Then the clearance, fit, and finishes were put back to the 1980's drawing specifications. The seal piston controls were repaired, and the float switch system put back into working condition. Some compromises were made so components could be salvaged and avoid replacement costs.

We verified the dimensions of the seal components and made some adjustments to fit tolerances and surface finishes in the machine shops. Then we fitted the seal components on the unit as best we could back to original specifications. The outer seal cover had been damaged during the last seal failure and an attempt to fix it with a weld repair (prior to Hydro Tech's involvement) had left the outer seal cover in a distorted condition. We attempted a repair to this cover when we added the bronze band. No attempts were made to change any of the additional mechanical issues as they had not revealed themselves as causes of seal failure up to this point.

Photos 13: Blocked pressure monitoring port

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Photo 14: Damaged slot for anti-rotation pin

A thorough study of the startup procedure was carried out. During the life of this unit some design changes to seal controls were made. This complex control system provided a method to maintain water pressure to the back face of the seal piston and a rudimentary float switch system used to monitor water discharge from the drain of the casing. Both components had feedback to the turbine control system and alarms to indicate functionality.

Photo 15: Seal control panel

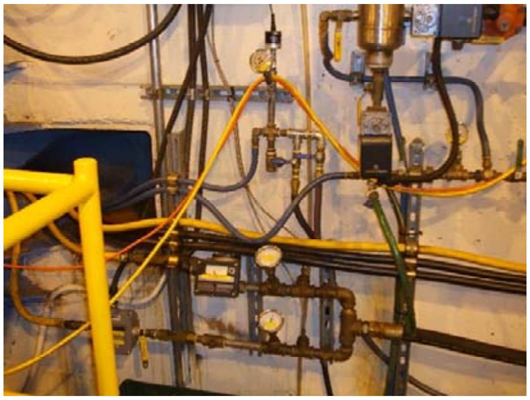

Photo 16: Air/water distribution headers

To remove the potential that silted river water was a factor, this additional supply was necessary. Because city water supply volume was limited and was also used to cool the bearing oil through a heat exchanger system, a secondary source of water was established. This secondary supply came off the penstock and was filtered through a strainer. This filtration system was reconditioned.

Clean water flow across the seal face was achieved. In order to verify this, we installed flow meters and pressure gauges on the piston water supply and on the seal face water supply.

With tremendous support from the EONY personnel, we began to rebuild the seal components to get this unit online.

On 3 May 2018 the seal assembly work was completed, and the unit commissioned for start-up. Within 4 days (7 May 2018), the seal had worn down to 3 mm remaining and the turbine shut down.

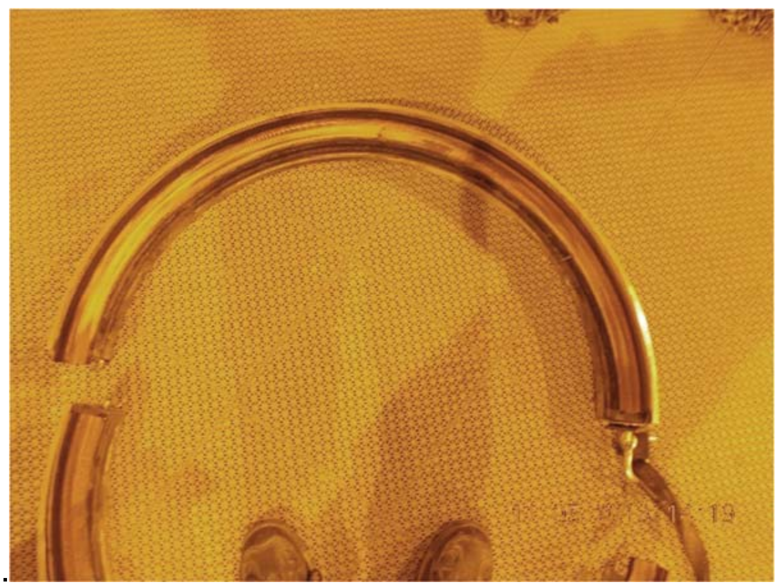

Photo 17: Worn L/D seal element in piston

Photo 18: Worn L/D seal element after removal from piston

FAILURE INVESTIGATION MAY 2018

A failure Investigation began on 14 May 2018 and the resulting changes were made:

- The 304 stainless steel flange ring was replaced with 400 series stainless steel, heat treated to give better wear properties.
- We replaced the Bestobell element with a natural fiber wood (Lignum Vitae), which has much better properties for sealing on rotary faces and complements the choice of 420 or 410 stainless steel. Lignum vitae is a wood that has natural lubricating properties. While Lignum Vitae should always be kept wet, it is able to self-lubricate should water lubrication and cooling conditions be marginal.

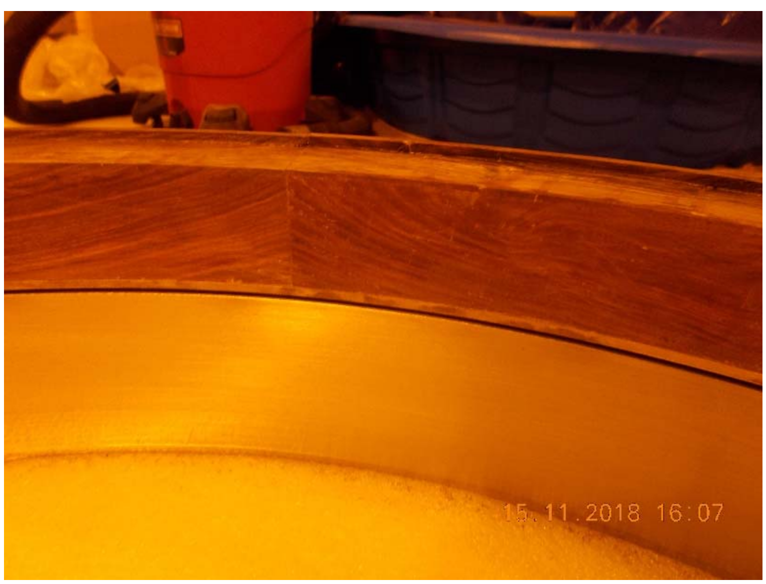

Photo 19: Lignum Vitae segment glued

- We ran laboratory tests to verify Lignum Vitae would outperform Bestobell material. Test results are discussed later in this document.
- The runner flange split line does not have a clamping arrangement to hold the splits together. Since we are using harder materials for the sealing surface, any misalignment of the split line could result in shear on the seal element and high torques can be achieved through the components and the anti-rotation devices. To correct this, we added two additional dowel pins and bolts to the split line surfaces. Proper surface finish techniques, surface grinding techniques and alignment of the split line alleviated the shearing effects and pickup.
- We provided a bronze backing plate with the segmented Lignum Vitae seal elements to retain the sections and to prevent rotation of the element in the piston cavity.

Photo 20: Segmented Lignum Vitae seal element with bronze backing rings

Photo 21: Seal element trial fitted into piston

- The turbovent provided the evidence that ruled out vacuum conditions.
- We fitted the Lignum Vitae seal element into a 304 stainless steel piston.
- Hydro Tech manufactured a new outer seal cover with additional split line fasteners so the conical end would not open under pressure and changed the material to 400 series stainless steel.

LIGNUM VITAE AS A VIABLE SOLUTION

Hydro Tech possesses an in-house material testing bench for measuring wear under both wet and dry conditions. This bench was manufactured using ASTM Standards for testing material properties and the test equipment itself meets ASTM requirements.

Testing in the Hydro Tech Laboratory

Excerpt Test Report Series 8

In this group of tests, we compared the wear characteristics of lignum vitae to Bestobell non-asbestos resin composite fibre 24358/L/D

In the application of this test we looked for a performance model that compares these materials in a seal face environment where the seal is in a fixed position and the wear face rotates against the seal. The key measurement is the P/V ratio [pressure/velocity], therefore we will look at wear under a constant speed with variable pressures to see how the P/V ratio is affected by wear.

The parameters of this group of tests are as follows:

- Comparison of lignum vitae versus Bestobell seal under flooded wet conditions
- A total of (8) runs were made
- The first set of (4) runs started with unworn samples, 1 of each material under a load of (2) weights.
- After the run-in period, the second run was made with zero weights.
- A single weight was added for run #3
- Two weights were added for run #4.
- Two new unworn coupons were inserted for a repeat of the first set of (4) runs.
- Runs (1) and (5) were under non-conforming sample conditions to establish break-in wear rates.

Results:

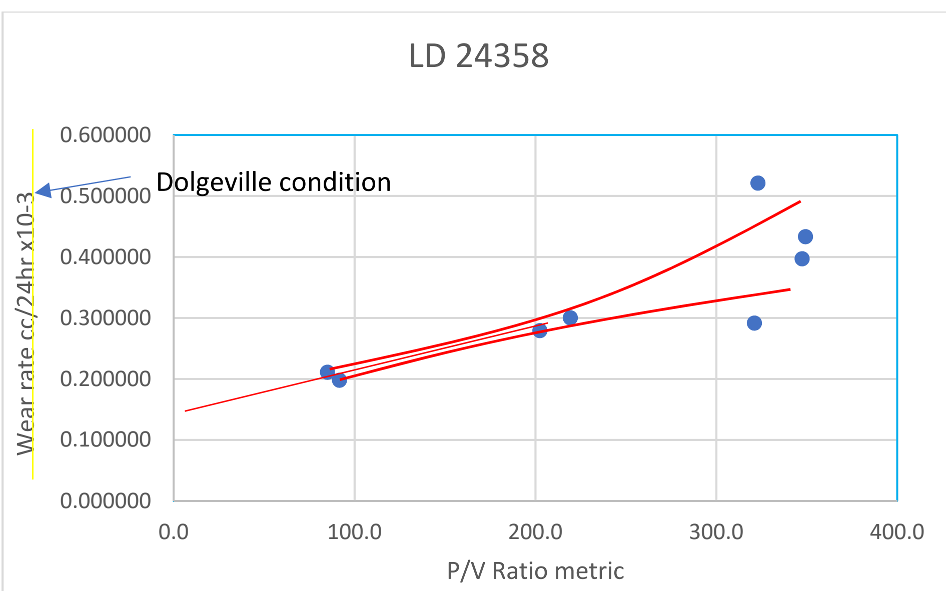

Figure 7: Test Condition - Flooded

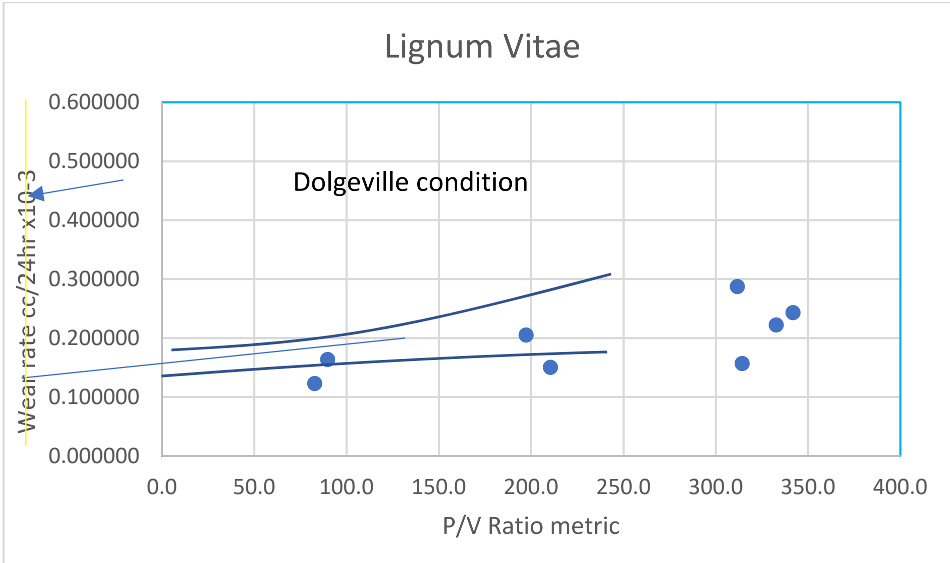

Figure 8: Test Condition Flooded

The results indicate that as the P/V value goes up, the LV seal outperforms the L/D material. In flooded conditions, under the P/V value the seal element is set up to run at Dolgeville, the wear characteristics favour the LV material only marginally.

We then ran the same tests dry to see what happens under no-water conditions.

Lignum Vitae – SM Seals 24358L/D Surface Finish Comparison

By: Seán Whelan Marcolini November 2018

Abstract:

After completing two sets of dry wear tests comparing two separate samples of Lignum Vitae and SM Seals 24358L/D material (a cresol-formaldehyde based fiber-resin composite), it was observed that the wear journals used for both L/D samples had sustained more surface damage compared to the Lignum Vitae. Measurements of the surface finish were recorded.

Procedure:

Completed two sets of dry-running ASTM D77 wear tests, comparing Lignum Vitae (LV) to SM Seals 24358L/D fiber-resin composite seal material (L/D).

The block-on-ring wear testing machine used for these tests has four sets of wear rings (made from 316 Stainless Steel) for four independent samples, with rings #1 and #2 used for LV, and #3 and #4 used for L/D. Surface speed for all rings was 0.23 m/s, on sample wear-area size of 6.6 mm x 15.85 mm. Each sample is independently loaded.

The first set of dry tests on wear rings #2 and #3 (given by three notches on the samples) occurred over a time period of about 60 hours, with four loads: an initial load of 14.2 kg for initial wear-in (about 24 hours), 6.5 kg for 12 hours, 10.3 kg for 12 hours and 14.2 kg for 12 hrs.

The second set of dry tests on rings #1 and #4 had more aggressive loading - a constant 37 kg. This test ran for 4 hrs.

Surface finish measurements were taken using a surface profilometer across the axial direction only (the tangential direction was too narrow for the profilometer to sit on the ring).

Comparison of the wear ring surface finishes after testing:

Note: the wear ring #3 surface finish was dressed before profilometer readings could be taken, but it exhibited the same deep circumferential grooves that were observed on ring #4.

Photos 22a/b/c/d: Test set #1: (60 hours, incremented loads)

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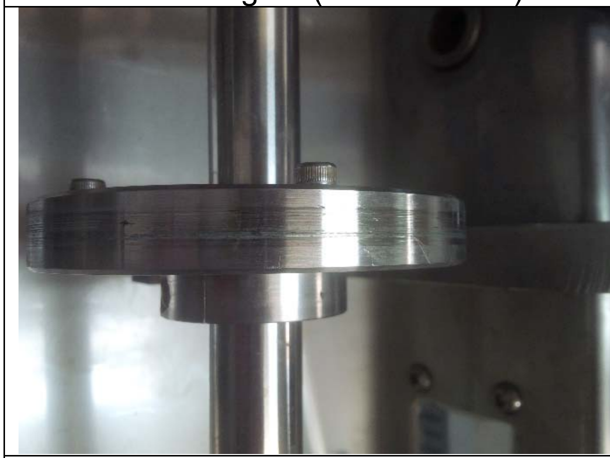
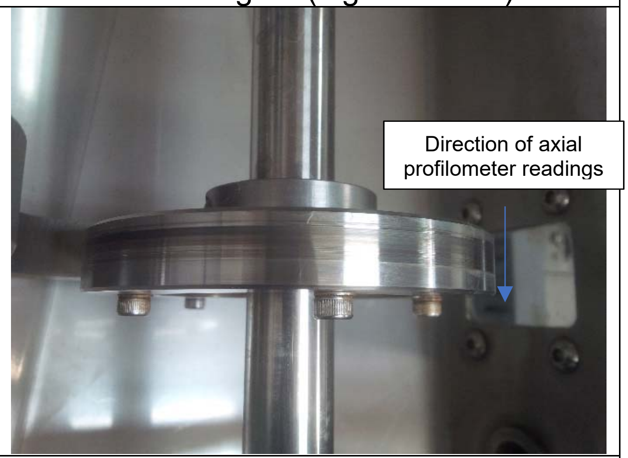
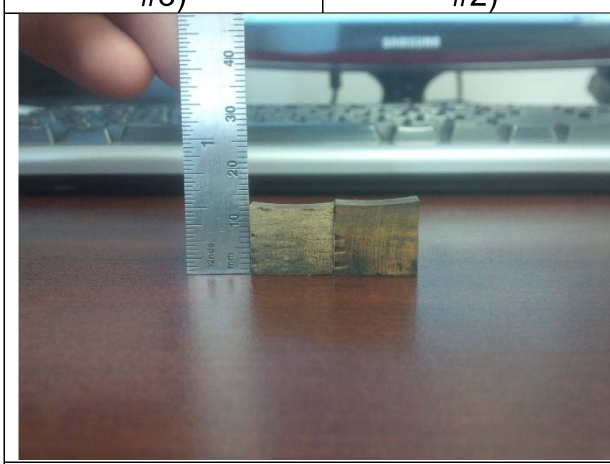
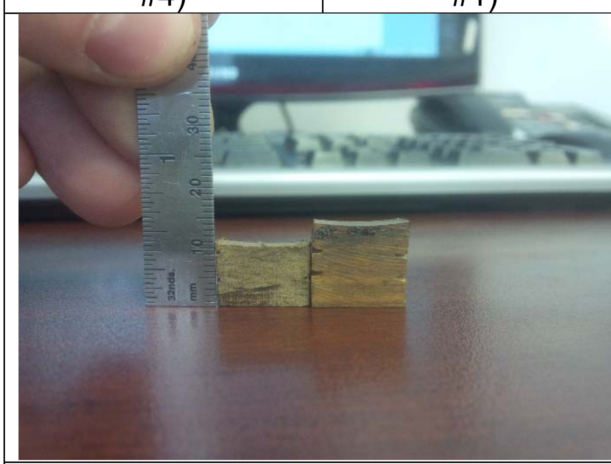
<tr><td></td><td>
</td></tr>
<tr><td>Axial Surface Finish R_z: (Na)</td><td>Axial Surface Finish R_z: 0.29 µm – 0.37 µm</td></tr>
</table>

<table>
<tr><td>L/D Sample (Ring #3)</td><td>LV Sample (Ring #2)</td><td>L/D Sample (Ring #3)</td><td>LV Sample (Ring #2)</td></tr>
<tr><td colspan="2"></td><td colspan="2">
</td></tr>
<tr><td colspan="2">L/D Sample Total Wear: 3.19 mm</td><td colspan="2">LV Sample Total Wear: 2.27mm</td></tr>
</table>

Photos 23a/b/c/d: Test set #2: (4 hrs., single large load)

Wear Ring #4 (L/D)	Wear Ring #1 (LV)
	
Axial Surface Finish R_z: 2.68 µm – 2.90 µm	Axial Surface Finish R_z: 0.33 µm – 0.39 µm

<table>
<tr><td>L/D Sample (Ring #4)</td><td>LV Sample (Ring #1)</td><td>L/D Sample (Ring #4)</td><td>LV Sample (Ring #1)</td></tr>
<tr><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td colspan="2">L/D Sample Total Wear: 5.75 mm</td><td colspan="2">LV Sample Total Wear: 2.11 mm</td></tr>
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Dry Test Conclusions:

These limited tests indicate that Lignum Vitae is much less abrasive to shaft journals then an equivalent material under dry conditions. One possible explanation might be that the L/D samples have been observed to produce large quantities of fine particulate, which may result in increased abrasive wear on the collars. Another explanation is the natural lubricating properties of Lignum Vitae.

Photo 24: Worn LD material ring #4

Photo 25: Worn L/V material under ring #1

- The results of these tests supported our view that a lignum vitae seal element would work well in this application.
- These tests supported the manufacturer's position that L/D material needs to be continuously lubricated and that damages can occur to the stainless-steel counter-face.

IMPLEMENTATION

- An eight-segment brass backed Lignum Vitae replacement seal element and a new 350 BHN polished thrust ring were manufactured to fit into the existing seal housing with some modifications to improve water flow and flow direction.
- The piston was re-used.
- The outer seal was made from 410 stainless steel.

This addressed most of the mechanical issues and metallurgical issues thought to contribute to seal failure.

Reversing water flow was ruled out as this would have required changes that would not be reversible.

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RESULTS

The unit was put back on-line 4 December 2018.

Water flow was good; water temperature was recorded (inlet temp 38°F seal water temp 42°F)

Hydro Tech had opportunity to inspect the seal in July 2019. The seal was in good condition, however, there was evidence of loss of water conditions. It appears that although the seal had been operated under harsh conditions, little wear existed.

Successful trials with over twenty stop and start events and over 1400 hours of operation have shown very little break-in wear (<0.08") and the remaining thickness of the seal element has not changed after break-in. This break-in period occurred within the first month of operation with little to no wear after.
In contrast, the OEM composite seal element, under the same load conditions and with the modified water supply, was destroyed after three days and six stop/start cycles.

Photo 26: Outer seal cover with no evidence of twisting or pick up

Photo 27: Lignum Vitae seal after 1400 hours operating time

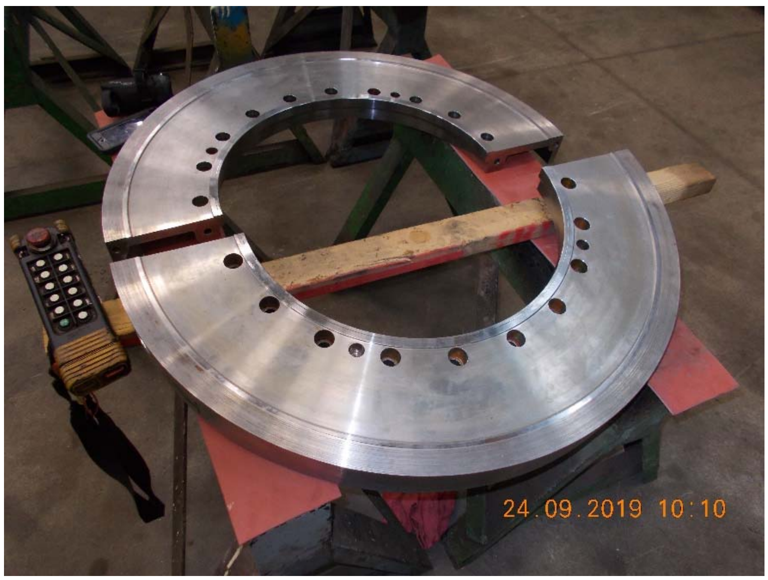


Photo 28: Flange ring Water Surfaces after 1400 hours operating time.

CONCLUSIONS

This paper has outlined the process of improving a turbine shaft water seal to allow us to offer solutions that have eluded power plant owners for decades. Lignum Vitae has proved to be a tough and reliable selection to solve marginal lubrication problems in horizontal Kaplan water turbines.

The results of the in-service period saw the average seal element wear at 2.51 mm and the percentage of wear 5.64%. Most of this wear occurred during the first month (the break in period). (ref.QA Document 272-0201-005-QA 6 Sep 19).

Start and stop episodes pose little difficulty in survival of this seal element and should prove over time the desirable life expectancies needed.

About Us

- Hydro Tech Inc. has offices at key border points between Canada and the USA, in Sault Ste Marie, ON and Niagara Falls, ON.
- US operations are conducted from Hydro Tech USA Inc. located in Brunswick, Maine.
- For the past 18 years, Hydro Tech has been designing innovative large thrust bearings for the hydro Industry.
- Hydro Tech is bringing its extensive knowledge of water lubricated bearings to correct and improve seal life in the hydraulic turbine sector using sound fluid dynamics and engineering principles.
- Hydro Tech also specializes in turbine and generator overhauls and mechanical design improvements for turbines and generator equipment, gate installations, and seals as described in this paper and sources.

Authors Credentials:

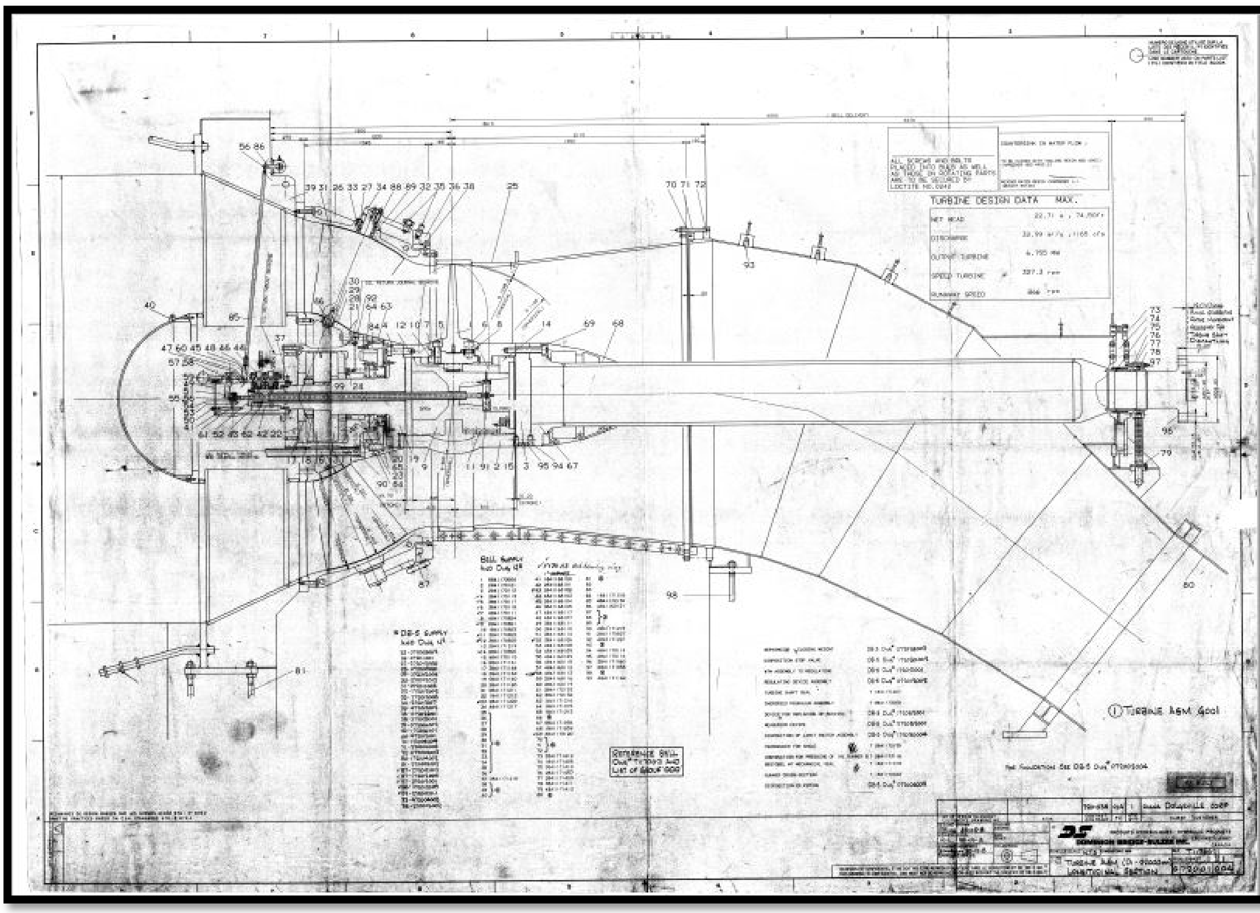
Michael Detenbeck, P.Eng.

- Senior Engineer - Mechanical and Metallurgical Expert
- 30+ years direct design and manufacturing experience for fossil, water, and alternative electric power generation systems in North America.

TABLE OF STATISTICS:

Conditions in January 2018:

- The turbine seal assembly has the thrust face mounted on the rotating component and the seal element on the stationary side.
- The seal is lubricated with dirty river water entering from the outer edge of the seal.
- The turbine runs at 327.3 RPM, seal water flow 1.25 USGPM minimum (single 3/8" dia. port). Seal pressure against wear surface is 19 psi.
- The thrust wear surface was 304 stainless steel
- The turbine works on a net head of 74.50 ft, output is 6.755 MW.
- The runaway speed is 866 rpm.



- The seal OD (working surface) is 33 11/16" and the ID is 31 5/16" nominal. This is the thrust face contact area. Seal nominal size 32 5/8” Dia. x 1 7/8” x 1 3/8”