Accelerated Lifetime Testing of Main Shaft Seals for Tidal Turbine Rotors

Miles Skinner,[1] Scott Lambert,[1] Robynne Murray,[1] and Jonathan Colby[2]

1 National Renewable Energy Laboratory
2 Streamwise Development

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Technical Report
NREL/TP-5000-89380
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List of Acronyms

BF	barrier fluid
BFLR	barrier fluid lower reservoir
BFUR	barrier fluid upper reservoir
Gen5	fifth generation
IEC	International Electrotechnical Commission
IECRE	IEC System for Certification to Standards Relating to Equipment for Use in Renewable Energy Applications
in.	inch
ISO	International Organization for Standardization
kPa	kilopascal
L	liter
m	meter
mL	milliliter
mm	millimeter
MRI	Manufacturing Resources Inc.
MSS	main shaft seal
NBR	nitrile butadiene rubber
NREL	National Renewable Energy Laboratory
psi	pounds per square inch
rpm	rotations per minute
RTD	resistance temperature detector
SI	service interval

Executive Summary

This report briefly discusses the observations and results from accelerated lifetime testing performed by the National Renewable Energy Laboratory (NREL) on the main shaft seal for the Verdant Power fifth-generation (Gen5) underwater tidal energy converter turbine, which successfully performed at the Roosevelt Island Tidal Energy project in 2020–2021. To evaluate a 5-year service interval (SI) for this component, the main shaft seal was operated nearly continuously for 137 days at a rotational velocity of 160 rotations per minute while the test stand recorded water pressure, barrier fluid pressure, temperature, and number of cycles, representing ~ 40% of the SI. An additional separate test was conducted to measure the aging behavior of the rubber drive rings. For the SI evaluation the water pressure reservoir was held constant as 199,9 kPa (29 psi). Barrier fluid pressure remained relatively constant throughout the duration of the test but fell to as low as 69.6 kPa (10.1 psi). No barrier fluid leakage was observed throughout the test. A sudden failure occurred within the seal after the power to the test machine was interrupted for a scheduled building maintenance procedure. Upon restarting, the main shaft seal lost all ability to prevent water ingress. The exact cause is not known but is believed to be either a seal assembly issue or a change in the alignment of the seal components during or following the power outage. Following seal disassembly, one of the graphite sealing rings showed significant wear. Verdant Power, Dovetail Solutions LLC, and Garlock Sealing Technologies reviewed the seal wear for consensus evaluation of results. NREL returned the seal faces to Garlock, and a review of them indicated a misalignment of the test stand, both overall (shaft moving as a whole) and from front to back (more movement on the water side than the air side). Garlock further indicated that minor misalignment is usually absorbable by the seals; so, the noted wear leads to the conclusion of a test stand disruption. Therefore, the operation of the Gen5 seals to the point of power outage is indicative of long-term performance. Based on these results, it is recommended that follow-on testing be conducted through NREL's Testing Expertise and Access for Marine Energy Research (TEAMER) program to rectify protocol and assembly issues to further evaluate the SI of this component.

The support and participation of Verdant Power Inc., Dovetail Solutions LLC, and Garlock Sealing Technologies were instrumental in understanding the results. NREL thanks them for their contributions.

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1 Introduction

The lifetime assessment experimental platform described in this report is specifically designed for testing the low-speed main shaft seal for Verdant Power's (Verdant Power Inc., New York, New York) fifth-generation (Gen5) underwater power generation turbine. This project is jointly sponsored by the New York State Energy Research and Development Authority and the U.S. Department of Energy as part of the Roosevelt Island Tidal Energy Project. The seal test stand was designed and built by Manufacturing Resources Inc. (MRI, Cleveland, Ohio) and originally put into operation by Garlock (Garlock Sealing Technologies, Palmyra, New York) for validation and initial testing in 2015 (Adonizio, Corren, and Calkins 2021). The seal passed validation testing on this test stand and was incorporated into four Gen5 turbines manufactured and installed by Verdant Power in the 2020–2021 time frame. A third-party power performance assessment of the Verdant Power Tri-Frame with three Gen5 turbines, (defined as the tidal energy converter) was conducted according to International Electrotechnical (IEC) Technical Specification 62600-200 Ed. 1 and was issued by the European Marine Energy Centre as the world's first IECRE[1] Test Report for the marine energy sector. This report verified the system performance, including the overall efficiency of the tidal energy converter. The opportunity for the National Renewable Energy Laboratory (NREL) to conduct accelerated lifetime testing of the main shaft seal using the modified MRI/Garlock test stand provided a unique effort to demonstrate wear and tear and predict the 5-year performance of an underwater component of the 5 m (16.4 ft) Gen5 turbine. For reference, the modified MRI/Garlock test stand is substantially similar to the description in Section 2.1 Operation Test Stand, with the only difference being the number and type of sensors.

This report discusses the test stand construction, instrumentation, and test methodology, followed by the observations and recorded data. Finally, a brief discussion of the results and their implications are presented.

[1] IECRE is the IEC System for Certification to Standards Relating to Equipment for Use in Renewable Energy Applications.

2 Experimental Methodology

2.1 Operation Test Stand

Figure 1 shows a diagram and photograph of the seal test stand. All the primary components are visible, including:

- The variable-speed gear motor, which is used for the power input to the system
- The main shaft seal (MSS), which is filled with the barrier fluid (BF) from the upper reservoir (UR) and lower reservoir (LR), necessary for lubrication and to separate the water from the rest of the components
- The pressurized water tank, which simulates the aquatic operating environment, and various sensors and valves necessary for operation and data collection.

This test stand closely follows International Organization for Standardization (ISO) 6194 Rotary shaft lip-type seals incorporating elastomeric sealing elements, which is intended for accelerated lifetime testing of lip-type seals incorporating elastomeric sealing elements. Some deviations were made from the standard: the BFUR volume was increased, and a sampling valve was added below the seal. These modifications were made to facilitate the periodic collection of BF to monitor particulate size and generation rate.

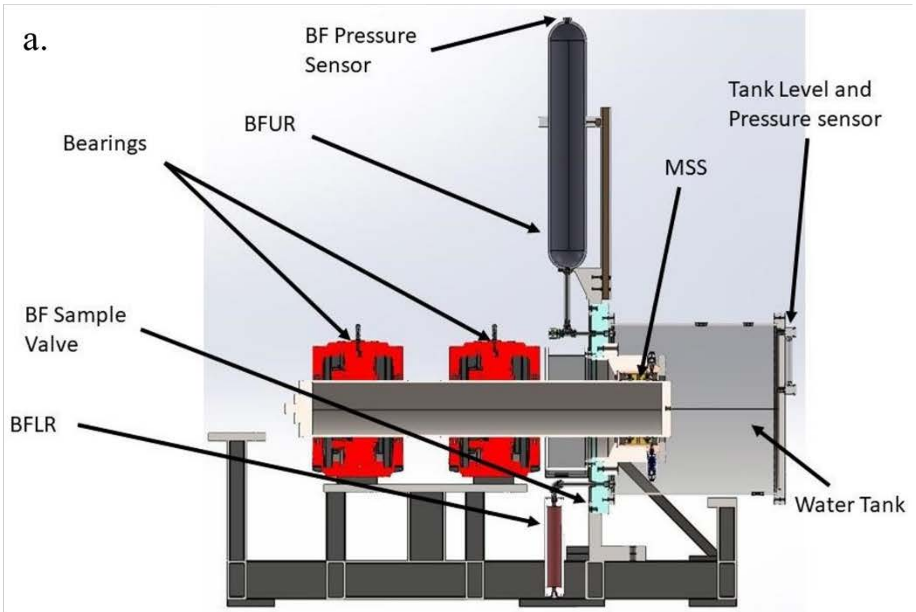

Figure 1. (a) Cross-sectional diagram of the accelerated lifetime MSS test stand with major components labeled. The electric motor, gearbox, and oil hoses are not shown. (b) Seal test stand as built in the NREL test facility.

Illustration and image by Miles Skinner, NREL

The electric motor and gearbox are designed to rotate the main shaft at 160 rotations per minute (rpm) with a duty cycle of 59.5 minutes on and 0.5 minutes off repeating 24 times per day. This is a duty cycle of 99.2%. This rotational velocity is 4 times the expected operational rotational velocity of 40 rpm, and the duty cycle is 6 times the expected operational cycle of four times per day. Under these operating conditions, the seal will complete its desired service interval of 5 years, or 78.8 million revolutions, 24 times faster than operational counterparts. The bearings are

SAF534 pillow block bearings (SKF, Gothenburg, Sweden) (SKF 2023), which support all rotational and translational forces except for axial rotation and thrust loading.

Figure 2 shows a detailed view of the MSS with all primary components. The water tank was pressurized to approximately 199.95 kilopascals (kPa) (29 psi) to simulate the 20.45 m (67.1 feet) water depth. Although this is greater than the expected operating conditions, it was deemed acceptable because it exceeded expected loading and would accelerate any leakage that may occur during testing. The BF was supplied to the MSS through the BFUR, and the BFLR provided a space for leaked water to collect and oil samples to be taken. Hoses connect the reservoirs to the housing.

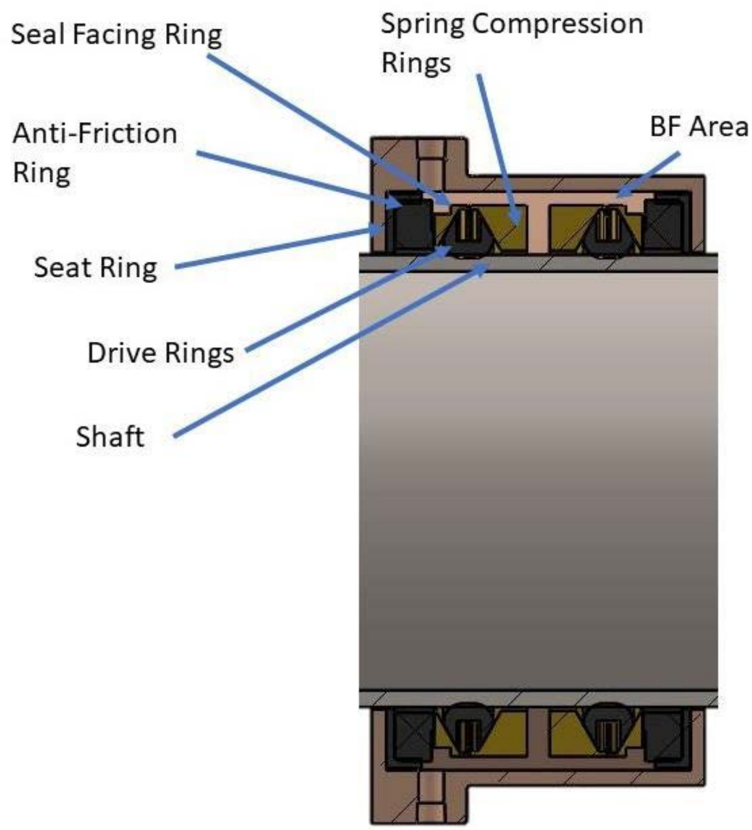

Figure 2. Cross section of the MSS and housing showing primary components. The seal is symmetric between the spring compression rings. Four springs in situ press the rings apart. The interior space is filled with BF with liquid-tight seals at the drive ring/shaft interface and the antifriction ring/seal-facing ring interface. The front, on the left, is upstream and faces the water tank. The back of the seal, on the right, is downstream and faces the atmosphere.

Illustration by Miles Skinner, NREL

The water tank was filled with 56.8 liters (L; 15 gallons) of fresh water. While the Gen5 tidal energy converter is expected to operate in brackish water, fresh water was chosen to minimize the risk of damaging nearby equipment, facilities, and surrounding laboratory environment in the event of significant seal leakage. It is recognized that potable water is less corrosive than the seawater or brackish water the TEC will be exposed to during operation. However, the sealing rings prevent large particle ingress, and the relatively slow intermittent in situ operation allows water to settle out of the barrier fluid. Given this, the use of potable water was deemed an acceptable compromise.

The MSS is a Syntron RP mechanical shaft seal produced by Garlock (Garlock Sealing Technologies 2010). The rotating components, shown in dark gray in Figure 2, interface with the shaft using the nitrile butadiene rubber (NBR) drive rings. These are compressed by bronze seal face rings and are kept in compression by four tension springs evenly spaced radially around the seal. The rotating components press against front and rear graphite antifriction rings. The entire assembly is held inside a brass housing. The seal face and antifriction rings are constructed from bronze and graphite, respectively. The rubber driving rings are molded from NBR. The entire housing is filled with Synturion 6 BF (ExxonMobil, Irving, Texas, USA) (ExxonMobil 2021). BF fills the seal as well as the BFUR and BFLR. Note, the MSS, produced by Garlock, is called Syntron and it is filled with a Synturion 6 BF produced by ExxonMobil.

The main sealing surfaces are at the antifriction ring/seal face ring interface, and the shaft/drive ring interface. If water leaks past the front antifriction ring interface without leaking through the rear antifriction ring, it mixes with the BF and pressurizes the housing. This water may remain in suspension or fall into the BFLR. The volume of this water can be measured either in the graduated lower reservoir or as a concentration in the oil samples. Leaking through the front and rear antifriction rings would be observed at the back of the housing and collected into a lower drip pan but would not pressurize the housing. Leaking between the shaft and drive rings would have similar outcomes.

2.1.1 Instrumentation

Pressure

The water tank and BF pressures are monitored individually throughout the experiment by Omega PX309-050A10V pressure transducers (Omega Engineering, Norwalk, Connecticut, USA). The pressure range for these sensors is between 0 and 344.7 kPa (0 psi to 50 psi), an accuracy of ±0.25%, and an operating temperature range of -40°C to 85°C (-40°F to 185°F). These sensors are connected to an etherCAT network via data acquisition to a computer, which records the pressure of each fluid every 5 minutes throughout the experiment.

Temperature

Temperature in the BF is recorded by a 3-wire Pt100 resistance temperature detector (RTD) throughout the experiment (McMaster-Carr number 6568T47, Robbinsville, New Jersey, USA). The RTD passes through the BF hoses and into the housing so the temperature of the seal can be directly measured. The temperature range for the RTD is from -20°C to 176°C (-4°F to 350°F) with an accuracy of ±0.12%, which is acceptable for this application where the temperature is expected to remain below 100°C (212°F). The RTD was connected to a data acquisition module and recorded at the same interval as pressure.

Lower Reservoir Leakage

The lower reservoir is a cylinder with an internal diameter of 38.1 millimeters (mm;1.5 inches [in.]) and an axial scale measures water leakage volume. Water collected during oil sampling is also measured with a graduated cylinder. The BF is initially under atmospheric pressure. The volume of water collected in the BFLR was recorded every 5 days.

Motor Current

The electric motor controller, which regulates the main shaft's rotational velocity, also monitors the current draw during operation. Any power spikes or dips larger than 1 ampere are recorded along with a time stamp. Due to the limitations of the controller, this was the smallest possible interval. Initially the motor current was highly variable and was recorded automatically by the system; however, the current stabilized after a period of approximately 5 days, at which point the current was recorded manually every 5 days along with leakage and rubber hardness, discussed below.

2.2 NBR Accelerated Aging

A second set of NBR drive rings were stretched around a 6-inch-diameter tube to simulate the main shaft seal. This was then submerged in barrier fluid and placed in a 90°C (194°F) oven. An RTD recorded temperature continuously for the duration of the test. The drive ring assembly was removed from the BF and allowed to cool to room temperature for Shore A hardness testing every 5 days. The hardness was measured with a Mitutoyo Hardmatic Type A tester, HH-331, (Mitutoyo America, Aurora, Illinois, USA). The drive rings remained saturated in BF while out of the oven, and the room temperature was consistently 21°C –22°C (21°F – 71.6°F).

3 Results and Discussion

3.1 Main Shaft Seal Wear

The main shaft seal was tested continuously for 137 days while recording the water tank pressure, BF pressure, and seal temperature. These results are shown in Figure 3. The test stand operated in cycles of 59.5 minutes on at 160 rpm and 0.5 minutes off. During this time, it recorded a total of 3,312 cycles and 31.5 million revolutions. In terms of revolutions, this is approximately 40% of the expected 78.8 million revolutions during the 5-year SI. These cycles occurred over 137 days.

The water tank was maintained at a constant pressure of 199.9 kPa (29 psi) with a regulator. The BF began at 11.4 psi, which is atmospheric pressure in the facility in which the testing was conducted. The initial drop in BF pressure after 5 days was due to the oil sampling discussed in Section 2.1. After taking the first oil sample, the BF pressure decreased to 10.2 ± 0.3 psi and remained there for more than 100 days. An attempt was made to collect another oil sample around day 50, but it was unsuccessful. Air was heard drawing into the valve, and only a few milliliters of oil were collected. This sample did not show any measurable water volume. For the remainder of testing, the system was left to operate while pressure and temperature were monitored continuously. During this time, pressure remained stable and no leakage was observed in the BFLR or through the back of the seal, indicating the seal was successful in preventing water ingress.

The test system was shut down due to a planned power outage for maintenance of the test facility and remained down for 5 days. Upon restarting the system, pressure had equalized across the main shaft seal, and significant water leakage was observed in the BFLR. After an oil sample was taken, it became clear the seal could no longer prevent water leakage at the front antifriction ring/seal-facing ring interface. This change in behavior could not be accounted for, so the testing was stopped.

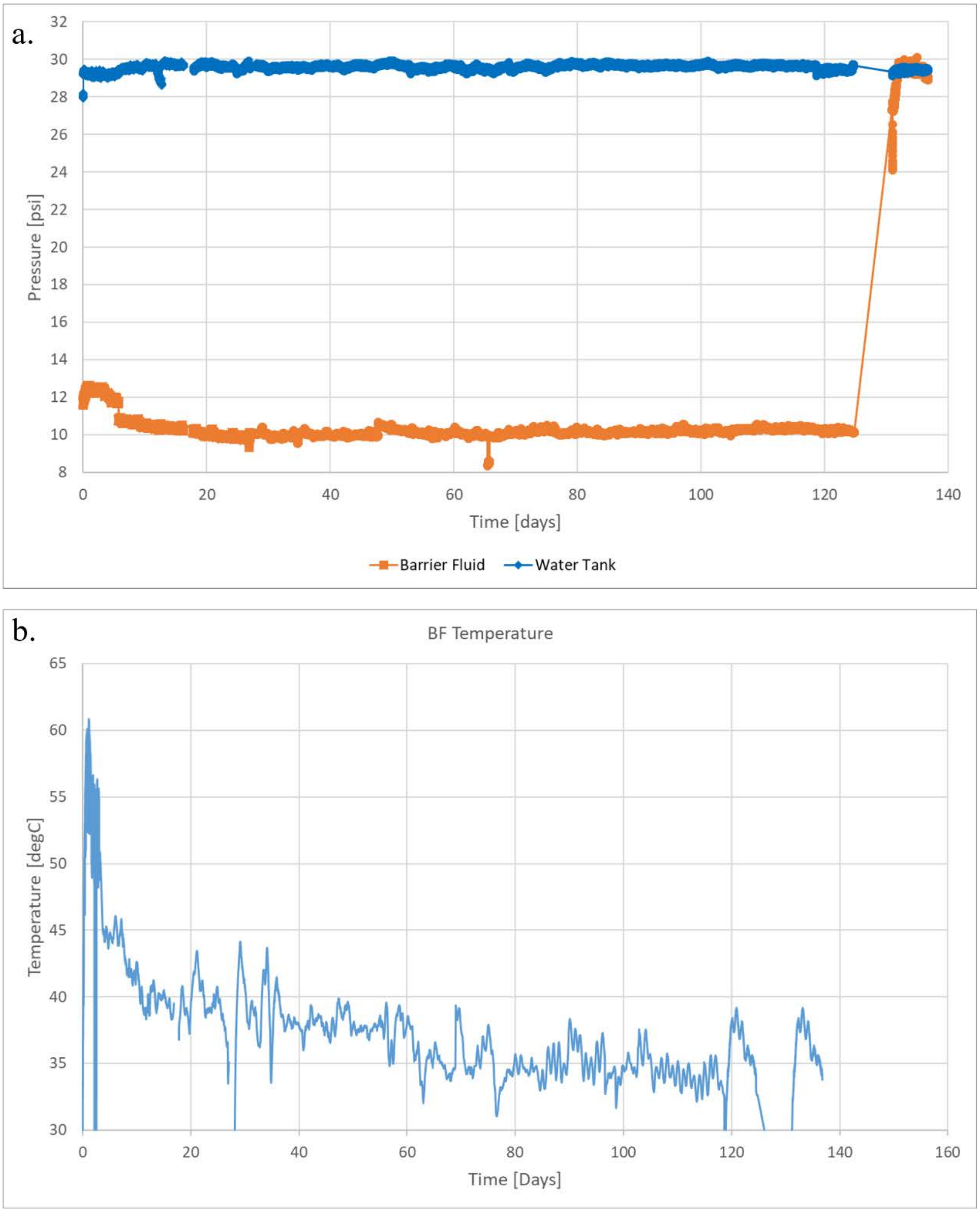

Figure 3. (a) Water tank and BF pressure during operation. (b) Temperature of the BF during operation. The BF temperature sensor was placed inside the housing to best measure the operating temperature of the seal.

Figure 3a shows the BF pressure increases marginally during the first few days of testing. During this time the BF temperature varied from a peak of 61°C (141.8°F) to a low of 30°C (95°F). The average temperature range for most of the testing was 35°C to 40°C (35°F - 104°F). The highest temperatures were recorded during the first 10 days. This period also corresponds to the largest motor current draw, shown in Figure 4, and when the most water leakage was recorded in the initial oil sample, 26 mm. It is likely this initial period corresponds to a brief wear in time where asperities between the antifriction rings and seal-facing rings are eroded away and smooth sealing surfaces form. After this period temperature stabilized and was significantly influenced by diurnal rhythms within the testing facility. Additionally, motor current draw reduced and stabilized.

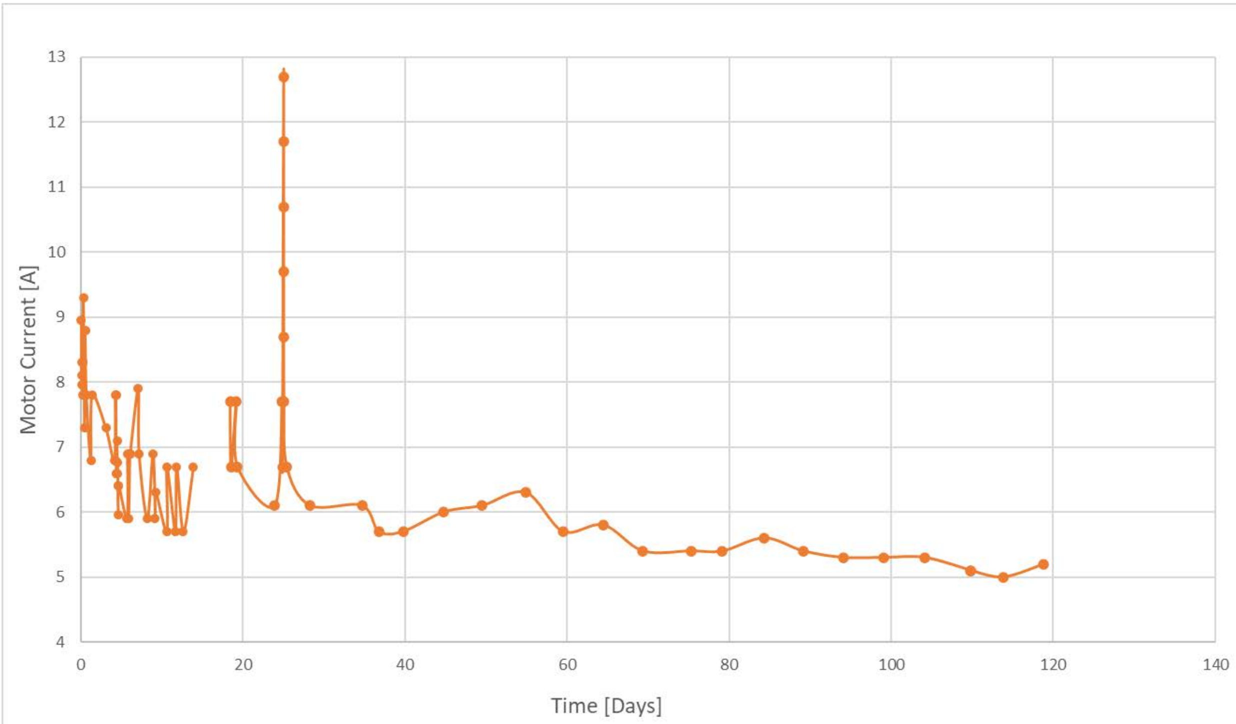

Figure 4. Motor current draw of test stand during operation. This measured the current required to maintain a constant 160 rpm.

The gap in the data at 18 days and the spike at 25 days were due to a circuit breaker failure and reactivation. The motor current draw was relatively large and chaotic initially during the theorized wear-in period before settling into a more stable, slowly decreasing trend for most of the testing. This is consistent with a brief wear-in period while the asperities are worn away between the mating surfaces. It also suggests that the BF and carbon particulate form an effective lubricating medium that produces more favorable interface conditions during operation.

At the end of testing at 137 days, an unexplainable step change in BF pressure was recorded. It was decided that any future results would not be meaningful, so testing was stopped for the final data collection. When draining the water and BF tanks, approximately 1.1 L (0.24 gal) of water was found in the BF, and an equal amount of BF was found in the water tank. Only minimal fluid leakage was recorded through the back of the seal into the atmosphere. The collected fluids are shown in Figure 5. Considering the seal maintains a pressure gradient between the water tank and the atmosphere, it was hypothesized a small pressure gradient would always exist across the seal. The observed fluid exchange from the end-of-test malfunction proved this was not the case.

Figure 5. Images showing the fluid removed from the water tank: (a) side view; (b) top view. The BF floats on top of the water and is stained black from suspended graphite particles worn away from the antifriction rings.

Images by Miles Skinner, NREL

Disassembly and Wear

The main shaft seal was disassembled at the conclusion of testing to inspect the components and measure the amount of material removed from the wearing surfaces. The mass of material removed from each ring is given in Table 1. The front, water-facing side of the seal suffered more wear than the back side. Additionally, the carbon antifriction rings suffered more wear than any of the other components, as expected. The carbon rings have lower abrasion resistance than the bronze seal-facing rings, so they wear fastest. As anticipated, the rubber drive rings did not experience any wear because they should not slip against the shaft or the spring compression ring.

Table 1. Mass Lost From Each Component That Experiences Wear During Operation

Component		Original Mass [g]	Final Mass [g]	Mass Loss [g]
Graphite Antifriction Ring	Front	204.5	202.0	2.5
	Rear	205.0	204.5	0.5
NBR Drive Ring	Front	80.0	79.5	0.5
	Rear	80.0	79.5	0.5
Bronze Seal-Facing Ring	Front	350.0	349.0	1.0
	Rear	349.5	349.0	0.5

In addition to experiencing asymmetric wear between the front and back antifriction rings, the individual rings were also worn radially asymmetric. The carbon ring wore on a lip at the interface between the antifriction ring and the bronze seal-facing ring. The original and final height of this ring are given in Table 2. The front ring experienced more wear than the rear ring. The height difference between the highest and lowest point was 0.6 mm. This creates a plane angled 0.2 degrees relative to the back of the ring that does not experience any wear. The rear antifriction ring had significantly less wear with a maximum height difference of 0.21 mm. After testing, the wear surface of the front ring was almost completely worn away at the lowest point (see Appendix).

Table 2. Change in Height of the Wearing Surface on the Graphite Antifriction Ring

Component		Original Height [mm]	Minimum Height [mm]	Maximum Height [mm]
Graphite Antifriction Ring	Front	1.167	0.27	0.87
	Rear	1.167	0.88	1.11

Failure

The system showed no signs of failure prior to the shutdown at 137 days, which is typical for shaft seals of this type. While minor leakage may occur during operation, the combination of graphite wear surface and BF is expected to be extremely robust (Jiang et al. 2019; Lee et al. 2012; Klein et al. 2010). This system was exposed to potable water with minimal contamination from any particulate debris. There were also no external loads applied to the main shaft such as pitch or yaw moments, which could be expected in deployed tidal turbine systems. Finally, the test procedure had this seal operating on a 99.2% duty cycle at 160 rpm as opposed to a more realistic 75% duty cycle at 40 rpm. It is possible that more realistic test conditions would result in more significant leakage. Mimicking tidal conditions, as was done in the 2015 test, to include dissolved minerals or suspended particles in the water tank might result in more leakage.

The axial asymmetric wear observed on the graphite antifriction rings is a concern. It is not entirely clear why the front ring experienced wear more drastically than the rear ring. The compression ring should apply equal force on the front and rear ring, so compressive forces should be equalized front to rear.

Considering only the front graphite ring, the circumferentially asymmetric wear is more difficult to understand. The most likely explanation is an assembly error or disruption to the test stand setup machine during the power outage. The MSS was assembled according to instructions provided by the manufacturer and functioned continuously for 137 days, suggesting the seal was assembled correctly; however, the wear pattern suggests nonuniform loading from the compression rings. The compression rings use a series of springs and alignment pins to create constant outward pressure in the wear surfaces and increase sealing potential. Cocking of the springs or pins in the housing, combined with the more hazardous environment at the front of the seal housing, could create the necessary conditions for the observed wear pattern. Aside from the observed results, there is little evidence to support this theory, so further evaluation of the seal components is warranted.

The results suggest the MSS may not have completed the entire SI before failure. The front graphite ring was worn through 77% of the interfacial lip on one side, suggesting failure was likely to occur before completing the SI, but this is difficult to determine because the wear rate is uncertain. Based on the wear-in period shown in Figures 3 and 4, the wear rate is not constant, but its actual value cannot be determined without more information. The objective of BF sampling was to determine the graphite particle generation rate, but this was unsuccessful because BF pressure prevented sampling.

3.1.1 NBR Accelerated Aging

The data from the accelerated rubber aging tests is shown in Figure 6. The rubber drive rings began with a hardness of approximately 65 Shore A. The recorded hardness increased asymptotically up to 90 Shore A in 70 days. This is consistent with previously reported results (Jiang et al. 2019). Significant hardness variation was observed during testing and is largely attributed to variations in temperature. Hardness testing required handling, so samples were allowed to cool from 90°C to room temperature. However, the ambient temperature varied significantly from 16°C to 23°C (60°F-73.4°F).

After wear testing completed, the drive ring hardness for the in-situ rings was measured as 76.8 and 77.4 Shore A for the front and rear rings respectively. The wear testing was conducted for approximately 130 days at 35°C–40°C (95°F-104°F), depending on ambient temperature and heat generated from friction. This suggests the seal hardness is highly dependent on ambient temperature. Considering that wear testing simulated 40% of the expected SI, the drive rings are unlikely to reach full hardness before the SI completes.

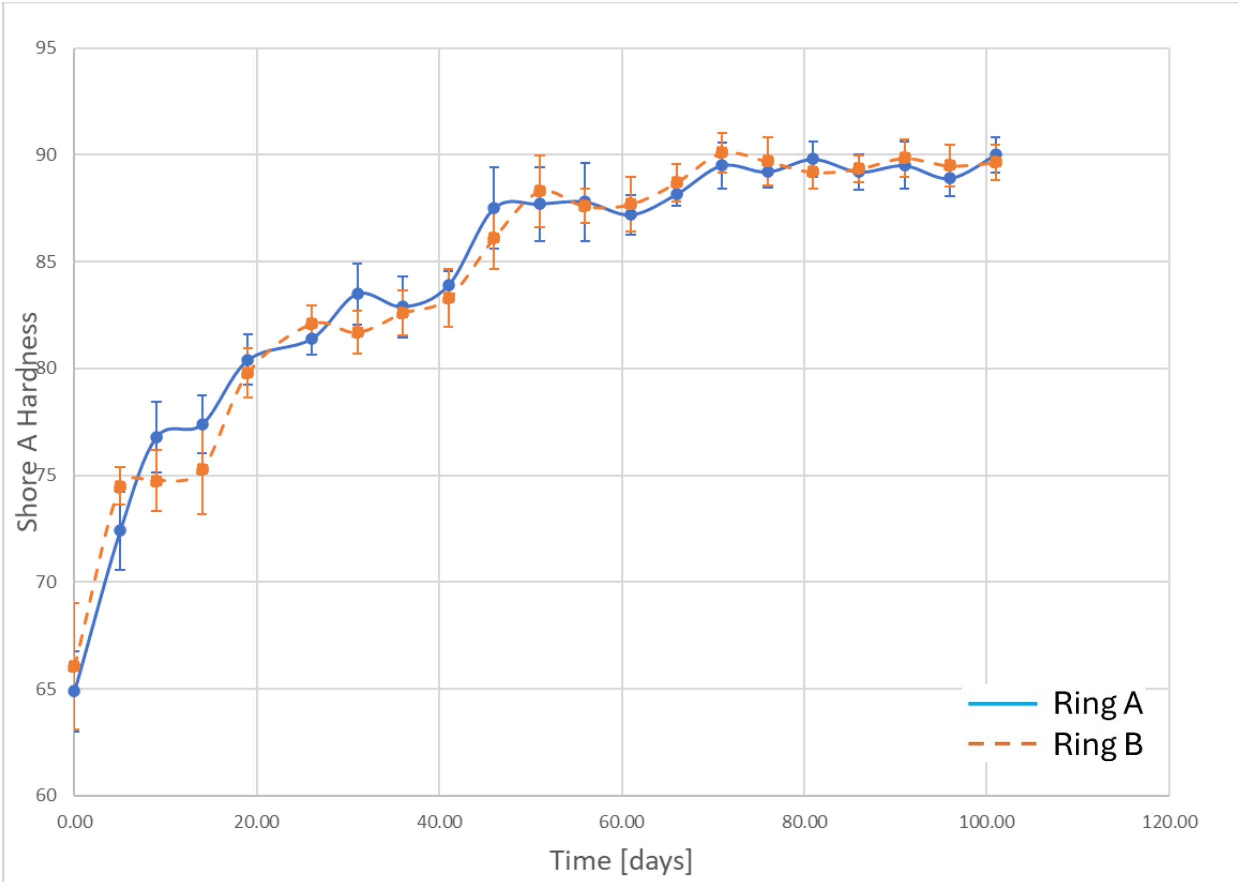

Figure 6. Shore A hardness of the NBR drive rings during accelerated aging testing. The rings were allowed to cool to room temperature before measurements were taken. Note, the orange and blue lines identify the two NBR drive rings tested.

4 Review and Recommendations

In this study, the MSS test rig provided by Verdant Power was modified to include BF sampling and pressure measurements of the water tank and BF. Additionally, the motor current and water leakage volume were monitored periodically during testing. This testing apparatus met the requirements for shaft seal testing outlined in ISO 6194. The objective for accelerated lifetime testing was to determine the MSS ability to function continuously for the 5-year SI and what factors could be used for health monitoring of the seal. An accelerated aging test was also conducted on additional rubber drive rings to measure the change in rubber hardness over time.

Following the test, the findings were reviewed with Verdant Power, Dovetail Solutions LLC, and Garlock Sealing Technologies for consensus evaluation of results. A review of the test articles (seals) tells a story of misalignment during testing. Misalignment is seen both overall (shaft moving as a whole) and from front to back (more movement on the water side than the air side).

- The through-hole of the bronze components is 154.0 mm (6.0625 in), and by adding the wear band radial distance 2x, Garlock determined the diameter of the area the rotary faces had contacted the stationary faces during operation.
- On the air side, the rotary was contacting the seal facing ring in a 168.0 mm (6.615 in) diameter circle, and on the oil side the seals were contacting in a 168.7 mm (6.645 in) diameter circle. Both patches were of even width all the way around the face, so it is expected the movement was the axis of rotation "wobbling" rather than the axis of the shaft being pulled to one side but not moving.
- Garlock examined the outer diameter of the raised face of the rotary, as it was machined to 165.9 mm (6.531 in). In a theoretically perfectly aligned shaft to seal, with perfect shaft runout, the contact patch would have a matching diameter of 165.9 mm (6.531 in). This means there is evidence of 2.1 mm (0.084 in) diametric wobble on the air side and 2.9 mm (0.115 in) on the oil side.
- Garlock indicates that minor misalignment of up to 0.38 mm (0.015 in) is usually absorbable by these seals, but generally the shaft stays perpendicular to the mounting surface. It is unusual to see one size have 0.76 mm (0.030 in) more movement than the other.

4.1 Wear

Initially, the MSS showed a brief wear-in period where temperature, pressure, and motor current demands were highly variable. Based on the volume of material missing from each ring, the most significant wear occurred on the graphite antifriction rings at the interface with the bronze seal rings. It was during this period that the largest graphite particles were produced, and the rubber drive rings experienced the most significant change in hardness. This phenomenon was not seen in the 2015 testing. After the wear-in period, the seal behavior became much more predictable with lower temperatures, constant pressure, and stable motor current demands. This continued for most of the testing. The seal completed 31.5 million revolutions, or 40% of the 78.8 million revolutions expected in the SI prior to the NREL lab power failure and likely disruption of the test stand.

The seal showed significant asymmetric wear between the front and rear wearing surfaces as well as circumferentially around each ring. The exact cause of this asymmetry is not known, but possible causes are an assembly issue or the suspected misalignment. Documentation should be created to detail the proper assembly of the seal and, if possible, ways to inspect the assembly during operation to ensure consistent performance.

Given the examination by Garlock and the likely misalignment of the test rig, the only conclusion on long-term wear of the seals is that they were performing as expected up to the test rig disruption, an equivalent of 2 years (40% of the 5-year life cycle).

4.2 Leakage

An interesting result is the total lack of water leakage during this steady-state interval. BF pressure remained approximately 1 psi below atmospheric pressure, and water leakage could not be seen in the BFLR. It is also possible the potable water used in the water tank did not provide an appropriately hazardous environment and therefore artificially reduced wear and leakage rates. Based on the expected deployments of these tidal turbines, further research is recommended in seawater, brackish water, and water with suspended solids such as sand.

The MSS was able to maintain a 137.9 kPa (20 psi) pressure differential between the water tank and atmosphere, which was an unexpected result that did significantly impact the project's ability to determine wear rate based on particle generation measured from BF samples. A pressure relief valve could have alleviated this issue but was not included because BF pressure was used to monitor seal health—i.e., water leakage into the BF would change the BF pressure, so a valve would have negatively impacted the leakage monitoring objective.

The significant leakage and ultimate failure of the system is not entirely clear; however, it is likely due to an external perturbance of the system, either caused by the power outage or some other disturbance, such as an accidental collision with another piece of equipment. Therefore, the seal failure cannot be taken as conclusive evidence of this seal's inability to meet performance requirements.

4.3 Protocol

The current test ended after the system was stopped for facility maintenance and suffered a catastrophic malfunction at that time. This behavior is unusual for the type of seal used in the test stand, and it is surmised that assembly, movement, or other disturbance of the test stand during the outage was responsible, which was verified by the Garlock seal article inspection.

The project employed a long duty cycle of 99.2%, but the much shorter duty cycle seen in situ may also contribute to higher leakage rates. This should be explored further in future studies.

4.4 Recommendations

The team reached out to the MSS original equipment manufacturer, Garlock, for recommendations to prevent this kind of failure in the future. The seal was sent to Garlock, and their recommendations are as follows:

- Garlock recommends the next steps for root cause determination should include examining the test rig itself to see how the shaft is moving relative to the surface the seal is mounted to and compare that to the worst case expected in the real world.

- Prior to the outage, the results indicate that the seal successfully completed 40% of the desired 5-year service interval. Follow-up testing should be conducted through NREL's Testing Expertise and Access for Marine Energy Research (TEAMER) program to reinitiate the accelerated testing by:
    - Improving the setup protocol and conditions
    - Involving Verdant /Garlock in assembly at setup.

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Appendix A. Additional Images

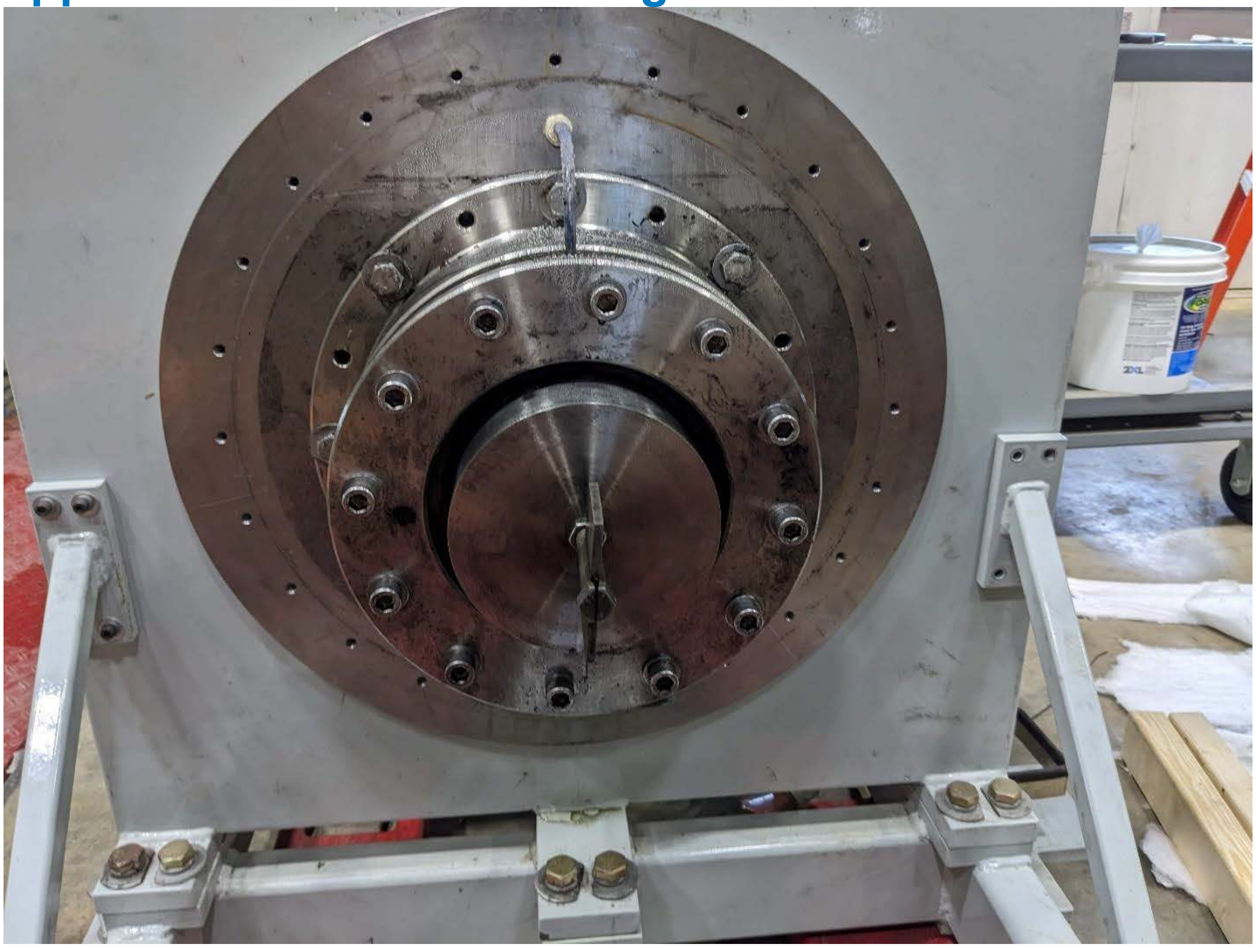

Figure A-1. Front of the assembled MSS with water tank removed.

Image by Miles Skinner, NREL

Figure A-2. Front of the MSS housing with water tank removed. A black paste can be seen extruding from the seal. The paste is a mixture of graphite and BF.

Image by Miles Skinner, NREL

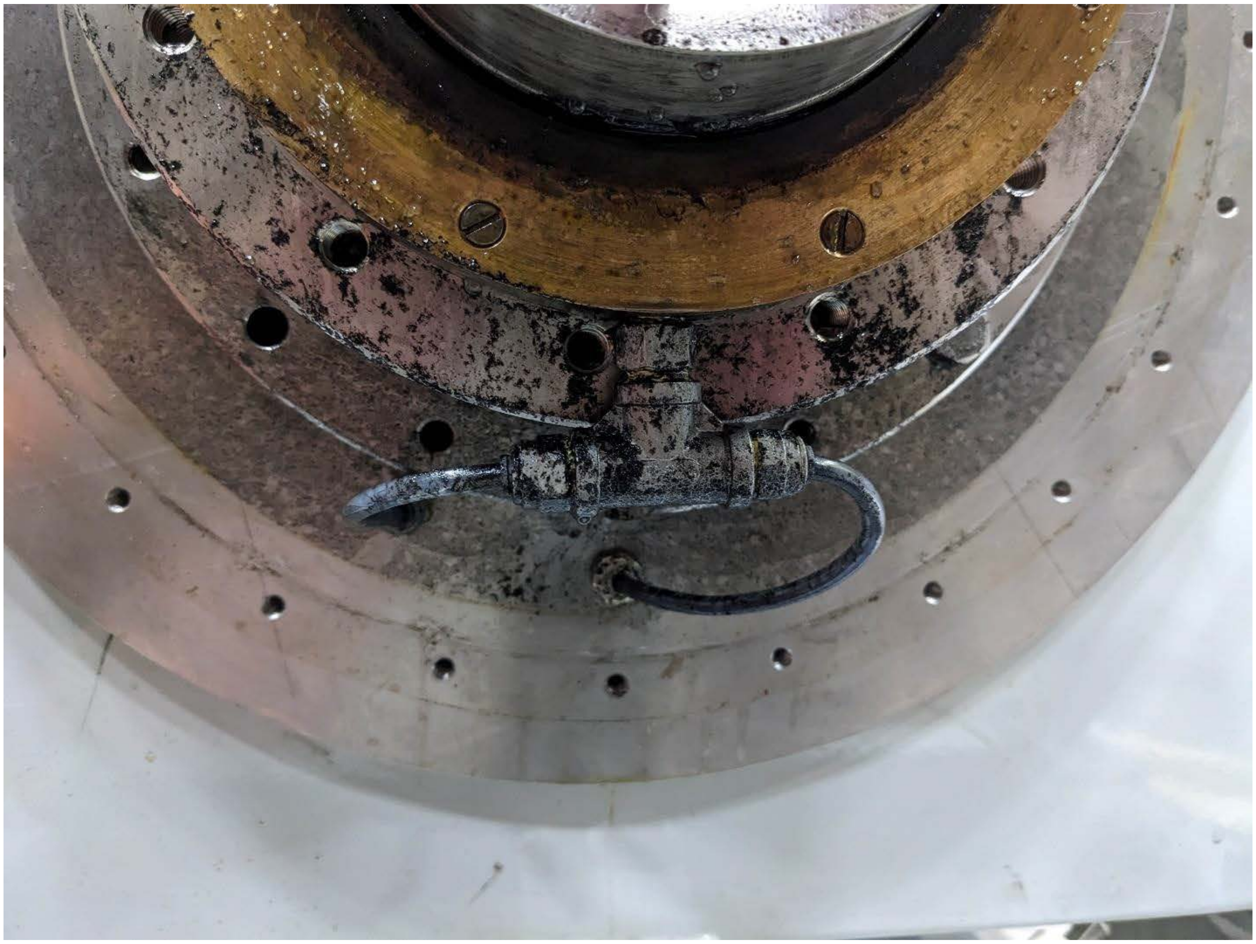

Figure A-3. Bottom side of the MSS showing the connection from the seal to the oil sample valve and BFLR.

Image by Miles Skinner, NREL

Figure A-4. MSS and housing removed from the shaft. The inside of the seal shows accumulation of graphite/BF paste. More accumulation can be seen at the front (top) of the seal than the back (bottom).

Image by Miles Skinner, NREL

Figure A-5. Wear surface of the front graphite antifriction ring with accumulation of graphite/BF residue. The visible surface interfaces with the bronze seal face ring.

Image by Miles Skinner, NREL

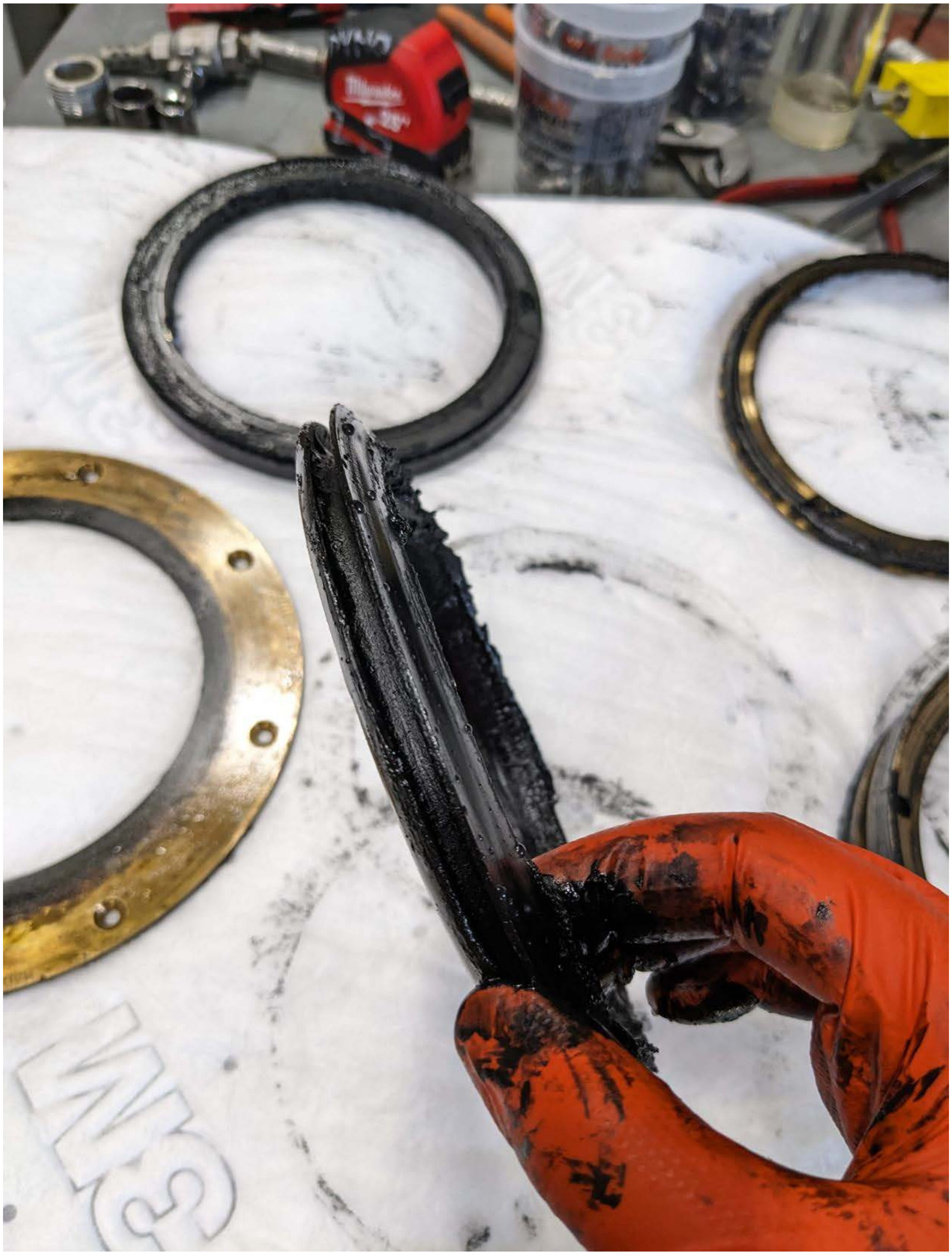

Figure A-6. Front drive ring showing permanent deformation caused by constant pressure from the seal face and compression rings.

Image by Miles Skinner, NREL