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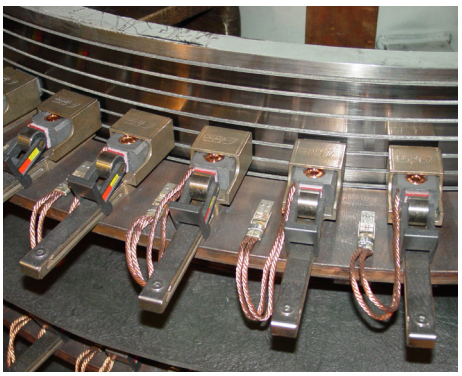
Spring Fatigue Test for Hydro Generator Applications

By David Duch, Engineering Manager

Abstract

For hydro generators, the industry had a rule of thumb to keep the total runout on the slip rings below 0.010 in. (0.25 mm) for good brush and spring life. Staying below 0.010 in. (0.25 mm) runout has been difficult, so we were challenged by our customers in the hydro industry to determine the actual requirement for total runout. The fatigue life for a constant force spring in a brush holder is very difficult to determine analytically. Therefore, we conducted a spring fatigue cycle test that simulates the slip ring on a hydro generator with several different amounts of runout. We also evaluated several different spring options including spring force, coil diameter, brush size, number of coils in the spring, and the brush top surface design.



Objectives

The main objective was to understand the fatigue life of the springs running on slip rings with different total runouts. The study also focused on the effects of other significant variables on spring fatigue life like alignment and number of coils in the spring.

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The testing needed to simulate as closely as possible the operating conditions seen on hydro applications, while also being practical enough to allow the testing to be run easily in a laboratory. The end goal is to educate the hydro customer base with accurate and feasible recommendations for the desired life. As part of the testing and learning, we also studied improvements to the holder, spring, and brush designs to improve the life performance at extreme operating conditions.

Test Setup

The test stand consists of a motor and gear reduction drive connected to six rings. The motor and drive were sized appropriately to run continuously for several years.

The six rings are made from brass with offset centers to create six different levels of total indicated runout (TIR). The runout levels that were tested were 0.015 in. (0.38 mm), 0.030 in. (0.76 mm), 0.060 in. (1.52 mm), 0.075 in. (1.91 mm), 0.090 in. (2.29 mm), 0.125 in. (3.18 mm).

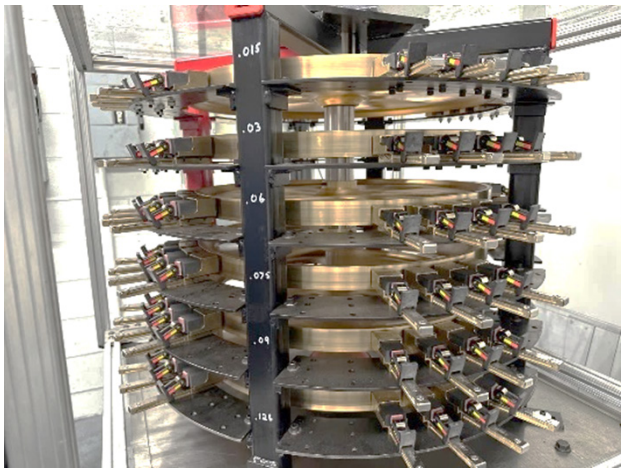

Figure 1: Test Stand

The ring diameter is 24 in. (610 mm) and running continuously at 180 rpm which equates to a surface speed of 1130 ft/min (5.74 m/s). Because the main purpose of the testing is to determine the spring fatigue life, power is not being transferred through the brushes. The wear rate is purely the mechanical portion of brush wear. See Figure 1 for a picture of the test stand. During the test, the runtime, cycle count, spring force, and brush wear were measured or checked periodically.

The options that were tested are as follows:

- Coil diameter: Standard and Larger - 6 lbs. (2.7 kg) (Coil diameter - 0.75 in. (19.1 mm)), 3.75 lbs. (1.7 kg) (Coil diameter - 0.625 in. (15.9 mm)), 3.75 lbs. (1.7 kg) (Coil diameter - 0.75 in. (19.1 mm))
- Red Top Pad: Slot for the spring (3/8 in. (9.5 mm) radius) or No slot (Flat). See Figure 2 and 3
- Spring Force (Pressure): 6 lbs. (2.7 kg) = 4 psi (0.281 kg/cm^2) and 3.75 lbs. (1.7 kg) = 4 psi (0.281 kg/cm^2)
- Brush Sizes: 1.5 in. (38.1 mm) x1.0 in. (25.4 mm) and 1.25 in. (31.8 mm) x 0.75 in. (19.1 mm)
- Coil count: Single and Double. See Figure 4

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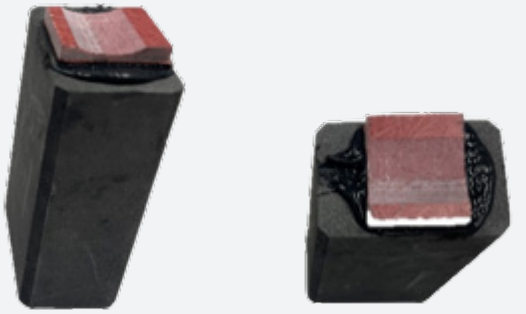
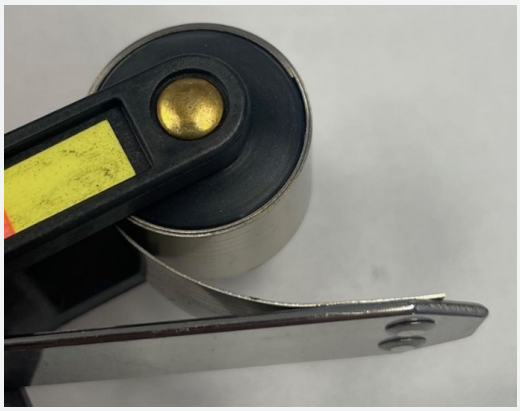
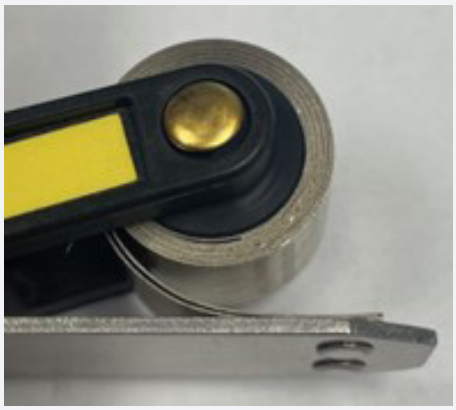


Figure 2: 3/8 in. (9.5 mm) Radius Red Top Pad

Figure 3: Flat Red Top Pad



Single Coil Spring



Double Coil Spring

Figure 4: Single vs Double Coil Spring

Results

Assembly Fixture Alignment

At the beginning of the testing, we saw four low hour failures (3 on the 0.125 in. (3.18 mm) TIR ring and 1 on the 0.090 in. (2.29 mm) TIR ring). After investigating the issue, we found that misalignment of the coil spring on the back plate during assembly was the cause of the early hour failures. Based on this discovery, we created an improved assembly fixture that allowed for better alignment between the coil spring and the back plate before riveting the coil spring to the back plate. The new assembly fixture was used for the rest of the testing and production. The improved fixture extends the constant force spring farther which allows more precision in the angle of the spring to the back plate. The improved fixture also allowed the process to be fine-tuned and less reliant on the skill of the operator. Further testing has shown that the better alignment allowed the springs to become more forgiving on the higher TIR rings. See Figures 5, 6, and 7.

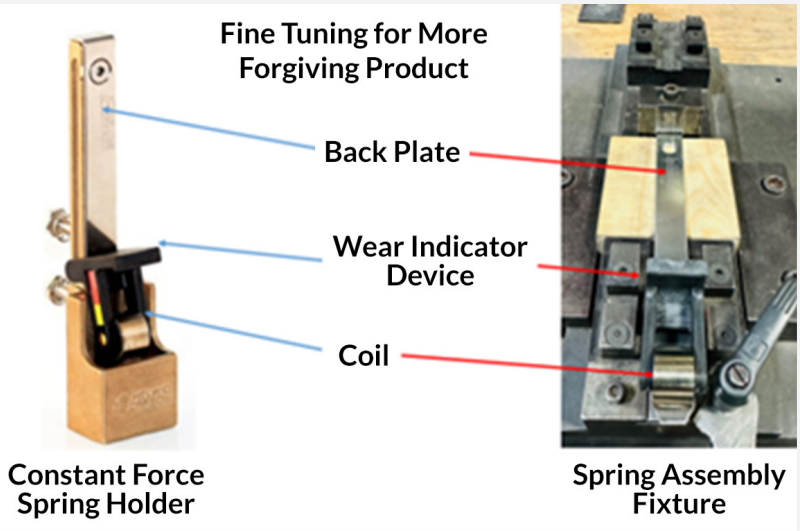


Figure 5: Holder and Fixture Used to Attach the Spring to the Back Plate

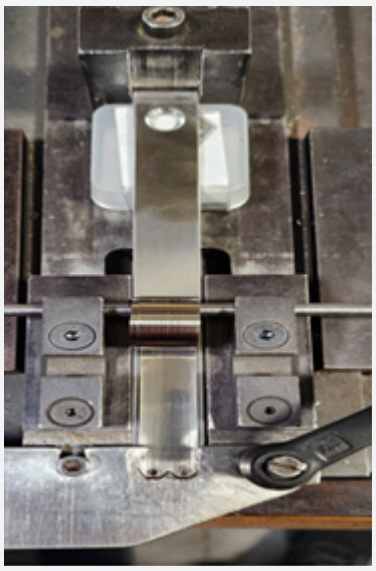


Figure 6: Original Assembly Fixture

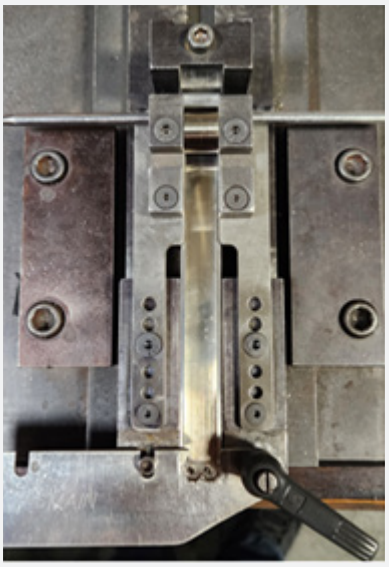


Figure 7: Improved Assembly Fixture

Single and Double Coil Springs

The double coil springs had fatigue failures at around the same number of cycles as the single coil springs. Also, when one of the layers of the double springs failed, the coil got caught on the brush and prevented the brush from moving freely. The pattern of the spring failure shows shear and step cracks across the coil. As expected, the failure point seems to be at the highest stress point or close to max amplitude. See Figure 8.

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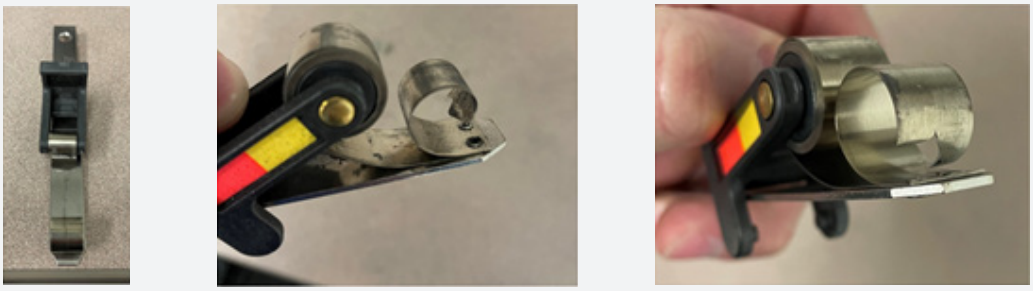

Figure 8: Typical Spring Failure

Spring Life

The testing has run for over 185 million cycles at 180 rpm which is 1.95 years of continous running or equivalent to 3.5 years continous running at 100 rpm. A total of 220 springs have been tested or are currently on test. 27 of those springs have failed. Testing with three diferent versions of the brushes, spring force, and spring diameters shows no failures at 0.030 in. (0.76 mm) and 0.015 in. (0.38 mm) runout. See Figure 9. Figure 9 shows the number of cycles vs the TIR of the rings. The quantity to the right of the blue bar shows the quantity of springs still running and the orange bar shows quantity of failures, range of failure cycles, and average failure cycles. As can be seen in the graph, there were a number of failures at all the TIRs at and above 0.060 in. (1.52 mm). The data also shows that the fatigue life of the springs has a high amount of variablity.

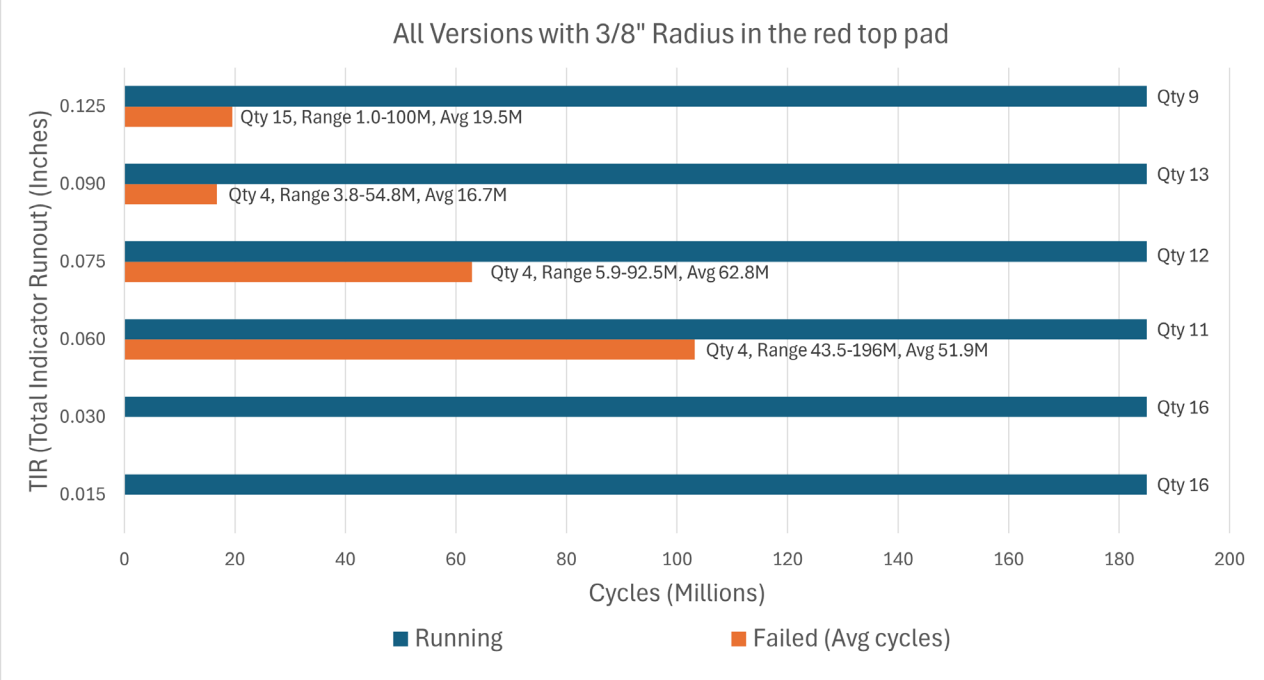

Figure 9: Spring Life (All Versions)

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Figures 10, 11, and 12 show the individual data for three versions that were tested. The fatigue life of the 3.75 lb. (1.7 kg) springs seems to be slightly better than the 6 lb. (2.7 kg) springs but both springs had failures on rings with a runout of 0.060 in. (1.52 mm) and above.

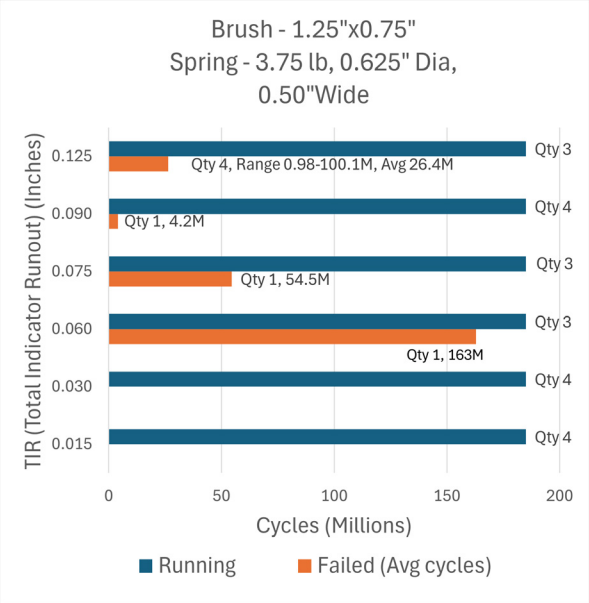

Figure 10: Spring life 3.75 lb., 0.625 in. Dia

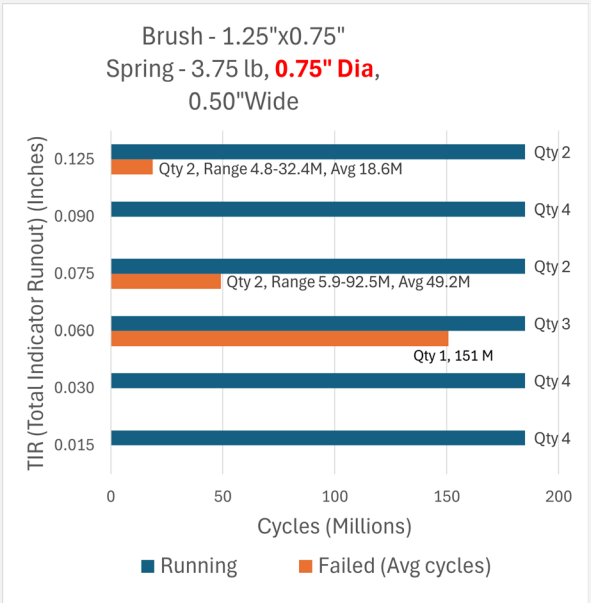

Figure 11: Spring life 3.75 lb., 0.75 in. Dia

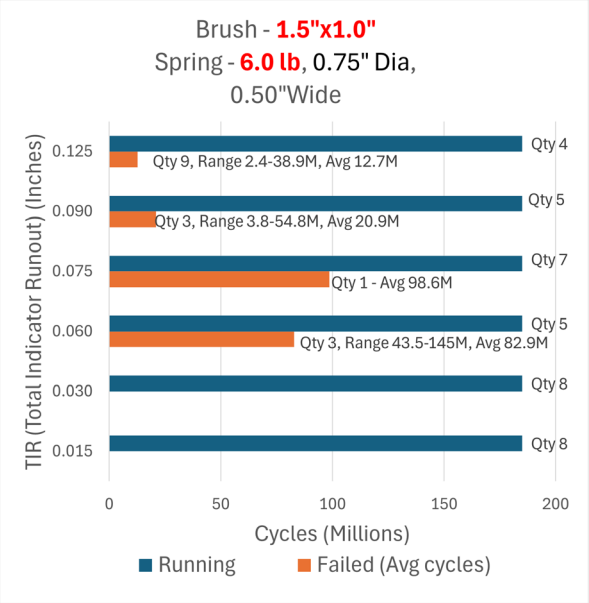

Figure 12: Spring life 6.0 lb., 0.75 in. Dia

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Figures 13 and 14 show the spring fatigue life comparison between the 3/8 in. (9.5 mm) radius in the Red Top Pad vs a flat Red Top Pad. The flat Red Top is showing better fatigue life compared to the 3/8 in. (9.5 mm) radius on the rings with TIRs of 0.060 in. (1.52 mm) and 0.075 in. (1.91 mm). The 3/8 in. (9.5 mm) radius is holding the spring in a specific location and the flat pad allows the spring to move to its natural position. Because the 3/8 in. (9.5 mm) radius is restricting the motion of the spring, the spring is likely seeing higher nominal stress which would reduce the fatigue life.

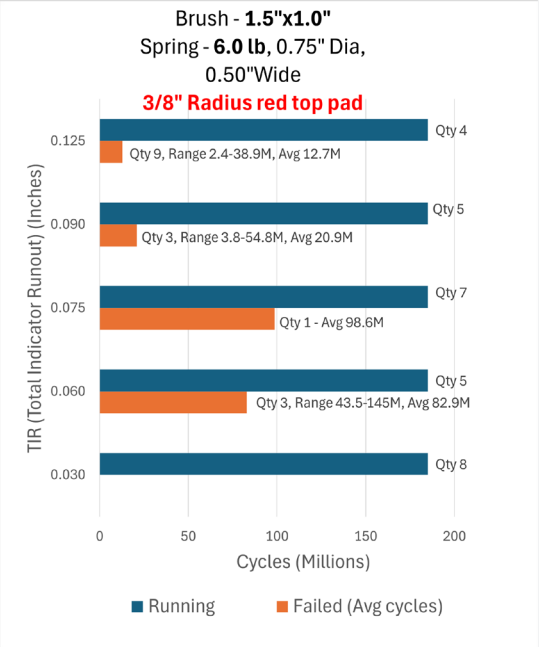

Figure 13: 3/8 in. Radius Pad Spring Life

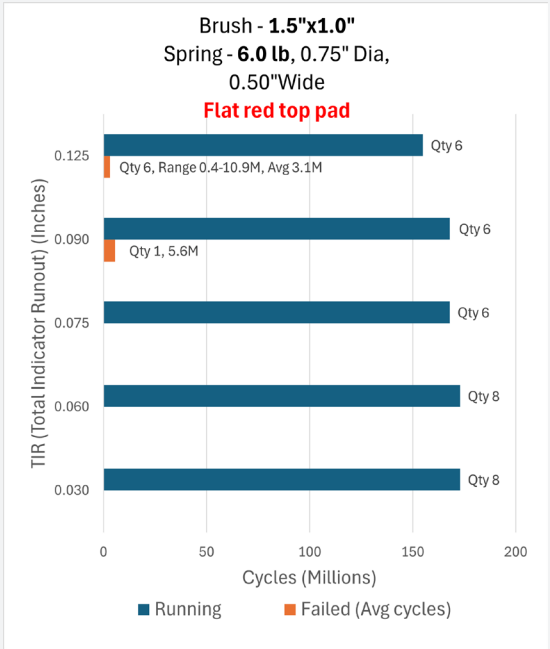

Figure 14: Flat Top Pad Spring Life

Brush Wear

As stated previously, the brush wear is mechanical wear only because current is not running through the brushes during the testing. The results show relatively high wear initially and then as the patina forms on the rings, the wear rate starts to slow. The brushes running on the 0.125 in. (3.18 mm) runout ring have a higher wear rate after the initial wear rate compared to all the other rings. See Figure 15 A, B, and C.

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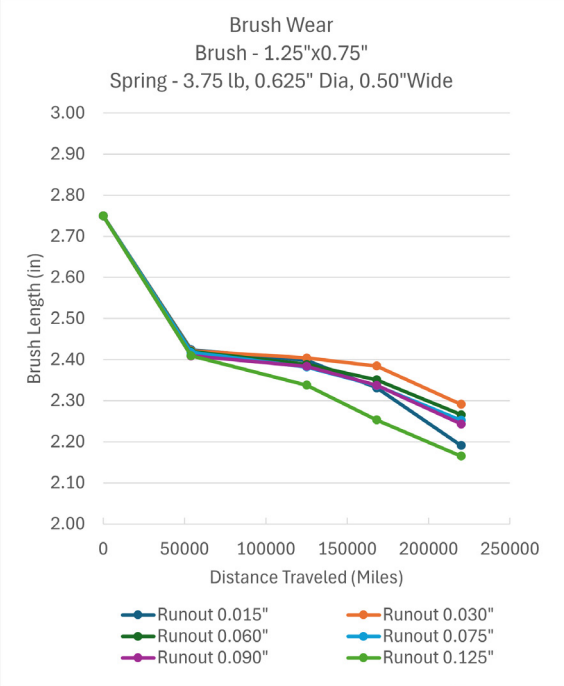

Figure 15A: Brush Wear 3.75 lb., 0.625 in. Dia.

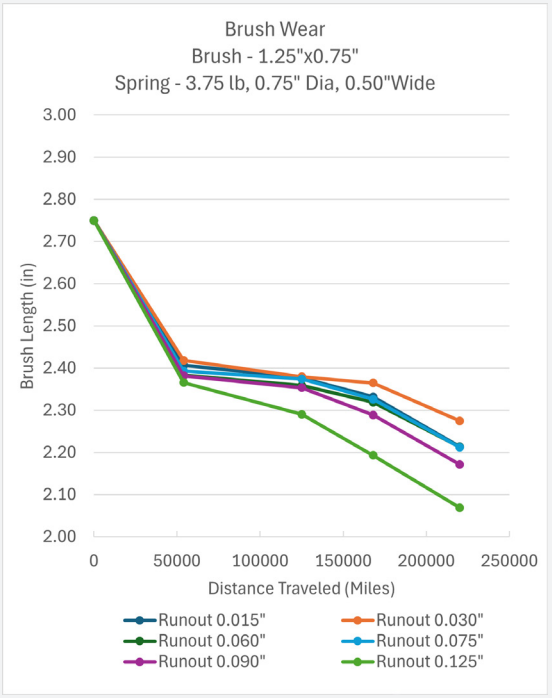

Figure 15B: Brush Wear 3.75 lb., 0.75 in. Dia

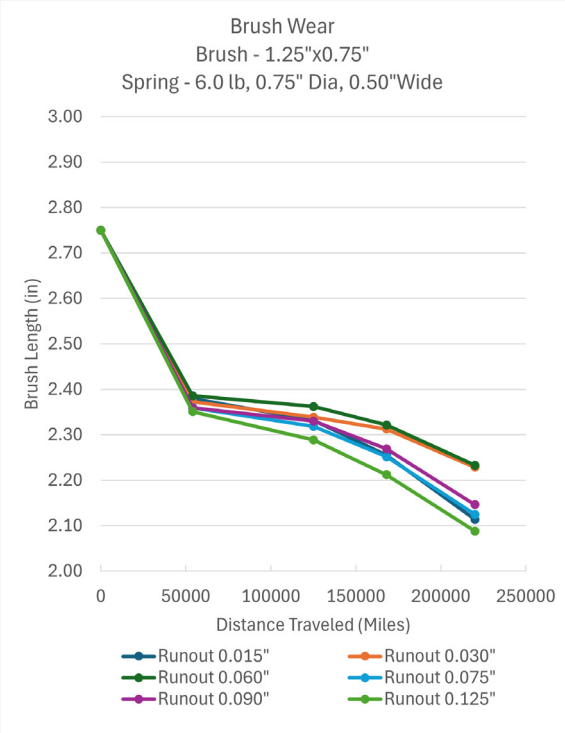

Figure 15C: Brush Wear 6.0 lb., 0.75 in. Dia

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Spring Force

The spring force reduced by an average of about 35.5%. The springs used on the lower TIR rings showed the lowest spring force reduction of about 27% for the 3.75 lbs. (1.7 kg) springs and 26% for the 6 lbs. (2.7 kg) springs. See Figure 16.

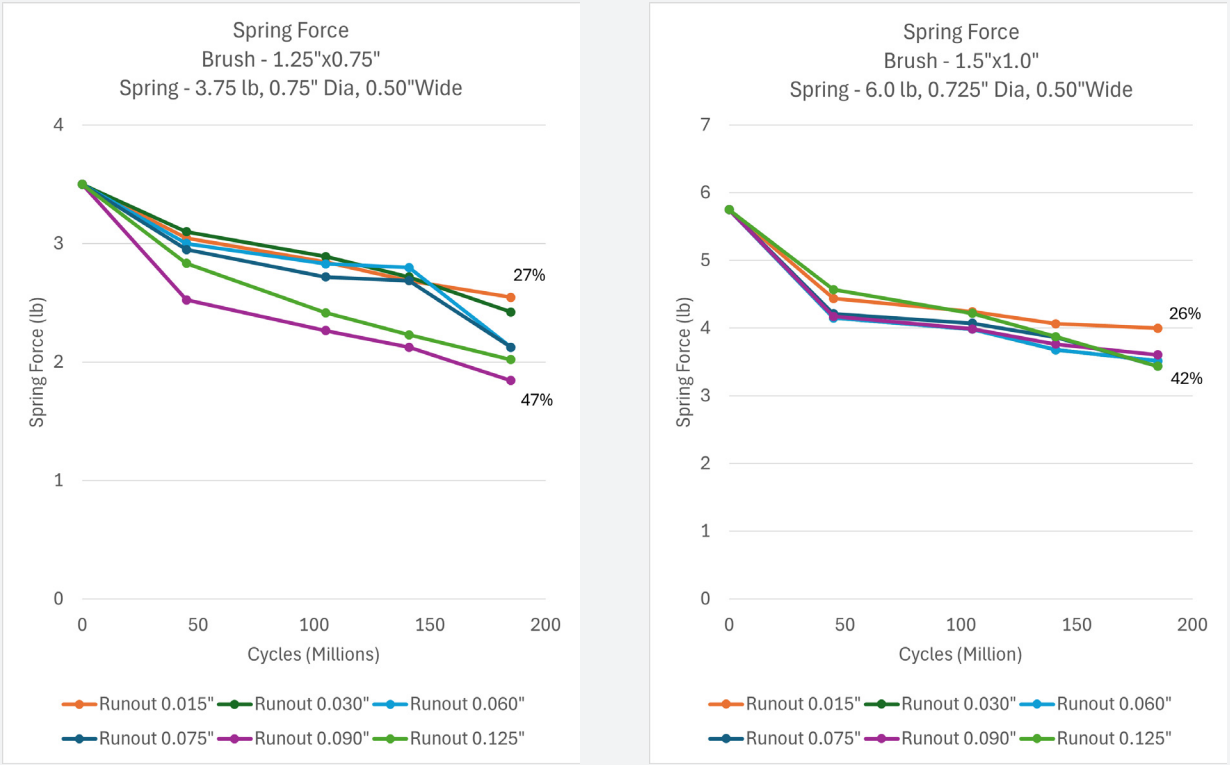

Figure 16: Spring Force Change

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Conclusions

As expected, a higher failure rate was found on the higher total runout rings with the 0.125 in. (3.18 mm) runout rings having the highest failure rate. All the springs that are running on the 0.015 in. (0.38 mm) and 0.030 in. (0.76 mm) TIR rings are still running and have over 185 million cycles on them. On the rings with TIR at and over 0.060 in. (1.52 mm), both single and double coil springs, both diameters, and both spring forces had failures. After closely monitoring failures, we were able to fine-tune the assembly. The fine tuning helped limit the failures and increase the time to failure for the 0.090 in. (2.29 mm) and 0.125 in. (3.18 mm) TIR rings. The quality and consistency of the manufacturing assembly process is critical to extending spring fatigue life. The radius Red Top Pad had more failures compared to the flat Red Top Pad. The radius Red Top Pad is holding the spring in a set position which is likely creating more stress in the spring. Continued testing is needed to draw a definite conclusion. The higher spring force loss is on the higher TIR setups. The spring force was slightly more affected by the ring TIR than the mechanical wear of brush. Double coil springs or larger diameter coil springs do not necessarily mean higher spring life.

Recommendations

- To get the best spring life and protect the hydro AC generator, the slip ring's TIR should be kept below 0.030 in. (0.76 mm).
- A lower slip ring runout may be needed to prevent other issues with the generator.
- Slip rings running at higher speeds (>5000 ft/min (25.4 m/s)) will likely require a lower TIR to maintain the same spring fatigue life.
- The springs should be made by a quality spring manufacturer to prevent early life failures.
- The springs should be monitored closely for failures because any type of spring failure will significantly affect the brush and slip ring life.
- Spring force should be measured each year and if any of the springs have degraded, all the springs on the generator should be replaced.
- Mixing springs with different designs from different manufacturers is not recommended.

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