

IMPROVED MECHANICAL SEAL FOR HIGH PRESSURE PUMPS IN SEVERE ABRASIVE TAILINGS SERVICES

by

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technical papers relating to slurry seal testing and application. Prior to joining BW/IP (1987), he worked for ATI Industries, an aerospace contractor. He is a member of ASME.

ABSTRACT

The cost of high pressure gland water and increasing environmental concerns have made the use of flushless mechanical seals attractive in many slurry applications. One such application exists at Syncrude Canada Ltd (SCL). SCL is a large scale mining operation that extracts crude oil from tar sand. The by product of this process, 150°F sand and water tailings, is pumped to remote storage sites. Tailings pumps are staged in series to reduce the number of pump stations. As a result, tailing line pressures can reach 300 psig. Recently, SCL expanded their tailings storage by adding a new remote site. Due to the high cost of installing and maintaining a gland water line, the decision was made to use flushless mechanical seals in place of packing. At the time of the expansion, a large slurry seal capable of handling the high pressure tailings was not available; however, one seal manufacturer had a prototype seal design that had potential. This seal was chosen with the thought that it could be developed and refined to provide consistent 2000 hr life. Having made the decision to use mechanical seals, the pipeline system was designed with the mechanical seal requirements in mind.

Shortly after startup of the new tailing lines, seals began to fail. With the line now in operation, it was necessary to quickly resolve the problems. An intensive effort was launched to determine why seals were failing. Initial investigation showed the process, controls, and new equipment were working properly. The failures appeared to be seal related. Over the next months, joint work conducted by SCL and the seal manufacturer identified a range of seal problems and interim measures were implemented to increase seal life. As a result of this work, an improved seal design was achieved. Extensive lab and field testing of the improved seal has proven its performance in this aggressive service. Active seals now average 3000 hr, with some seals having a life of over 6000 hr.

INTRODUCTION

Syncrude Canada Ltd (SCL) is the world's largest oil sands plant, producing approximately 12 percent of Canada's oil supply. Bituminous infiltrated sand is open pit mined, extracted using steam and hot water then refined to synthetic crude oil. This sand

tailings system transports 100,000 gpm via five 24 in diameter pipelines.

Three of these pipelines transport slurry to the new South West Sand Storage (SWSS) facility (Figure 1). Each line utilizes two pump houses with three centrifugal pumps in series to transport the sand, water, and bitumen mixture at 150°F (Figure 2). These pumps are driven by 1650 hp variable speed electric motors. They rotate at a maximum 500 rpm and develop over 300 psig discharge pressure in the third stage pump. The specific gravity of the slurry varies from 1.35 to 1.55, depending on the ore feed at the front end of the plant (Figure 3). For the new SWSS tailings project the furthest pump house is approximately 7.0 miles from the plant. Due to the high cost of constructing and maintaining a gland water line for the pumps, a corporate decision was made to use a flushless mechanical seal that relies on face lubrication from the slurry. Mechanical seals were not chosen on their own merit, but rather as a better option than packing.

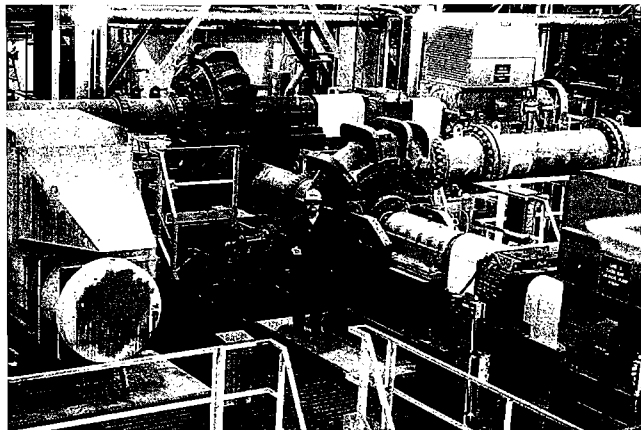


Figure 1. Pump House #690 for SWSS Line, Houses Nine Pumps, Three in Series for Each Tailings Line.

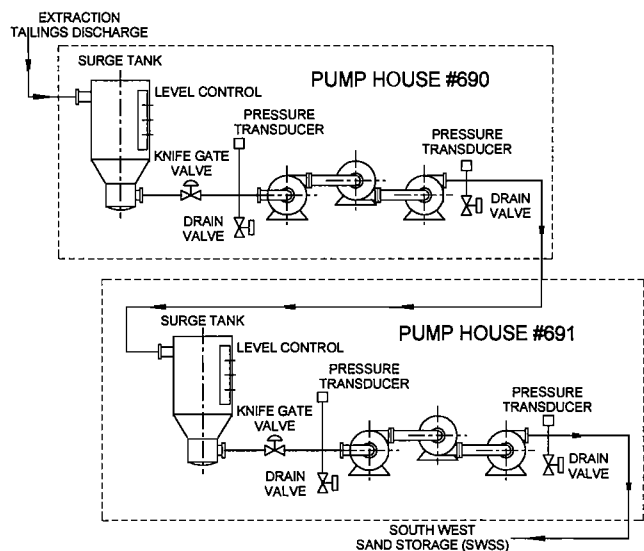


Figure 2. Flow Schematic of SWSS Tailings Line.

The company initiated a worldwide search for a commercially available flushless mechanical seal design to fit a 7.125 in diameter pump shaft. It was found that “off the shelf” seals for this size and service simply were not available. As a result, a prototype design was chosen and the fact that some development could be required was accepted.

SPECIFIC GRAVITY	1.35-1.55
WEIGHT PERCENTAGE	
-WATER	46.0%
-OIL	0.4%
-SAND	53.5%
PARTICLE SIZE DISTRIBUTION	
-150 μm (% passing)	50%
-88 μm (% passing)	30%
-44 μm (% passing)	20%
-22 μm (% passing)	16%
-11 μm (% passing)	9%
-5.5 μm (% passing)	6%
-2.8 μm (% passing)	1%
Notes:	
1.	Slurry is oxygen saturated
2.	Slurry can contain large lumps to 2”
3.	There is considerable variability in slurry composition depending on ore grade

Figure 3. Typical Tailings Slurry Density, Particle Size and Distribution.

The initial seals supplied to the company were from a flushless cartridge design with provision for an external quench (Figure 4). The seal flange design was conventional and mounted to the atmospheric side of the seal chamber, a “dryside design.” The design incorporated a rubber encapsulated bellville spring to provide spring force and a dynamic seal for the rotating seal face. The spring was seated in grooves in the sleeve and rotating face body. Compression of the spring produced a rocking action that was nonclogging. The seal also utilized a rubber cup to support the tungsten carbide rotating face and act as a seal. The stationary sintered SiC face was mounted in the flange with an O-ring seal.

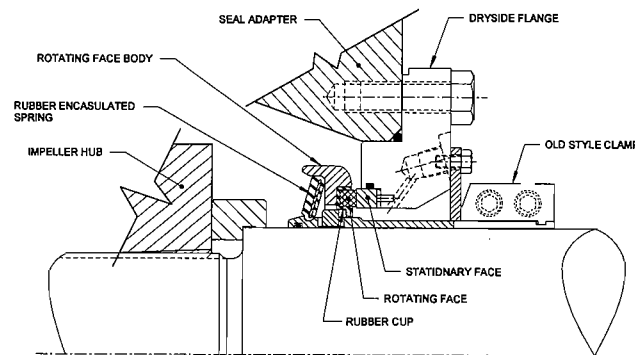


Figure 4. Group 5 Prototype Seal Cross Section, Dryside Flange Design with Early Clamp and No Quench.

PREPARING THE TAILINGS LINE FOR MECHANICAL SEALS

After the decision was made to go with mechanical seals, the pipeline system was designed around the requirements of the seals and avoidance of pump wear (Figure 3). The system included surge tanks on the suction side of the first stage pumps to hydraulically

isolate the pump houses. Flush sequences were used to clean the sand out of the pumps before shutdowns and to flood the pumps prior to start up. In addition, variable frequency drives were chosen to ensure all three stages turned at the same speed. A suction pressure override feature was added to switch process control from level to pressure control, if the first stage pump suction pressure dropped below the seals minimum operating parameters. Finally, procedures to protect the seals during emergency manual operation were implemented. A totally automated system was designed for "normal" startups and shutdowns where the pump speeds, tank levels, and even flush valves were electronically controlled, eliminating the possibility of incorrect valve sequencing in the remote pump houses.

The new slurry pumps were subjected to extensive checks to ensure shaft deflection, axial float, bearing clearances and seal chamber alignment were suitable for the new seals. The piping in the pump house was subjected to several QA checks to ensure piping loads did not cause deflection of the pump casing and thus ruin the alignment of the seal inside. The massive pump bases were laser aligned and doweled in to ensure no movement occurred. Finally, the process alarms and control board corrections were real time monitored for future examination should a problem occur. As a result, there were no major changes to the process control system after startup.

STARTUP OF THE NEW TAILINGS LINES

On startup, the seals ran well during the 72 hr commissioning when the lines were transporting water. When the line switched to slurry, the seals overheated to failure within minutes. To resolve this problem, the spring load was reduced and pump house cleanup water (trucked water) was piped to the quench port in the seal flange. The quench added cooling to the inside diameter of the seal face. These changes increased seal life from minutes to days and allowed operation of the plant while attempts were made to understand the seal failures. Initially, it was speculated the seals were failing due to process fluctuations that created adverse pumping conditions. Process control data were thoroughly analyzed after each successive failure with no repeatable pattern or sign of the root cause. The procedures and control logic were then reexamined, but no reason was found for the random seal failures.

INVESTIGATING SEAL PROBLEMS

After a number of failures, it was found some seals failed because they were severely over compressed. Careful inspection revealed the sleeve had moved during operation, the sleeve clamp did not provide adequate radial clamping force (Figure 4). To eliminate the problem, an improved clamping device was designed and installed (Figure 5). Looking for other possible causes for seal failures, it was hypothesized the spring load may be too high or inconsistent. To investigate this possibility, 10 springs (mounted on sleeve and rotating body) were tested using a load cell arrangement to measure spring force (Figure 6). This testing clearly showed the elastomer coated spring had a high spring rate. It only took 0.050 to 0.080 in deflection to achieve the desired spring force. To obtain a consistent spring load, each seal would require a different setting plate thickness. To simplify the process of setting the seal and achieve a consistent spring force, a weight was built and used to deflect the spring to the desired load. This led to the discovery that the rubber coated spring would creep for several hours after the deadweight was applied. Assembly specifications were changed to allow time for the elastomer to creep until the deflection stabilized.

Because of the high spring rate, axial movement in the pump was investigated to determine if it could be reduced. The thrust bearing of the pump was located at the coupling end. The shaft radial bearing floated toward the impeller due to unbalanced hydraulic loads acting on the impeller and to shaft thermal growth.

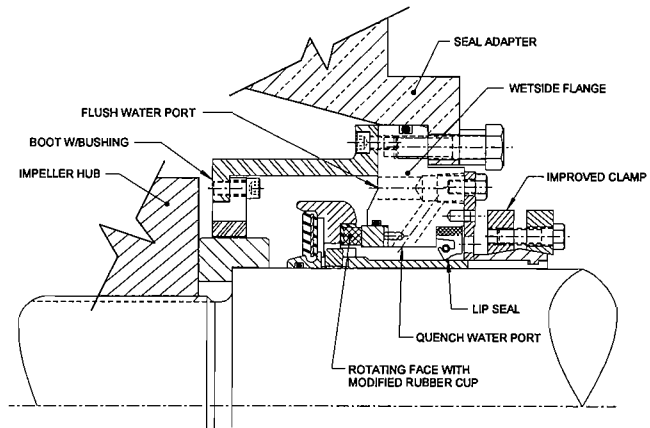


Figure 5. Group 5 Interim Seal Cross Section, Wetside Flange Design with Boot/flush, Quench, and Improved Clamp.

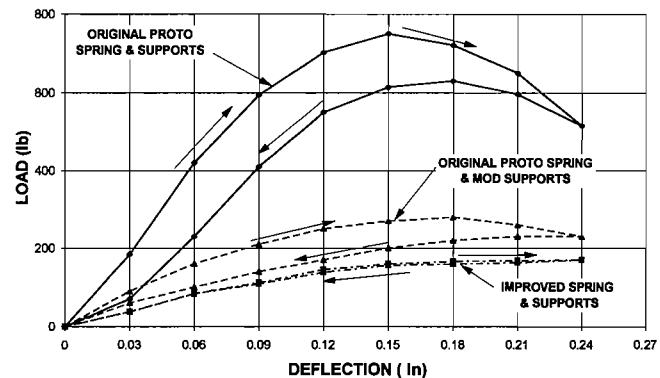


Figure 6. Group 5 Spring Force vs Deflection Curves Comparing Prototype Spring Installed in Seal and Between Flat Plates and New Spring Installed in Seal.

To reduce axial movement, the thrust bearing was redesigned to place it close to the seal. This effectively fixed the shaft at that location and directed shaft thermal growth toward the coupling. This resulted in 0.015 in less axial movement. Although the bearing modification helped, the 0.028 in pump casing expansion, due to pressure, could not be changed. To correct the problem, the spring rate needed to be reduced, so that normal shaft movement would have little effect on spring load.

Monitoring the Seals and Pumps

By this time, intensive field and research test programs had been initiated. The first problem was seals were failing randomly in production tailings lines, not in a research lab. At the operations site, a team was formed to find the root cause of the seal failures. This team listed every possible process and mechanical change or movement that could effect seal performance. High speed chart recorders were purchased and the pumps and seals were instrumented to look for clues. The suction pressure on the first stage pump and the discharge pressure on the third stage pump were instrumented along with the horizontal, vertical, and axial movement of the pump shaft. In conjunction with this, the relative seal face temperature was measured by embedding a thermocouple in the stationary seal face approximately 0.080 in from the surface. In addition, a proximity probe was used to measure face wear. Finally, temperature of the slurry was measured. The system was then operated under normal conditions, while the instruments monitored the equipment real time using the high speed chart

recorders. It was found that seals ran with small temperature variations for a period of time, then showed small temperature spikes that became progressively worse. Finally, the proximity probe began to indicate face wear, which continued till the seal failed.

At this point, the only noticeable temperature improvements were the addition of seal quench water and reducing the spring load. To improve heat transfer from the rotating faced to the liquid, the rubber lip molded around the inside diameter of the face was cut out, exposing more of the seal face to the quench water. Next, a lip seal was added to the atmospheric side of the flange to totally immerse the area under the faces in ambient water (Figure 5). The quench water was then changed from top feed to bottom feed to ensure the seal inside diameter was immersed in water. These changes lowered quench water requirements from 3.5 gpm to 0.5 gpm. In addition, quench water flowmeters were installed to record and control the exact amount of flow to each seal. Since it was recognized the seal operated better in water, for production reasons, a boot was installed on all but three test pumps. The boot formed a water shield around the mechanical seal (Figure 5). Five to seven gpm clean flush water was injected into the boot at a higher pressure than the slurry. The boot required a reliable high pressure water supply. If the flush water supply failed, the booted seal would fail within a short time. Flush water flow had to be carefully monitored.

The seal price and installation labor costs were high but production costs were enormous. There simply had to be a faster turnaround when a seal failed. The initial seal design called for a seal to be installed from the pump bearing side (dry side). This meant the pump, gearbox and electric motor had to be realigned during every seal replacement. This could take up to 24 hr with the size of equipment that had to be dealt with. A "wet side" seal was designed and installed with a significant reduction in downtime, often less than one shift (Figure 5). This seal could be installed by removing the suction spool, the suction side plate, the impeller, and the seal. The alignment did not have to be disturbed, only checked.

Field experience and pump dynamic measurements indicated that seal face load was still a concern. Earlier static tests proved face loads were high, however, it did not address how the seal performed under dynamic conditions. A rotating rig was built that allowed testing for two things. First, the normal cartridge was tested varying seal chamber pressure. Second, the seal chamber pressure was fixed while face load was varied (essentially varying the seal balance). Various face combinations (including sintered SiC vs WC, sintered SiC vs SiC and WC vs WC) were used to find any measurable differences with surprising results (Figures 7 and 8). The prototype seal exhibited a distinct face load vs stuffing box pressure "envelope" for each face material combination tested. For example, with rotating WC vs stationary sintered SiC, the seal operated ideally while pressure was increased, it only began to show unstable behavior as the pressure was being relieved. This repeatable behavior matched the "random" field failure observations. It was now clear the root problems could be found.

Additional dynamic testing was conducted by the seal manufacturer, using the prototype seal cartridge as installed in the pump (without boot and flush). This testing was conducted in water at normal process conditions. Prior to each test, the seal was carefully prepared to make sure spring load was consistent from test to test. As mentioned earlier, the rotating face was vulcanized in a rubber cup (Figure 5). The rotating face/cup was pressed into the rotating body during seal assembly. It was difficult to maintain consistent face flatness with this design, small differences in radial squeeze had a considerable effect on face flatness. Although new seal faces were installed for each test and carefully lapped flat, seal performance varied from test to test. Inspection of the seal faces after test showed signs of heavy face loading, with all faces having some degree of heat checking. The interesting thing was some seal

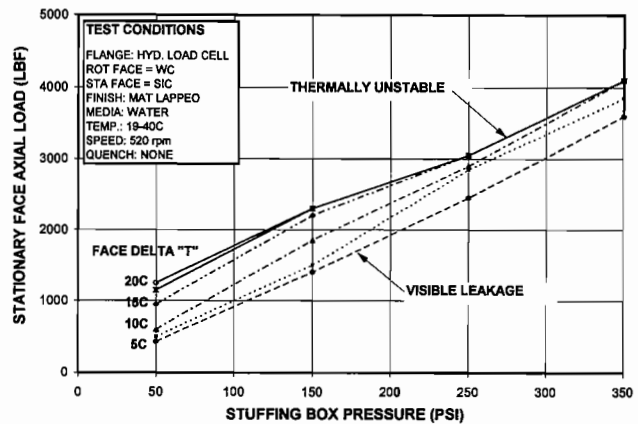


Figure 7. Load vs Pressure Operating Envelope, Tungsten Carbide vs Sintered SiC, Prototype Seal Face Design, Including Face Delta Temperature.

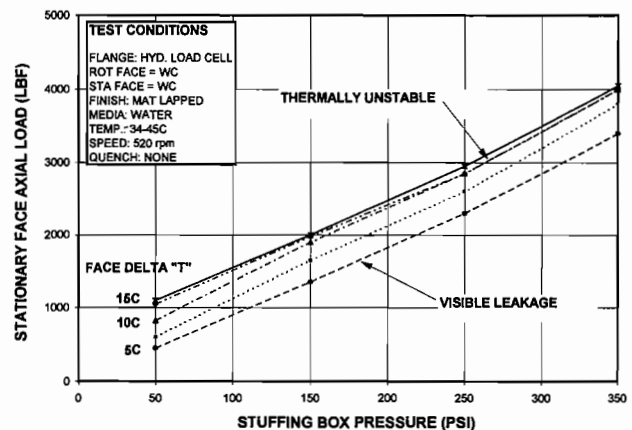


Figure 8. Load vs Pressure Operating Envelop, Tungsten Carbide vs Tungsten Carbide, Prototype Seal Face Design, Including Face Delta Temperature.

faces had little damage while others suffered severe wear. This was also seen in the seal face temperature. The seals that ran well had light contact on the inside diameter of the face while, seals that ran with high face temperatures had contact at the outside diameter. Inspection of these faces showed they had distorted due to relaxation of the rubber and were no longer flat.

DEVELOPMENT OF THE IMPROVED SEAL

To this point, all efforts had focused on determining why seals were failing prematurely. Although small changes had been made in pump operation, the process and controls were working well. Results clearly showed the problems were seal related. Early experiments with standard seal settings indicated the spring load was excessive and varied from seal to seal. Additional testing confirmed that a standard compression would not work. The use of a dead weight made it possible to achieve a consistent spring load, but due to differences in spring rate, compression varied from 0.050 to 0.080 in. This was not good since the pumps typically had about 0.050 in axial movement, a lower spring rate was needed. Dynamic testing with the variable balance test rig showed that seal balance ratio was too high and that available face materials would not work well unless balance ratio and spring load were reduced. Further lab testing of the seals in controlled conditions revealed

that the existing seal design had a problem with face flatness. The rubber cup surrounding the rotating face contributed to distortion that adversely affected seal performance.

Reducing Spring Rate

Knowing that high spring rate and axial pump shaft movement resulted in erratic seal face load, the first thing to eliminate was the high spring rate. It was desirable to have a spring capable of maintaining a constant load over a reasonable displacement. Initially, rubber coated springs were load tested between flat plates without radial compression. Loads measured this way were quite low but still significantly higher than the uncoated (bare) spring. Next, the same springs were fitted on a sleeve with the spring OD supported on a flat plate. This time, loads were significantly higher and hysteresis increased. Similar tests were conducted mounting the spring in a rotating body and finally with both ID and OD mounts. The load curve of the spring mounted with ID and OD supports was more than twice the unconstrained spring. Results of this testing correlated well with previous testing. It became clear the high spring rate was primarily due to the compressive stresses in the elastomer. A series of tests followed to find ways to reduce the elastomer load component (referred to as false elastomer load or FEL) produced from radial compression of the elastomer. The outcome of this testing produced a spring/support system that had the same load characteristics as the coated spring between flat plates (Figure 5). The aforementioned work showed that the rubber coated spring as installed in the seal could achieve a flat spring curve. Unfortunately, even with improvements to the supports, the coated spring load curve was still too high. Additional work was conducted to achieve a technology that eliminated FEL. If the coated spring was mounted in a fixture at the desired deflection and heat treated for several hours at an elevated temperature, the spring load was reduced to that of an uncoated spring. This testing confirmed that a consistent low spring load could be achieved by modifying the spring, rubber coating, and spring supports. Ultimately, the spring, rubber coating, and supports were redesigned.

Reducing Seal Balance Ratio

Another static test rig was built to measure the combined hydraulic and spring forces acting on the seal faces. This rig allowed measuring the actual force acting on the seal faces over a range of pressures from zero to 300 psi, and had the capability of testing different spring support designs. The rubber coated spring is supported, at the outside diameter, by the rotating face body and in a groove in the sleeve, at the inside diameter. Its behavior is similar to a bellows seal. The early seal design was tested first and found to have a high balance ratio. This correlated well with previous lab test results. A series of tests followed to evaluate different spring/support configurations. The result was a low balance design that produced hydraulic loads similar to those found to work well in previous balance test work.

Lab and Field Testing of Interim Design Seals

At this point, interim design seals were built incorporating the above improvements (Figure 5). These seals still had the rubber cup rotating face, but special modifications were made to minimize distortion. A series of lab tests on water were then conducted. A significant improvement was noticed. In fact, some seals showed no measurable wear, seal faces looked like new after testing. Inspection of these faces showed them to be flat or have a small amount of convergence. Unfortunately, some interim seals ran poorly. Inspection of these faces showed the faces had distorted and contacted at the OD. Based on the above success, interim seals were built for field testing. Because of the face distortion problem, each interim test seal was lab tested on water before going into field test. A number of interim seals were tested in first, second,

and third stage pumps. Interim seals were field tested in pumps with boot and flush and flushless. Compared to the old design, these seals exhibited stable seal face temperatures, provided consistent performance and ran better at high pressure. Seals of this design set new records in their respective high pressure pumps in the field.

Putting it all together, the improved seal, based on lab test results, it was clear a seal face redesign was needed. Even with modifications to the rubber cupped rotating face design, face distortion continued to be a problem. Results of testing to date clearly showed face flatness was critical to increase life and reduce water consumption. In addition to distortion caused by the rubber mount, face flatness could be affected by pressure and thermal distortion and flange induced distortion. A number of seal face designs were considered for the improved seal (Figure 9). Many hours of lab testing in conjunction with finite element modelling produced a design that maintains flat seal faces over the range of operating conditions found in this difficult service.

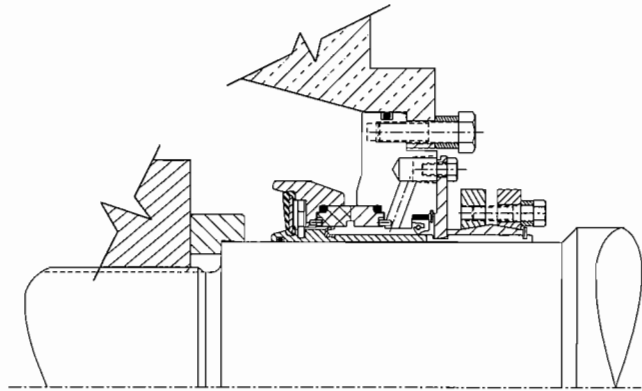


Figure 9. Group 5 Improved Seal Design, Flushless Cartridge with Quench.

Results of previous tests indicated that the seal would need to be accurately loaded for best performance. To attain this goal, a new spring was designed. The new design is deflected to 0.240 in when installed and has little change in load when it is unloaded to 0.140 in deflection (Figure 5). The 0.100 in constant load region is more than adequate to handle worst case pump shaft axial shuttle. Using results from interim seal designs, spring supports for the improved seal were designed to produce a fixed low balance under all operating pressures. The new spring and support system showed excellent repeatable performance in lab testing, even under adverse operating conditions.

The final consideration was seal face materials. Earlier dynamic testing in water had shown that SiC vs WC could handle a higher contact pressure than sintered SiC vs SiC with the prototype design. For this reason, sintered SiC vs WC was chosen for the first improved seals installed in the field. At this time, the improved design has shown outstanding results with sintered SiC vs SiC in lab testing and will be field tested at the plant. In addition, SiC vs SiC faces have shown excellent performance in many other applications with the improved design.

To date, a number of improved seals are running in pumps in the field at SCL and each seal has provided consistent good performance. These seals are installed as flushless seals (no boot or flush) and have a 0.5 gpm quench on the atmospheric side of the seal. They are operating in second and third stage pumps with over 3000 hr and are still running well. Booted prototype and interim seals are now averaging 3000 hr, with some seals having a life of over 6000 hr (Figure 10).

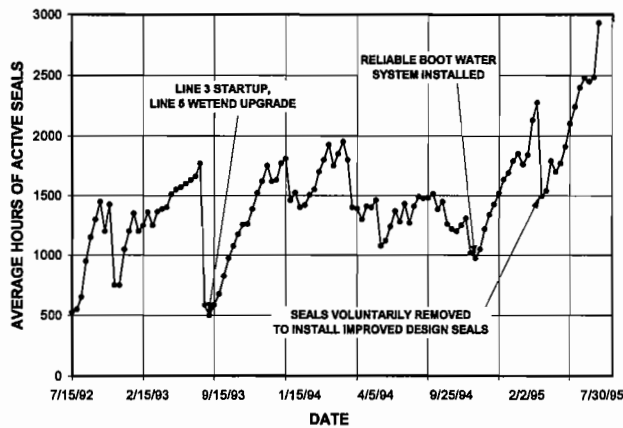


Figure 10. Swss Tailings Line Seal Chart Comparing Avg Hr of Active Seals vs Time.

CONCLUSIONS

It is now possible to economically use a large diameter flushless mechanical seal in an abrasive high pressure sand slurry system and obtain seal life over 3000 hr. Special attention must be paid to seal face flatness, spring load and balance ratio to ensure that face loads are within a narrow operating envelope. The initial 2000 hr seal life goal has been exceeded and the current goal is a consistent 4000 hr with low water usage.

The success of this development effort has been in large due to a cooperative exchange of ideas and information between SCL and the mechanical seal company. The close working relationship between the two organizations clearly made the difference in overcoming problems and ultimately developing the improved seal.

There are several important lessons that have been learned through this development process:

- The mechanical seal must have sufficient axially movement to accommodate normal pump shaft movement.
- The seal must maintain a nearly constant spring force over the pump's axial shuttle to handle high pressure slurries.
- Flat seal faces and low balance ratio are required to achieve best seal life in high pressure slurries.
- The seal chamber must be designed to self vent and provide adequate circulation in the seal area.
- The seal faces rely on the slurry for lubrication, hence the seal must see a positive pressure and liquid at all times during operation.
- A quench increases seal life in higher temperature slurries. Slurry type and chemistry must be considered when deciding whether a quench is required. Normally, dissolved solids or caustic or acids will require a quench a higher temperatures than a straight water/solids slurry.

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