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Breakthroughs in Active Rotating RCD Development: A Discussion on Technical Development and Performance Evaluation Through Testing

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Abstract

Typical Rotating Control Device (RCD) offerings used for offshore Managed Pressure Drilling (MPD) operations do not include active sealing elements with the ability to rotate. An active/rotating RCD can significantly increase seal life through added operational control and real-time monitoring. This paper presents major performance advancements for an active/rotating RCD using hyperelastic FEA of the sealing element, rapid prototyping through correlation with testing, and development of a high-performance bearing system.

Improvements to the sealing element life were realized by utilizing FEA techniques to optimize the seal design and correlate with full-scale test results. The results of this analysis/design/test approach will be discussed, emphasizing the evaluation of the sealing element with an off-center drill string.

The development of a new bearing system is also presented with a comparison to roller bearings. The bearing system was designed to withstand rigorous testing simulating operational conditions such as side loads due to off-center drill string. The laboratory test results demonstrate the enhanced performance of this RCD and validate its suitability for demanding applications.

Finally, several product use cases for this RCD are explored, demonstrating its ability to withstand pressure from above and below for broad operational applications, such as In-Riser, Below-Tension-Ring (BTR), and Riserless systems. Future work includes API 16RCD (API 16RCD, June 2022) Qualification and Service Application Testing to replicate anticipated field conditions.

Introduction

An RCD is a piece of equipment "allowing for rotation and axial movement of the drill string while simultaneously containing wellbore pressure (API 16RCD, June 2022)." They are critical equipment for MPD operations allowing for additional methods to manage wellbore pressure. As MPD operations become more widely adopted, there is a need for these devices to operate in tighter drilling windows with more consistent performance. RCDs often do not last entire hole sections and must be retrieved for unplanned maintenance. If they do last an entire hole section, it comes at the expense of rotary speed, rate of penetration

(ROP), and tripping speeds. Furthermore, there is a need for a low-profile in-riser RCD that is a capable of running in smaller rotary table sizes (i.e., 49-1/2″).

To address these performance gaps, a detailed look at the available passive/rotating and active/non-rotating RCDs highlights the need for a robust, low-profile active/rotating RCD. An active RCD sealing element can engage and disengage on the drill pipe as needed by the driller, limiting the amount of time the sleeve must be engaged on the drill pipe. This reduces wear and can extend the life of the sealing element for a complete hole section. Similarly, the rotating design allows the sealing assembly to rotate with the drill string on a set of bearings, further reducing element wear over time as compared to non-rotating designs.

The design philosophy for the active/rotating RCD was broken into two phases: (1) iterative prototype design and testing, followed by (2) API qualification and Service Application Testing. This paper focuses on Phase 1 and utilizes the guidelines for an Active RCD per API 16RCD. The goal of Phase 1 was to reduce risk through iteration and optimization of the design prior to completion of full RCD qualification in Phase 2. For an active/rotating RCD, the two main design components are a highly wear resistant sealing element and a robust bearing and rotary seal system built to withstand operational axial and radial loading conditions.

Hyperelastic finite element (FE) analysis enabled rapid prototyping and test verification of the active sealing element. Standard, off-the-shelf bearing solutions were evaluated for this RCD, resulting in the identification of an alternate solution currently used in the downhole tool space. Pressure-velocity (PV) curves were used to down-select both the bearings and rotary seals. Beyond API 16RCD testing, bearing system life under the application of full axial and radial loads was identified as a key risk for Phase 1 mitigation. Through testing, the rotary seals selected in Phase 1 were shown to exceed the API endurance requirements.

Specifically, this paper will cover the following topics:

- Typical RCDs in the market today
- Active/Rotating RCD design considerations and applications
- Phased design approach to mitigate technical risk
- Sleeve analysis & test program
- Bearing & rotary seal down-selection & test program
- Phase 1 test results

Statement of Theory and Definitions

Typical RCDs in the Market Today

A typical use of RCDs in MPD systems is to control bottomhole pressure (BHP). A conventional drilling system is considered "open to atmosphere," and BHP is controlled via changes to mud weight or pump flowrate. Changing mud weight is effective but not immediate and changing pump rate is immediate but not effective. In formations with narrow or unknown drilling windows, this becomes a limitation of the system. In a "closed loop" MPD system, BHP can be controlled with changes in mud weight and pump flowrate but also with surface back pressure (SBP). This additional control gives drillers the ability to respond to changing formation conditions quicker and stay within the drilling window easier when compared to conventional systems.

The appeal of MPD is performance and efficiency at the expense of complexity and cost. For the MPD value proposition to make sense, the former must outweigh the latter. A robust and effective RCD that reliably isolates the wellbore without sacrificing drilling performance tips the scale in favor of drilling with MPD. An unreliable RCD that requires maintenance during operations tips the scale the other way. Drilling limitations imposed by RCD performance and maintenance are a major pain point in the MPD market today.

An RCD with improved and consistent performance would allow wells to be planned and drilled with less risk of non-productive time.

Typical RCD types include passive/rotating and active/non-rotating designs. Passive/rotating RCDs have a simple sealing mechanism (interference fit between the sealing element and drill pipe) coupled with a rotary system that allows the sealing element to rotate with the drill pipe. Active/non-rotating RCDs have a more complex sealing mechanism (similar actuation to that of an annular BOP) but do not contain a rotary system. MPD wells have been drilled with these systems for years, but not without limitations, driving the development of the active/rotating system. While the active/rotating system will become more sophisticated (active sealing mechanism and rotary system), the understood trade-off is superior performance that justifies the complexity. Although active/rotating concepts have been utilized in rotating heads (BOPs), this paper does not draw comparisons to these products since this RCD is intended for different applications.

Active RCD Design Considerations

As discussed, the two main design elements of an active/rotating RCD are the sealing element (a sleeve for this active/rotating design) and the bearing system. To design and evaluate these components, it is important to first define the requirements for the RCD for use of application. The key parameters that drive the design of an active/rotating RCD include: (1) axial loads from the drill string; (2) rotational speed of the drill string; (3) wellbore fluid temperature range; (4) radial loads from off-center drill string; (5) pressure differential across the RCD sleeve; (6) tripping speeds; (7) API life target vs. usage case; (8) system break-out torque; and (9) availability of activation pressure.

A comparison of the key design features and performance trade-offs for different types of RCDs and their associated complexities are shown in Table 1. While an active/rotating design may have the highest performance metric, it also has the most complex design. Due to the complex nature of the active/rotating design, it was paramount to separate the sealing sleeve and bearing system prototyping to allow for quick technical learnings.

Table 1—Design Matrix Comparing RCD Type versus Complexity and Performance

RCD Type	Sealing Mechanism	Rotary Design	Stripping Wear Resistance	Rotational Wear Resistance
Passive / Rotating	Simple	Complex	Low	High
Active / Non-Rotating	Complex	N/A	High	Low
Active / Rotating	Complex	Complex	High	High

The active/rotating RCD is designed to operate by energizing a sleeve with hydraulic pressure to seal on the drill-string. When closed on the drill pipe, the sleeve must provide sufficient grip, exceeding the breakout and running torque of the bearings to prevent slip in the sleeve. This allows the sleeve assembly to rotate with the drill string, limiting relative motion between the sleeve and drill pipe to axial translation of the drill pipe during operations. The layout of this RCD is shown in Figure 1. To engage the sleeve on the drill string, an activation chamber is defined, wherein activation pressure can be built up to positively displace the sleeve or removed to allow the sleeve to retract. A symmetric bearing system encompasses the sleeve and rotor assembly, capable of withstanding the axial and radial loads of the system imparted by the wellbore pressure and drill pipe movement in both directions. Wellbore facing rotary seals prevent mud ingress from the wellbore into the RCD.

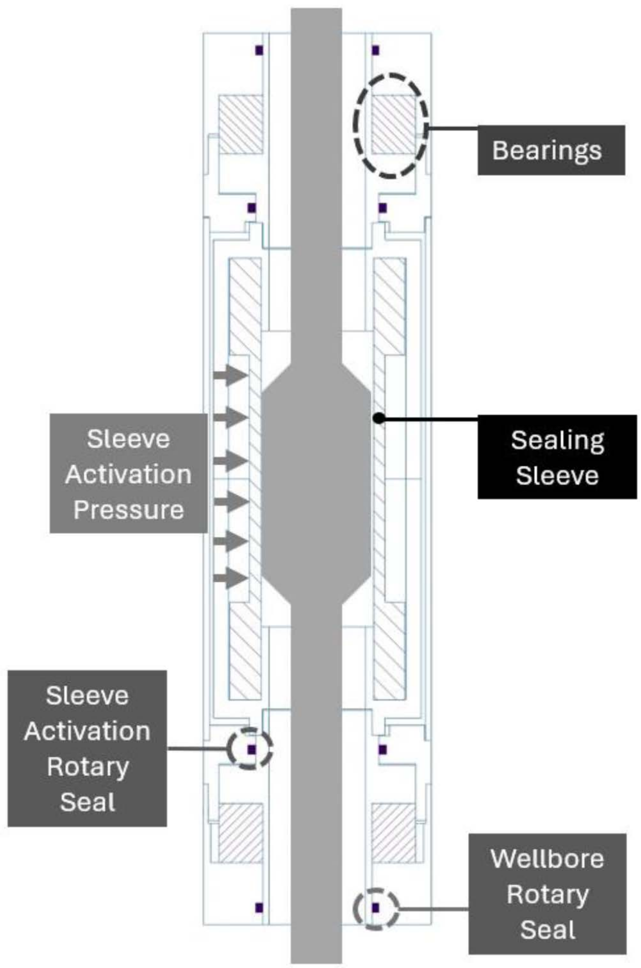

Figure 1—Key System Architecture of Active/Rotating RCD

Applications for Active RCD

RCDs are used in a variety of MPD applications. The ones of particular interest for this active/rotating RCD are:

Offshore Deepwater: RCDs on dynamically positioned rigs such as drill ships or semi-submersibles are installed below the tension ring as part of the marine riser system in an Integrated Riser Joint (IRJ) and referred to as BTR (Below Tension Ring).

Riserless Drilling (RLD) is employed during low pressure, top-hole sections that do not require installation of a BOP. Water-based drilling mud and cuttings are returned directly to the seabed rather than being circulated back to the surface through a riser. This is commonly referred to as "pump and dump." For the remaining well operations when reservoir pressures are expected and a BOP is required, several limitations prevent RLD from being practical and the rig transitions to conventional, marine-riser drilling. However, there are drawbacks to marine-riser drilling that have pushed the industry to re-evaluate the use of RLD beyond just top-hole sections. Increasing the reliability of RCDs is just one of several technical innovations required to make this practical. In an RLD system, the RCD prevents drilling returns from discharging to the environment. When coupled with a subsea pump, the RCD provides the flexibility of drilling with a dual-gradient RLD MPD system.

Technical Development Path

Prototype testing of the complete RCD adds complexity to the scope, test infrastructure, and material cost while slowing down the iteration process. For these reasons, the first phase of this RCD development project focused on rapid prototyping of the sealing sleeve separate from the bearing system, allowing for quick iterations on lower-complexity test set-ups. Similarly, the rotary seals were tested in dedicated fixtures ahead of full bearing system tests to optimize life and operating pressures before system integration.

The RCD must be able to affect a wellbore seal under several conditions, including:

- across a range of wellbore temperatures,
- while the drill string is stationary,
- during axial drill pipe movement while tripping, stripping, and connections
- during axial movement combined with rotation of the drill string while drilling ahead, and
- with off-center drill pipe.

De-coupling the sealing sleeve and bearing system allows for testing the sealing sleeve for all operational functions except for rotation. For an active/rotating RCD, the interaction between the sealing sleeve and the bearing system is critical, ensuring that the activated sleeve has sufficient grip to prevent slippage of the drill string which will increase wear. While testing separately, focus must still be given to the rotational requirement, ensuring the friction, breakout and running torques, and resultant axial loads measured from the sealing sleeve can be compared to the bearing system to ensure full system functionality.

A major focus of Phase 1 was to demonstrate the ability to pass the formal API 16RCD qualification tests. This allowed for the design to focus on wellbore pressure integrity across the temperature range and stripping life. The sealing sleeve test program excluded the Dynamic Pressure Rating Test, which will be conducted along with the bearing system in the next phase.

In parallel, the rotary seals and bearing system were tested under rotation with pressure, mimicking the loads from the sealing sleeve during the Dynamic Pressure Rating Test without adding the sleeve to the test set-up. Stripping breakout force tests were performed on the sleeve prototypes to determine the Coefficient of Friction (CoF) between the sleeve and drill string, allowing for calculation of sleeve grip for a direct comparison to bearing system torque. Table 2 and Table 3 summarize the tandem analysis and test approach for the sleeve and bearing system.

Table 2—Sealing Sleeve Development Plan – Phase 1

	Analysis	Test
Closure Shape	✓	✓
Sealing Characteristics	✓	✓
Static Pressure	✓	✓
Fatigue		✓
Stripping	✓	✓
High Temperature	✓	✓
Low Temperature		✓
Off-center, Side Load	✓	
Dynamic Pressure (Rotation)	✓	

Table 3—Bearing System Development Plan – Phase 1

	Rotary Seals	Bearings & Rotary Seals
Static Pressure	✓	✓
Off-center, Side Load		✓
Dynamic Pressure (Rotation)	✓	✓

Description and Application of Equipment and Processes

The Phase 1 de-risk and testing effort focused on mimicking the qualification tests defined in API 16RCD to the extent possible with a de-coupled sleeve and bearing system. Additional characterization tests were performed to assess design functionality and allow for correlation to FE models and the interaction between the two sub-systems.

A modular sleeve test fixture was designed to anticipate sleeve design iterations throughout the process and enable utilization of existing BOP testing infrastructure.

Rotary seals were tested in a dedicated fixture to optimize performance ahead of testing with the full bearing system in a dedicated fixture. The bearing system fixture is also modular, allowing for evaluation of multiple rotary seal and bearing concepts. The fixture can apply full axial and radial loads to simulate operational conditions, including off-center drill pipe.

Sleeve Hyperelastic Analysis

Hyperelastic FE analysis of the sleeve was conducted using ABAQUS software, setting up the Phase 1 execution plan using a design, test, correlate, optimize and iterate structure. The goal was to maximize analytical iterations using FE analysis, reducing overall project manufacturing and test schedule and budget by minimizing the number of sleeve prototypes produced. Each prototype cycle should reduce design time with the addition of more test data, allowing for greater performance increases for each prototype.

Use of Analytical Modeling for Rapid Prototyping. Subscale material testing to obtain mechanical properties for input to FE models is critical, especially for hyperelastic analysis of elastomers. Key parameters:

- Tensile & tear strength (ambient and elevated temperatures)
- Compressive strength
- Volumetric compression
- Bonding scheme and interactions for dissimilar materials in the sealing assembly
- Shear strength

Figure 2 highlights the stress-strain curve of the sleeve elastomer material selected in Phase 1, covering the entire tensile and compressive range. This material curve was used for all FE models.

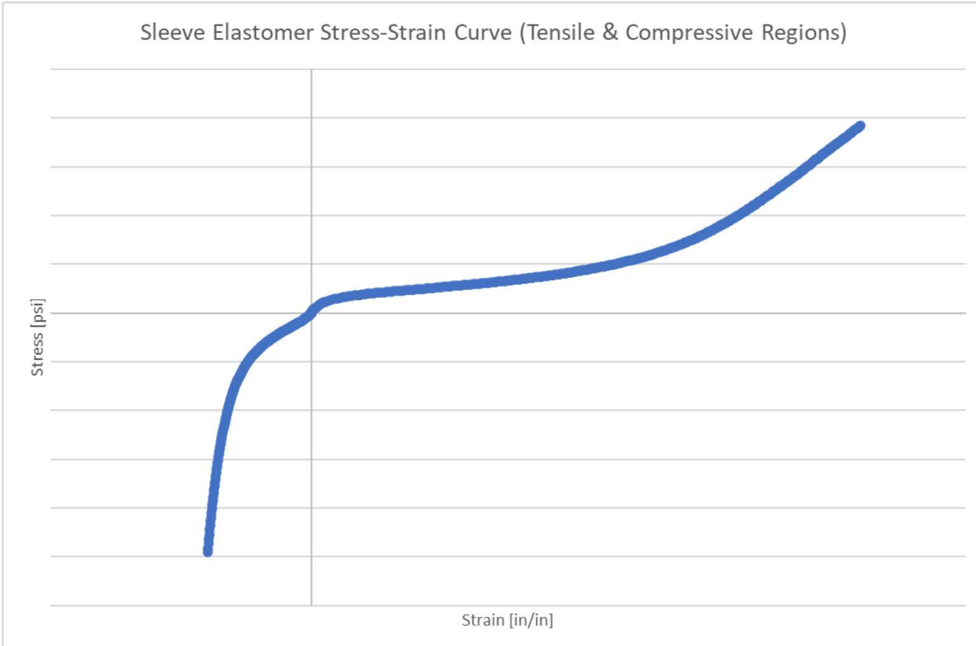

Figure 2—Sleeve Elastomer Stress-Strain Curve: Uniaxial Tension & Compression

The baseline sleeve design targets were developed using three test data points from early prototype concepts tested in 2017, along with lessons learned and best practices from BOP sealing technology. The iterative design process built upon this test data, including modelling of failures, from each prototype to enhance the FE models and update the design targets. By the end of Phase 1, the design targets included:

- Closure shape optimization (off-center drill pipe with a side load),
- Minimizing activation pressure required to establish contact pressure to affect a wellbore seal,
- Reducing strains and deflections for all load combinations, and
- Accounting for high temperature effects (operating temperature range and heat generated from friction during TJ passage).

Closure Shape & Side Load Analysis on Sealing Sleeve. The first of two main design challenges was elimination of a sleeve fold encountered during activation, observed during the first two prototype tests. The fold resulted in non-uniform sleeve to drill pipe contact, preventing a wellbore seal. Model refinement included replicating the fold in FE analysis and adjusting the sleeve geometry to eliminate the fold altogether, highlighting the need to study the closure shape of the sleeve using a 360-deg model. Early concept analysis utilized smaller wedge shapes with symmetry planes, adding artificial hoop stiffness. Limitations in hyperelastic FE systems of equations also resulted in over prediction of collapse under external pressure (Papadakis, 2008), necessitating the development of an exaggerated geometry model (EGM) to amplify and understand the fold formation.

The stiffness and aspect ratio of the sleeve's primary sealing length, along with the maximum radial distance travelled from the relaxed state to drill pipe engagement, were found to be the most effective design levers to ensure uniform closure. Figure 3 highlights the key learnings from the closure shape study performed while designing Prototype 3 after a fold was observed on Prototypes 1 and 2. Evaluation of Prototype 1 using the EGM resulted in a fold that matched test observations. New concepts were also evaluated with an off-center EGM pipe while under side load, demonstrating the need for a stiffer cross-section to prevent folding with the final Prototype 3 design. While API 16RCD does not require side load consideration for qualification tests, the effects were studied in Phase 1 to further reduce technical risk.

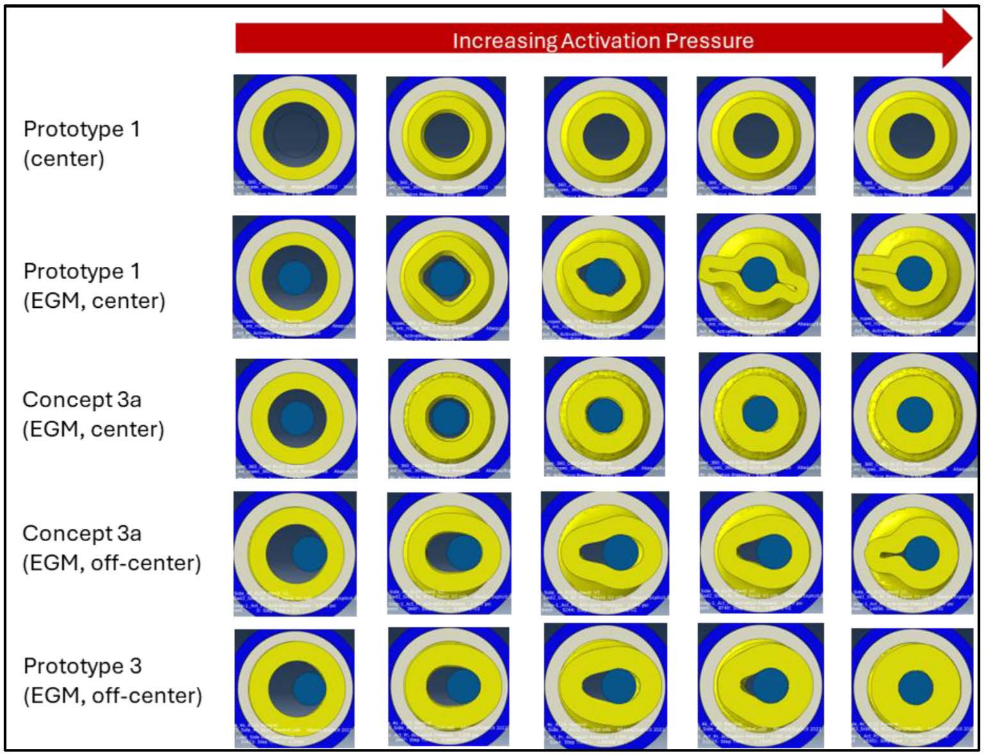

Figure 3—Closure Shape Study (Top View of Mid-Length Cross-Section), Sleeve Prototype Progression with Exaggerated Geometry Model and Off-Center Drill Pipe

The RCD sealing sleeve is activated by applying hydraulic pressure directly to the outer diameter of the sleeve, constricting it to close onto the drill string. The external pressure collapse factor is modified from API TR 5C3 pipe calculations (see Appendix A), utilized to help evaluate design concepts at the beginning of Phase 1 with only a few test data points (API TR 5C3, 2019). Another comparative metric developed is the ratio between the stiffness (area moment of inertia, to capture sleeve geometry) to the sleeve's distance travelled from the relaxed state to engagement on the minimum diameter of the drill pipe. Future work may include refinement of weighting factors for more accurate comparisons.

The shape factor study plot in Figure 4 includes both test and analytical data points (with the majority from analysis due to limited test data), highlighting the importance of sleeve stiffness and distance travelled to closure. The data set includes both on-center and off-center drill pipe (with distance travelled adjusted accordingly). The lower the stiffness (collapse factor) and larger the distance travelled to closure, the more likely a fold will develop. This study was used to prioritize sleeve prototype concepts through FE analysis, enabling the elimination of the fold during subsequent testing.

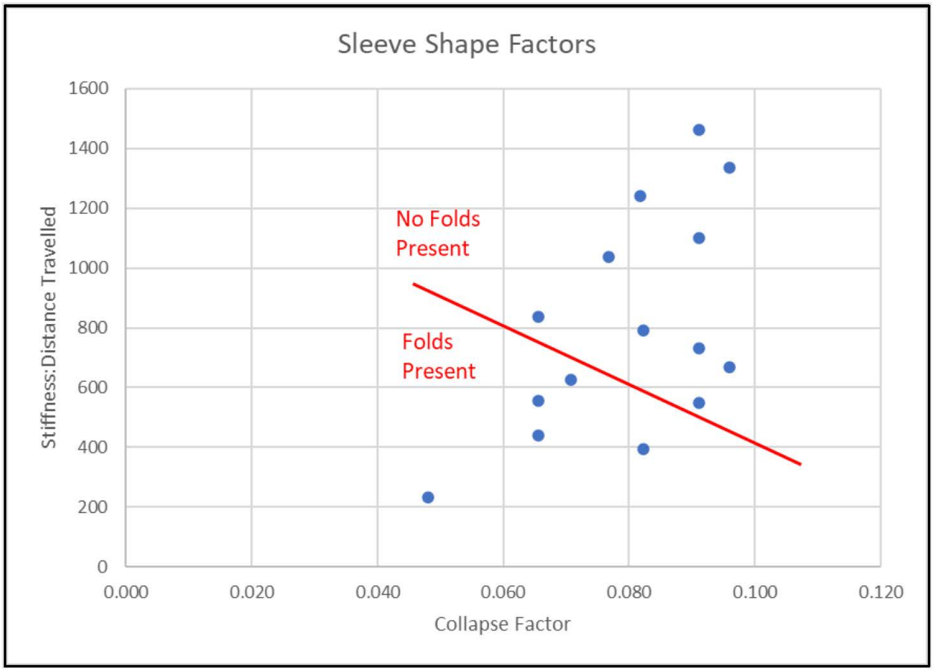

Figure 4—Sleeve Shape Factor Study, Fold Occurrence as a Function of Sleeve Stiffness and Distance Travelled to Close on Drill Pipe

Stripping Analysis on Sleeve. The second design challenge encountered during Phase 1 sleeve development was the need to minimize sleeve strains to extend stripping life. Stripping force breakout tests were used to develop the coefficient of friction between the sleeve and drill pipe. This allowed for proper hyperelastic FE modelling to predict strain hot spots during TJ passage and estimate the axial loads that must be carried by the bearing system. Once the fold was eliminated in Prototype 3, the first attempt of the Stripping Pressure Rating Test demonstrated that the failure location, matching the highest strain location in the design, needed to be addressed to increase stripping life.

The root cause of the failure was two-fold: high strains in the weakest area of the design, amplified by excessive heat generation in the conservative reciprocation test set-up for the API 16RCD test. In addition to optimization of the test set-up, the maximum strains during stripping were re-evaluated at a lower strain threshold to account for the elevated temperatures and refined design targets.

Figure 5 plots sleeve elastomer tensile strength at ambient and elevated temperatures. Figure 6 compares the maximum strain section in the Prototype 3 sleeve during stripping with the original strain filter to the reduced strain filter to account for elevated temperatures from friction of the TJ passage.

The Prototype 4 sleeve design included new geometry and sleeve to rotor interface optimization to reduce strains in the failure area, even at elevated temperatures, as shown in Figure 7. The maximum strain was reduced by 47%. Like the transition from the first two prototypes to Prototype 3, stripping tests of Prototype 4 resulted in a significant increase in performance, further cementing the accuracy of the FE model and design targets.

Figure 5—Sleeve Elastomer Stress-Strain Curves, Strength Reduction at Elevated Temperatures

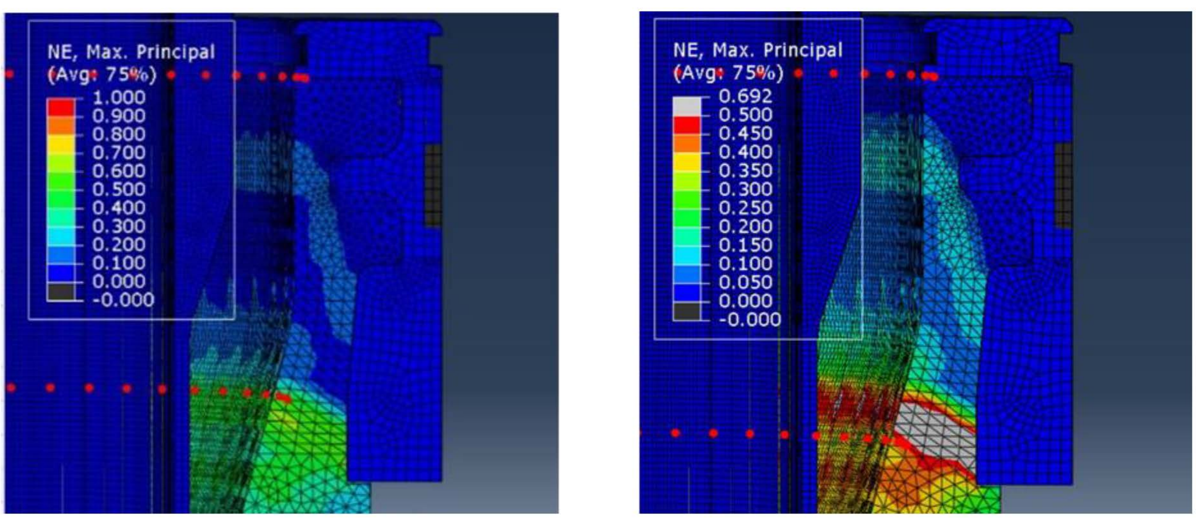

Figure 6—Sleeve Elastomer Max Principal Strain of Prototype 3 During Stripping, Original (Left) and Reduced Strain Filter (Right) to Account for Generated Heat, Shown on the Undeformed Shape

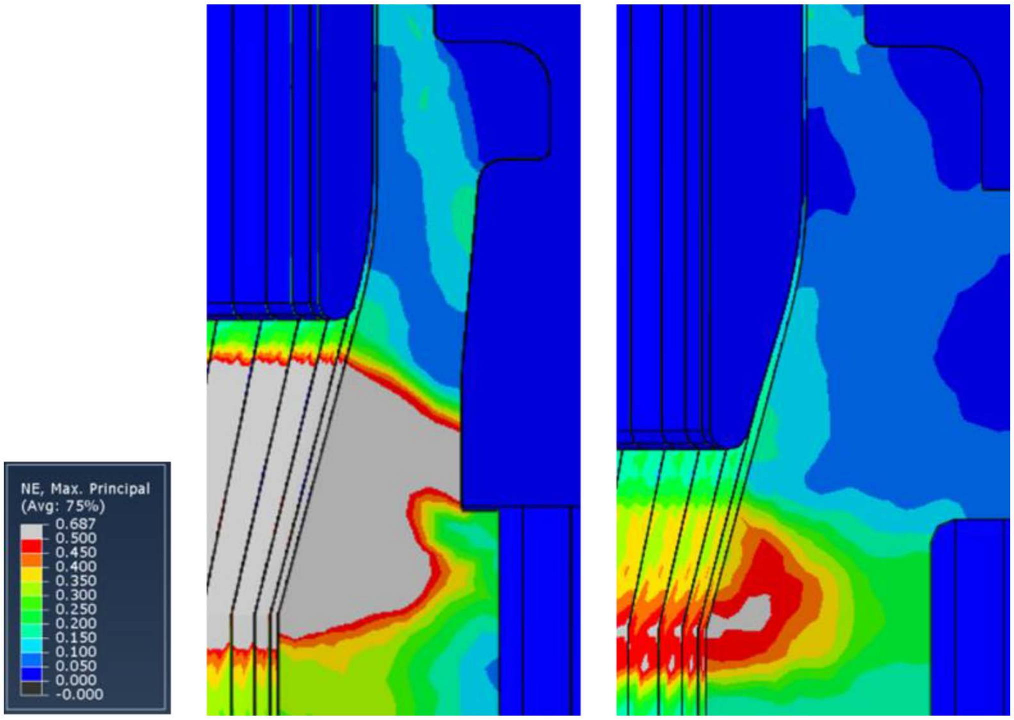

Figure 7—Sleeve Elastomer Max Principal Strain Reduction with Reduced Strain Filters at Prototype 3 Failure Location (Left) and Prototype 4 (Right), Shown on the Undeformed Shape

Sleeve Test Setup

A modular test fixture was designed for Phase 1 sleeve testing, incorporating features to allow for sleeve design iterations as needed. The fixture allows for sleeve length and end termination variations and can be used for static pressure tests (ambient and low/high temperatures) as well as stripping, as shown in Figure 8.

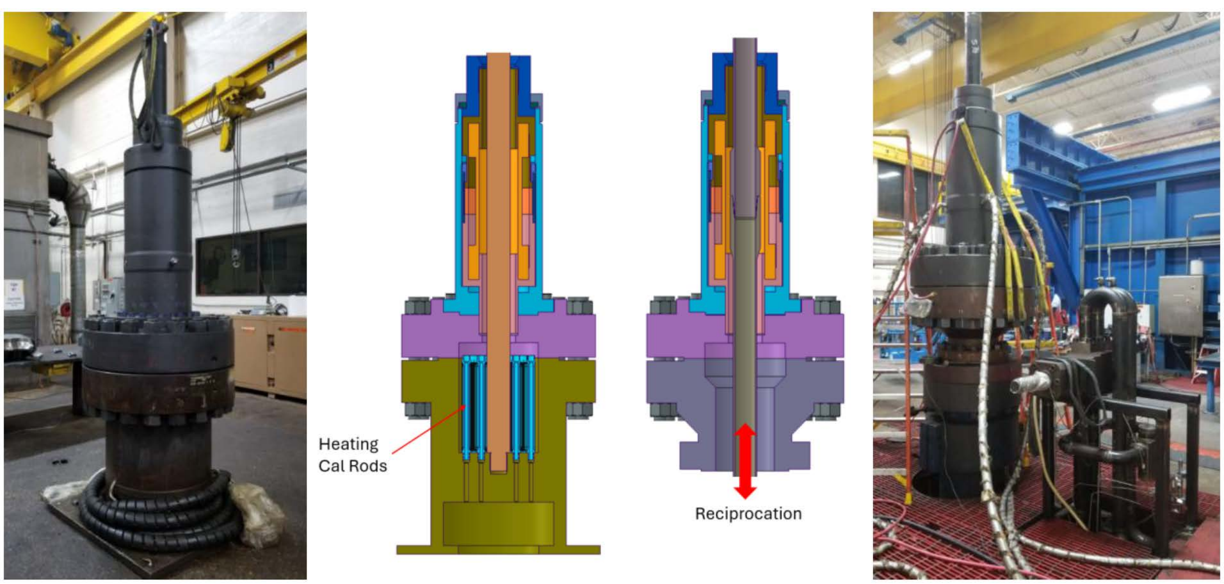

Figure 8—Sleeve Test Setup for Static and Temperature Tests (Left) and Stripping Tests (Right).

Bearing Down-Selection

After evaluating potential bearing solutions for the required usage cases, the system was configured to include axial/radial bearings and rotary seals. The system design was defined as: a system that satisfies both the axial and radial load requirements and meets rotational, operating temperatures, API endurance limits, and break-out torque targets. For an active/rotating design, the break-out torque of the bearing system is critical as it must be less than the torque generated by the sleeve and the drill pipe to prevent sleeve wear. The need for a robust design capable of operating in harsh conditions for an entire hole section necessitated the evaluation of alternate bearing and rotary seal designs that have not been traditionally used in the RCD space.

Evaluation of potential bearing systems begins with the fundamentals of a rotating system. To evaluate the operating window for any rotating equipment, a pressure velocity (PV) curve is used to understand design limitations. Similarly, rotary seal contact pressure is defined at the interface to the mating seal surface and the pressure differential applied across the seals. The PV value is the combination of the contact pressure between the bearing faces under operational loads and the application of the speed of rotation (calculated as linear speed) and is defined by the following equation (What, 2023):

$$PV = Circumference * Speed * Pressure \quad \left[\frac{psi * ft}{min}\right]$$

The PV limit, a function of the material properties and geometry, is the maximum operational threshold of a component. Once the PV limit is exceeded, the equipment may excessively wear, which can lead to premature failure. Thus, the three main design tenants for the bearing system were refined to be: (1) the system must isolate wellbore fluids from the sleeve activation chamber, (2) equipment must remain below their PV limit under operational loads, and (3) rotary seals must be able to withstand the pressure differential at expected RPM under load conditions.

Two main design efforts for the bearing system identified: (1) an alternative bearing concept beyond traditional roller bearings and (2) robust rotary seals that can tolerate increased extrusion gaps from drill pipe side loads. To evaluate the type of bearings needed for this application, key technical requirements were established to baseline performance requirements and detailed load path diagrams were constructed to understand the load capacities required for each design. The load path diagrams in Figure 9 define the interaction of the loads and forces within the RCD that are generated from sleeve activation, wellbore sealing, and the reciprocation and rotation of the drill pipe.

The bearing prototypes evaluated for Phase 1 were sized using the forces calculated from the load path diagram shown in Figure 9. A symmetric design of the sleeve and bearing system accounts for pressure from above or below the sleeve and the frictional load from drill string movement in both directions. When the direction of the drill string matches the direction of the wellbore pressure end load, the axial load on bearings increases in the direction of drill string movement and unloads the bearings on the opposite side.

Since roller bearings are a known and proven technology, they were considered the "baseline" technology in Phase 1. As an alternative to a roller bearing in this system, two concepts were evaluated as shown in Figure 10: Ceramic (Silicon Carbide) bearings and Polycrystalline Diamond (PCD) bearings.

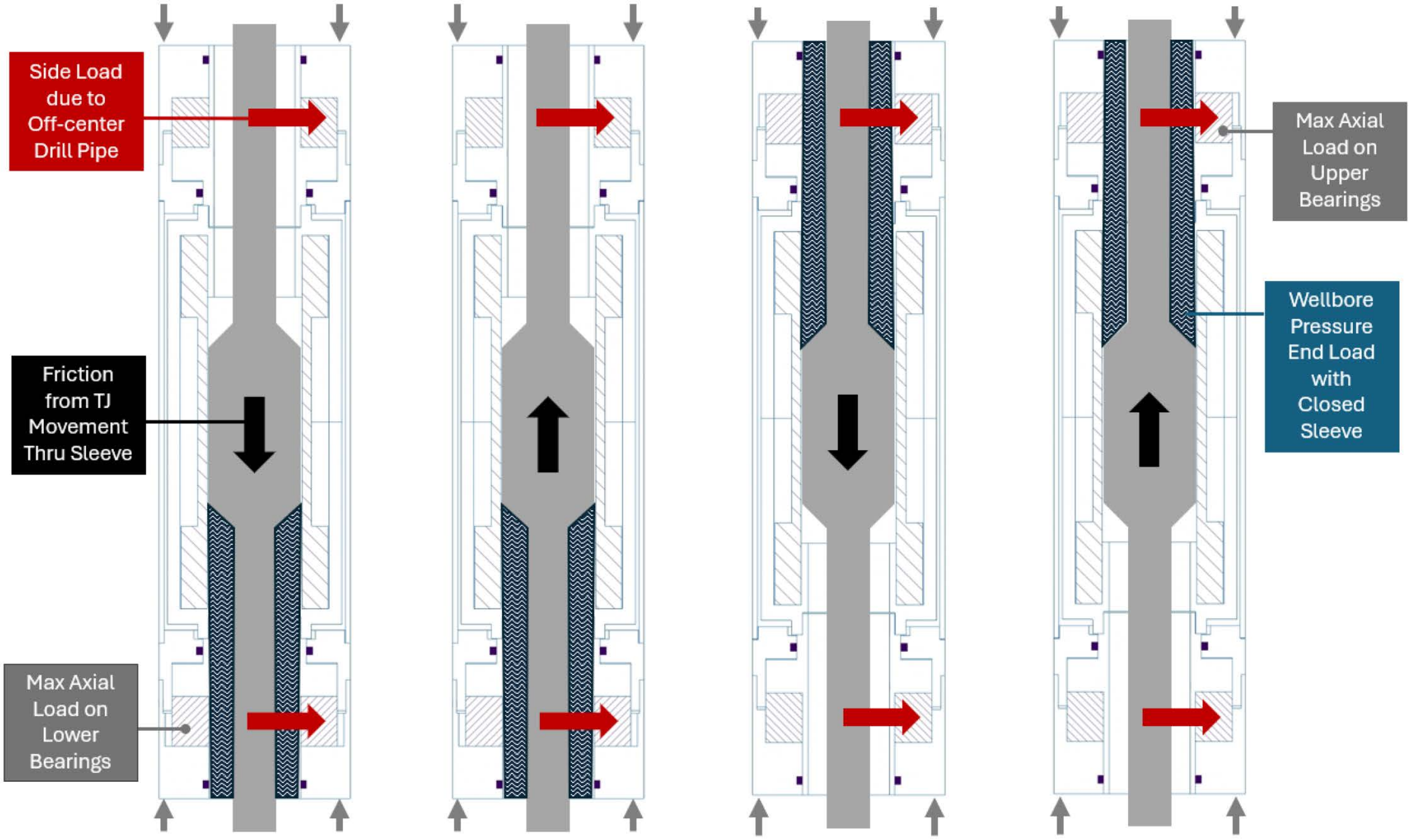

Figure 9—Load Path Diagram of RCD.

Figure 10—Overview of Alternative Bearing Concepts.

Figure 11 compares the PCD bearings to a Silicon Carbide ("SiC") sliding bearing and a Silicon Carbide sliding bearing with integral seals ("SiC Int"). The PV limit curves for each of these bearings are plotted against the expected maximum PV value that each of the bearing concepts would experience during operations. This figure illustrates that the two Silicon Carbide bearings would exceed the acceptable PV operating window, while the PCD bearing shows to be within operating limits. Thus, it can be concluded that the Silicon Carbide bearings could experience premature failure in operation and the PCD bearings would operate within their capacity window.

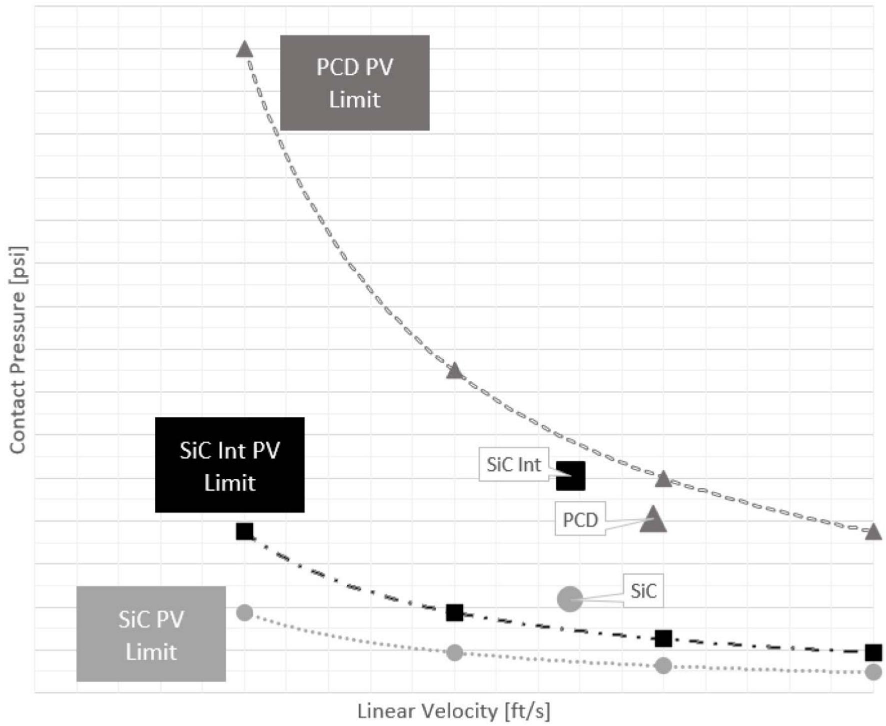

Figure 11—PV Curves for Different Bearing Types [Cooley,2012] [Petro,2016] [Park,2011].

From further down-selection, the PCD bearings were chosen as the alternative design path to evaluate against known roller bearing technology. PCD bearings utilize small diamond sliding pads with built-in channels between the pads to allow cooling flow to remove generated heat and maintain temperatures within the PV limits of the material. PCD bearings have documented use in Oil and Gas drilling environments, having been successfully utilized in down-hole applications including drilling motors, turbines, and steering devices. They were selected for potential use in the active/rotating RCD due to their extended operational life when compared to other solutions. The critical drivers during this down-selection process included material properties (PV limitations), minimization of breakout & running torque, maximizing fracture toughness for impact loading, misalignment tolerance, and lubrication complexity. When comparing the three bearing concepts, there were five main differentiators:

1. Material Properties: PCD bearings possess better thermal and strength properties compared to the other bearings considered.
2. Breakout & Running Torque: PCD bearings have a lower COF than the Silicon Carbide bearings resulting in lower breakout and running torques. Roller bearings have an even lower breakout and running torque.
3. Bearing Cooling: As the bearings rotate, heat is generated from friction under load. The PCD bearings have built in cooling channels between the pads which aid in this cooling. Silicon carbide bearings require active cooling while roller bearings do not always require cooling.
4. PV Limitations: PCD bearings have a significantly larger PV operational window than the Silicon Carbide bearings. PV limitations of other types of bearings contributes to concerns about premature failure in this application.
5. Debris Tolerance: PCD bearings can withstand more debris ingress when compared to alternative bearing solutions.

Bearing Test Setup

After dedicated rotary seal testing to optimize performance, the entire Bearing System (bearings and rotary seals) were tested in a dedicated fixture to simulate system performance and apply axial and radial loads under rotation and pressure. The setup for the bearing test fixture is shown in Figure 12.

The bearing system test program was designed to determine the following:

- Required cooling media and flow rate for optimal performance
- Radial bearing performance under side loads
- Rotary seal performance, including under side loads (extrusion gaps)
- System breakout and running torque
- Bearing preload optimization

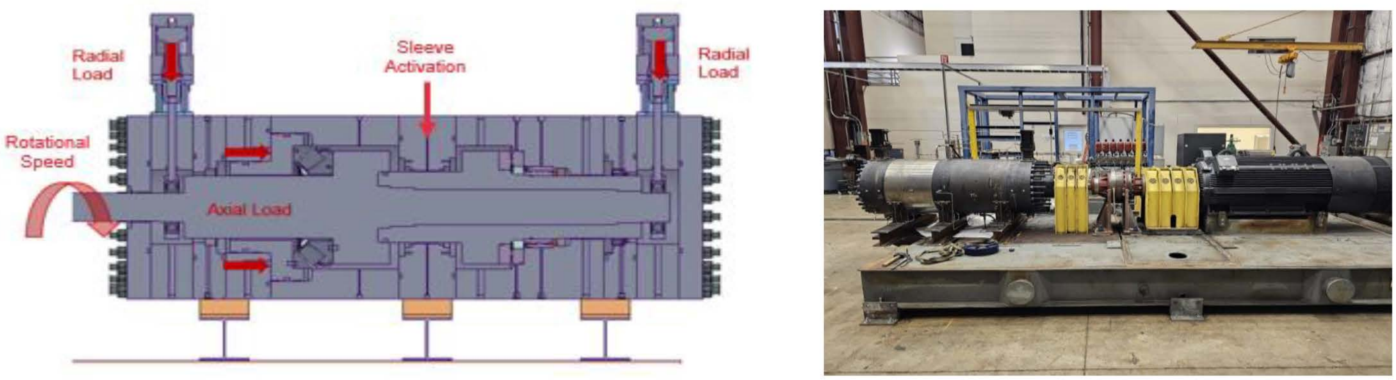

Figure 12—Bearing Test Fixture Layout and Load Diagram.

Rotary seal testing was conducted to ensure the following:

- Seal performance under continuous rotation under worst-case conditions
- Seals can meet the PV limits required by speed and pressures of this RCD
- Seal temperature and torque characteristics over time

Presentation of Data and Results

The Prototype 4 sleeve successfully completed all planned Phase 1 qualification tests, as summarized in Table 4. Leveraging hyperelastic FE analysis, this was accomplished by running over 113 FE models and testing only 4 prototype designs.

Stripping pressure rating cycles increased by more than ten times to exceed the API threshold of 400 TJ passages compared to Prototype 3. Additional design optimization opportunities have already been identified to further reduce strain to extend stripping life to ensure the sleeve can last an entire hole section with a sufficient safety margin.

Table 4—API 16RCD Sealing Sleeve Qualification Test Summary, Phase 1

API 16RCD Test	Requirement	Actual Test Parameters	Mandrel
Sealing Characteristics	(reportable)	(reportable)	5-7/8" Mandrel
Static Pressure Rating	1015 psi	1015 psi	5-7/8" Mandrel
Fatigue Test	1015 psi 52 pressure cycles	1015 psi Stopped after 54 cycles	5-7/8" Mandrel
Low Temperature Test	1015 psi 30°F	1015 psi 30°F	5-7/8" Mandrel
High Temperature Test	1015 psi 140°F	1015 psi 140°F	5-7/8" Mandrel
Stripping Pressure Rating (extends to Stripping Life Test)	1015 psi 400 TJ's minimum	1015 psi 452 TJ's 956 ft/hr equivalent	5-7/8" Drill Pipe with hard banding
Dynamic Pressure	1015 psi 200 RPM 100 hours under rotation	*(PHASE 2 SCOPE, TEST WITH BEARINGS)*	

Testing was conducted on rotary seals under loading conditions with the focus of extended life beyond API endurance requirements (100 hours of rotation at a minimum). Testing demonstrated that these seals could operate past API life thresholds, handle elevated PV with a pressure differential exceeding 1500 psi, and operate in expected control fluid for system use. A summary of the testing conducted on the activation chamber and wellbore wetted rotary seals is shown in Table 5. In addition, 52 fatigue pressure cycles were completed on the activation chamber seals, mimicking the API 16RCD Fatigue Test.

Table 5—Rotary Seal Test Results.

Rotary Seal	Pressure Differential [psi]	Test Duration [hrs]	Test Media
Activation	1640	340	Hydraulic Oil
Activation	2000	460	Hydraulic Oil
Wellbore	1500	340	Mud

Conclusions

The Phase 1 development program confirmed that breaking down the complex system of an RCD into component-level designs to be tested independently allowed for agile prototype iterations and risk reduction.

Thus, it is important to identify and isolate high-risk components in a complex assembly to rapidly iterate through design and testing. Taking this phased approach allowed for the use of modular test fixturing and iterative testing which accelerated the development process.

The sealing sleeve and bearing system in this active/rotating RCD have performed well in testing and analysis during Phase 1 and will be considered for API 16RCD qualification testing next, along with other enhancements.

Key learnings and recommendations from Phase 1 of this RCD project include:

- The importance of developing a representative hyperelastic FE model for the design of complex seal systems, allowing for test correlation to enable iterations through analysis, reducing the number of physical prototypes.
- Use of FE models to predict sealing element closure shapes with EGM techniques, resolving challenges encountered in early prototype testing.
- Understanding of PV limits and heat generation is critical to rotating equipment selection.
- Modular test fixtures designed to replicate known operating characteristics (i.e., side load) help mitigate risk and shorten the path to product qualification.

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Appendix

Appendix A: Modified API TR 5C3 Collapse (API TR 5C3, 2019)

API TR 5C3 Yield Strength Collapse Pressure Equation:

$$p_{Yp} = 2f_{ymn}\left[\left(\frac{D}{t}\right) - 1\right] \Big/ \left[\left(\frac{D}{t}\right)^2\right] \quad (1)$$

where
D is the specified pipe outside diameter
f_{ymn} is the specified minimum yield strength
p_{Yp} is the pressure for yield strength collapse
t is the specified pipe wall thickness

Metallic pipe equations were modified to remove yield strength, focusing on the geometry aspect ratios such that:

$$p_{sleeve} = \left[\left(\frac{D}{t}\right) - 1\right] \Big/ \left[\left(\frac{D}{t}\right)^2\right] \quad (2)$$