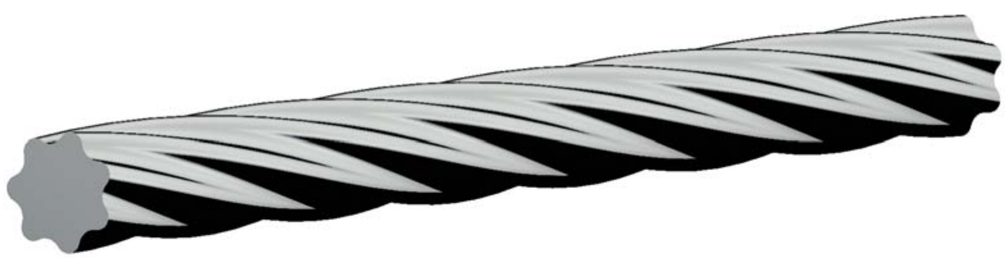
Chapter E3

Using Kalsi Seals in mud motors



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1. Introduction

Getting to the point without wasting time

This chapter is a general guide to handbook material related to designing mud motor sealed bearing assemblies. It is intended to help you get to the point without covering unrelated material. It uses schematics to orient the reader, and then references the handbook material that is relevant to specific design topics.

Why this chapter became necessary

Kalsi Seals® are widely used as mud motor seals in oilfield drilling operations. When this handbook was first introduced in the early 1990s, it focused primarily on mud motor sealing. Over the years, the scope has increased, and the handbook now provides guidance for a wide variety of equipment.

The sheer volume of information can be overwhelming. Even a seasoned engineer may be unable to *efficiently* find the material that is most relevant to mud motor design. For example, the chapter on lubricant reservoirs covers everything from simple gas charged reservoirs to complex circulating systems with computer-controlled valves.

2. Overview of what a mud motor is and does

An oilfield mud motor (Figure 1) includes a bearing assembly and a power section. The bearing assembly incorporates a shaft that is rotationally driven by the power section. The shaft positions and rotates the drill bit and loads it against the formation. Bearings guide and locate the shaft, and transfer drilling forces between the shaft and the drillstring. Rotary seals retain the bearing lubricant and exclude the drilling fluid.

Typically, the power section is a Moineau-type progressive cavity motor[1] that utilizes the flow of drilling fluid within the wellbore to produce rotation. A universal joint (not shown) connects the rotor of the power section to the shaft of the bearing assembly.

An optional bend angle between the power section and the bearing assembly provides the capacity to selectively drill straight or curved holes. Shaft rotation combined with drillstring rotation produces a straight, oversized hole. Shaft rotation in the absence of drillstring rotation produces a curve in the direction of the bend.

Severe service conditions dictate a systems approach to seal implementation

Mud motor assemblies are exposed to extreme operating conditions that challenge every component. The assembly is exposed to abrasive, high velocity, extreme pressure, elevated temperature drilling fluid that may contain chemicals that are unfriendly to

1 For those with historical interests, see Moineau's U.S. patent 2,028,407.

various materials. The pressure of the drilling fluid drops significantly as it passes from the wellbore to the annulus through the jets of the drill bit. Unless a flow restrictor/bypass arrangement is used, this exposes one of the rotary seals to high differential pressure. The assembly is also subjected to extreme radial, axial, and torsional loads, deflections, and vibration[2] related to factors such as weight on bit, bend angle, and drillstring stick-slip[3] and elasticity.

Because of these severe service conditions, a systems approach is required to successfully implement mud motor seals. For example, the design of the shaft and the locations of the radial bearings relative to the seals are critical to seal performance. This chapter is intended to serve as a guide to the handbook information that is most relevant to designing a mud motor sealed bearing assembly.

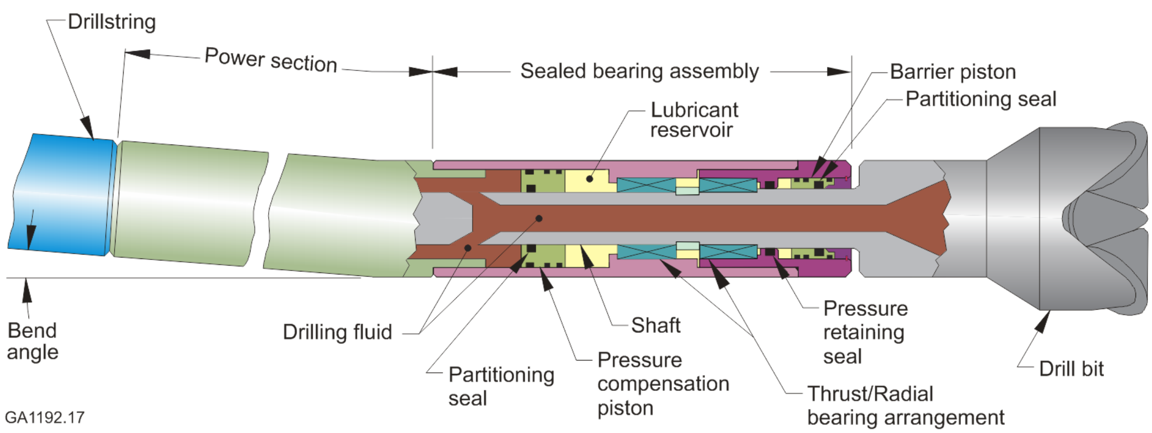

Figure 1

Schematic of a typical oilfield mud motor

3. Introduction to Figures 2 to 5

Figures 2 and 3 are schematic representations of conventional mud motor sealed bearing assemblies. Figures 4 and 5 are schematic representations of untested concepts that *may* have potential merit. They are included for consideration by organizations that are willing to take risks and try something new. The concepts are directed at reducing the extrusion

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2 For information on drillstring vibration, see SPE papers 14330, 15560, 15561, 15563, 15564, 145910, 147747, SPE/IADC paper no. 18652, and Erik Skaugen, et al., “**Experimental and Theoretical Studies of Vibrations in Drill Strings Incorporating Shock Absorbers**” (12th World Petroleum Congress, 1987).

3 Sticking causes the drillstring to wind up and store energy. In the slip phase, the stored energy is released resulting in rapid spinning and violent vibration. For information on stick-slip, see SPE 145910, "**Drill Pipe Measurements Provide Valuable Insight into Drill String Dysfunctions**”.

gap clearance at the pressure retaining seal, to accommodate higher temperature or higher differential pressure, while minimizing overall length.

4. Features common to Figures 2 to 5

Introduction

This section describes design elements common to Figures 2 to 5. Please note that most details, such as shaft diameter and journal bearing length, are not to scale.

Shaft strength and stiffness requirements dictate the use of journal bearings

In the early days of sealed bearing assemblies, rolling element bearings were the radial bearing of choice. The radial thickness of such bearings is significant. The space taken up by the bearings meant that the shafts in early sealed bearing mud motor designs were slender. This made the shafts prone to high lateral deflection, and fatigue failure.

To maximize shaft strength, and minimize shaft deflection, the use of relatively thin journal bearings is necessary.[4] Journal bearings allow the maximum shaft size to be realized within a given set of envelope constraints.

Kalsi Engineering was an early pioneer in the use of journal bearings in mud motor sealed bearing assemblies. We are not in the business of designing and manufacturing mud motors, however we have built and tested them for seal research purposes. Although our research tools used custom machined journal bearings, published literature on sealed bearing assemblies suggests that DU bushings are sometimes used. We elected to use custom machined journal bearings, to minimize bearing clearance. Radial bearing selection is your engineering decision, and we strongly recommend that you consult with experienced bearing manufacturers.

When implementing journal bearings in a mud motor, be sure to avoid the threaded connection related over constraint and binding issues that are described in conjunction with Figures 6 and 7 of Chapter D20.

Thrust bearing implementation

Thrust bearings are not illustrated in Figures 2 to 5 because there are a variety of ways to implement them in a mud motor sealed bearing assembly. At least two companies produce custom, high capacity roller thrust bearings that are specifically designed for mud motor sealed bearing assemblies.

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4 For examples of the use of journal bearings in mud motor designs, see U.S. Patents 5,727,641 and 6,416,225.

One convenient way to incorporate oppositely facing thrust shoulders on the shaft is to use a threaded connection. Another way is to use a split clamp arrangement that engages a shaft groove. The published literature also shows the use of a threaded ring.

If a split clamp is used, it should be carefully designed to provide 360° of support to the thrust bearing races. The split clamp should be rotationally driven by the shaft with little or no rotational free play, to avoid the slip/stick related angular acceleration/de-acceleration issues described in Section 9 of Chapter D20. As with every shaft detail, the groove that receives the split clamp must be designed with fatigue failure resistance in mind.

Before finalizing your mud motor design, study and avoid the thrust bearing related design mistakes presented in Chapter D20. Be sure to provide adequate support to the thrust bearing races, to avoid the problem described in Section 2 of Chapter D20. Take care in machining threaded connection shoulders to avoid the angular misalignment issue described in conjunction with Figure 8 of Chapter D20. (The potential for thrust bearing misalignment due to housing flexure is apparently mitigated to a certain extent by corresponding shaft flexure.) Also avoid the spacer and lubricant viscosity related issues described by Sections 10 and 14 of Chapter D20.

The bearing lubricant

The bearings are flooded with a liquid, oil-type lubricant. The lubricant must retain sufficient viscosity at the upper end of the intended operating temperature range to adequately lubricate the heavily loaded bearings. The selected bearing lubricant also must be chemically compatible with the material chosen for the rotary seals, and viscous enough to lubricate them. A general discussion of lubricants is provided in Chapter D12.

Gland design for Kalsi Seals

Gland nomenclature and design details are described in Chapter D5. Chapter D7 covers critical extrusion gap design considerations.

The lubricant reservoir

To avoid the pressure locking issues described by Sections 1 and 2 of Chapter D13, the pressure of the bearing lubricant is balanced to the pressure of the drilling fluid that is located immediately above the bearing region. This pressure balancing is typically accomplished by an axially movable, annular pressure compensation piston[5] which partitions the drilling fluid from the bearing lubricant.

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5 A "compensation piston" is sometimes referred to as a "compensating piston". Pressure compensation pistons were first described for a sealed bearing mud motor design in U.S. Patent 3,730,284.

The upward stroke of the piston accommodates thermal expansion of the lubricant. The downward stroke of the piston provides a lubricant reservoir to accommodate the hydrodynamic pumping related leakage of the Kalsi Seals that are served by the reservoir. The volume of the reservoir should be sized based on seal leakage at the lowest anticipated pressure drop across the pressure retaining seal during normal operation, and at the lowest anticipated operating temperature. Information on the hydrodynamic pumping related leak rate of Kalsi Seals is provided in the *"Catalog & Technical Data"* portion of the handbook.

The design of pressure compensation pistons is covered in detail by Chapter D14. The design of the recommended fit between the piston and the mating running surface is described in Chapter D15.

The rotary seal that partitions the bearing lubricant from the drilling fluid in the wellbore is typically mounted in a shaft guided pressure compensation piston, as shown by Figures 1 to 5. Other arrangements are possible.

Selecting partitioning seals

For operating temperatures below 300°F (148.9°C), a -11 HNBR Axially Constrained Kalsi Seal™ (Chapter C4) is ordinarily recommended for the seal that partitions the bearing lubricant from the wellbore fluid. If drilling fluids that are not compatible with HNBR are anticipated, consider adding a barrier seal (Chapter D10) that is made from a chemically compatible sealing material. For example, a spring-loaded FEPM lip seal may be appropriate for formate drilling fluids. Extensive information on chemical compatibility is provided by various third-party online databases. The subject is covered to a more modest degree by the *"Materials"* portion of this handbook. For operating temperatures greater than 300°F (148.9°C), see Section 7 of this chapter.

The pressure retaining seal

The fixed location, pressure retaining seal is exposed to the pressure difference between the bearing lubricant and the well annulus. The pressure difference can range from high to low, depending on whether a flow restrictor/bypass arrangement is incorporated above the pressure compensation piston.

The most common material we sell for the fixed location pressure retaining seal is our -11 HNBR material, which has been tested at temperatures up to 300°F (148.9°C).

Our Wide Footprint™, Hybrid, and Enhanced Lubrication seals™ are typically the best seal candidates. Axially Constrained seals may be appropriate when a flow restrictor/bypass arrangement is used. Seal selection depends on factors such as reservoir size, lubricant viscosity, differential pressure, operating temperature, available radial space, and whether the seal is exposed to the drilling fluid in the well annulus. The available types of Kalsi Seals are described in the *"Catalog & Technical Data"* portion of the handbook. In general, wider sealing lips provide better pressure related extrusion resistance and provide more sacrificial material to accommodate abrasive wear.

The seal running track and installation path

The seal running tracks should be ground and polished tungsten carbide. See Chapter D2 for coating and surface finish specifics. Seal installation path considerations, including installation chamfers, are covered in Chapter D3.

5. Conventional sealed bearing mud motor designs

Typical mud motor sealed bearing assemblies follow one of two basic design approaches. These approaches are represented schematically by Figures 2 and 3.

The mud motor bearing assembly shown schematically by Figure 2

The Figure 2 approach incorporates a barrier piston outboard of the pressure retaining seal.[6] The barrier piston is designed to limit shaft deflection, so the extrusion gap of the pressure retaining seal can be sized small enough to be compatible with high differential pressure. For general piston design and seal selection considerations, see the compensation piston related material above.

Since the barrier piston limits shaft deflection, it receives heavy side loads. Plan your journal bearing area and barrier piston anti-rotation measures accordingly.

The mud motor bearing assembly shown schematically by Figure 3

The Figure 3 approach incorporates a flow restrictor and a flow bypass orifice[7] to minimize the pressure differential acting across the pressure retaining seal. This allows the extrusion gap of the pressure retaining seal to be large enough to accommodate the lateral shaft deflection that occurs in the absence of a deflection limiting barrier piston. Consider using rotary seals with larger radial cross-sections that have more initial compression to better accommodate lateral shaft motion.

One disadvantage of the Figure 3 approach is that the pressure retaining seal is exposed to swab and surge pressure,[8] which promotes abrasive invasion of the dynamic sealing interface. This can be mitigated by using a barrier seal (Chapter D10). Another

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6 For examples of sealed bearing mud motor designs that incorporate a barrier compensation piston, see U.S. Patents 5,150,972, 5,248,204, and 5,664,891.

7 For examples of mud motor designs that use flow restrictors with bypass orifices, see expired US Patents 3,857,655 and 3,894,818.

8 For a description of swab and surge pressure, see U.S. Patent 6,220,087.

disadvantage is that the fixed location pressure retaining seal is exposed to more lateral shaft motion, compared to the other approaches.

6. Unconventional concepts that may have merit

Introduction to the mud motor bearing concepts

Figures 4 and 5 show two untested sealed bearing mud motor design concepts for your consideration. These relatively compact concepts should be evaluated for fatigue resistance before they are tried.

Commonalities between the Figure 4 and 5 concepts

In the mud motor design concepts of Figures 4 and 5, the pressure retaining seal is located between two radial bearings. This proposed arrangement would allow the smallest possible extrusion gap clearance to be used for the pressure retaining seal, without risk of heavily loaded metal to metal contact at the extrusion gap.

The lower radial bearing is protected by a fixed location partitioning seal. The lubricant reservoir for the lower radial bearing and the fixed location partitioning seal is balanced to the annulus pressure by a sliding pressure compensation seal. The sliding distance should be minimized, to minimize tool length, and to minimize the potential for cocking of the sliding seal.

Comments specific to the Figure 4 concept

In Figure 4, the fixed location partitioning seal is a spring-loaded lip seal having an outer static lip that is axially longer than the inner dynamic lip. This arrangement allows the inner lip to vent lubricant in response to an increase in lubricant pressure. As a result, hydrodynamic pumping related leakage from the pressure retaining seal passes through the dynamic interface of the lip-type partitioning seal, providing interfacial lubrication. Annulus pressure is communicated to the lower bearing region by a cross-drilled hole.

Comments specific to the Figure 5 concept

In Figure 5, the lower radial bearing is also protected by a fixed location partitioning seal. The fixed location partitioning seal is a rotary seal that has a lower hydrodynamic pumping related leak rate, compared to the fixed location pressure retaining seal. The bearing lubricant between the partitioning seal and the pressure retaining seal is balanced to the annulus pressure by a lip seal that can slide axially for a short distance. One lip is axially longer than the other, so that the inner lip can vent the hydrodynamic pumping related leakage of the pressure retaining seal, to prevent a pressure buildup between the partitioning seal and the pressure retaining seal.

7. A combined approach for hot wells

The conventional mud motor bearing assembly is not optimal for hot wells

Figure 3 schematically represents the now-conventional sealed bearing mud motor design, where a fixed location pressure retaining seal is located below and outboard of the lowermost radial bearing, and a partitioning seal is mounted in a pressure compensation piston that is located above the bearings.

The elevated temperature of a hot well increases compression set and reduces the modulus and extrusion resistance of elastomers, making elastomer seals unsuitable for bridging large extrusion gaps when significant differential pressure is present. The Figure 3 mud motor design is not recommended for high temperature operation because it is incompatible with using a small extrusion gap clearance for the pressure retaining seal.

Higher temperature testing of fixed location pressure retaining seals

The high temperature tests we have performed to evaluate seals for mud motor fixed location pressure retaining service have been at modest differential pressure, based on the assumption that mud motors for hot wells will be used with flow restrictors. For example, PN 507-5-31 seals have been tested at 350°F (176.7°C) with an ISO 1,000 viscosity grade lubricant and a differential pressure of 250 psi using a 0.010” radial extrusion gap clearance. For summaries of elevated temperature seal testing, see the materials section of this handbook, or contact Kalsi Engineering. Operation with -31 seals at temperatures above 300°F (148.9°C) typically requires wider seal grooves to accommodate elastomer thermal expansion. For example, a Wide Footprint Seal™ that ordinarily uses a 0.289” wide groove should have a 0.320” wide groove. Contact Kalsi Engineering for additional information.

Mud motor design recommendations for hot wells

We believe that mud motor sealed bearing assemblies for hot wells will eventually evolve to designs that locate the fixed location pressure retaining seal in the middle of the tool, where the seal is isolated from extreme shaft deflection. Such an arrangement will permit the use of a relatively small extrusion gap clearance, to help to offset the reduced seal extrusion resistance that occurs at higher temperatures. There may be enough room to implement a floating backup ring assembly (Chapter D17) near the thrust bearings, providing the high pressure rotary seal with the smallest possible extrusion gap clearance.

Figures 2, 4, and 5 are *schematic* representations of such tools. The basic theme of these concepts is that the fixed location pressure retaining seal is located toward the middle of the tool, where shaft deflection is considerably reduced, compared to the shaft deflection at the fixed location seal in the conventional Figure 3-type mud motor sealed bearing assembly. These concepts require that the sealed region above the fixed location seal be pressure compensated to the wellbore pressure, and require that the sealed region below the fixed location seal be pressure compensated to the annulus pressure. The best way to pressure balance this lower region ultimately will be determined by mud motor design engineers, field experience, and market forces. Figures 2, 4, and 5 merely provide ideas, as a possible starting point for your own engineering innovation.

The pressure compensation piston and barrier piston seals

When designing pressure compensation pistons and barrier pistons for temperatures greater than 300°F (148.9°C), use an axially spring loaded (Chapter 9) Kalsi Seal™ made from a material that is compatible with the anticipated operating temperature.

8. Review of seal related drawing details

We are typically willing to review customer engineering drawings to help to ensure that the seal related information in the handbook is being interpreted correctly. Such reviews are performed to help you achieve the quickest path to success. Such reviews are not a substitute for your organization's checking and review processes, but they can be a useful supplement.

When submitting such drawings for review, please provide a completed application questionnaire, so that we know the intended operating conditions of your tool. For a copy of the questionnaire, see Appendix 4.

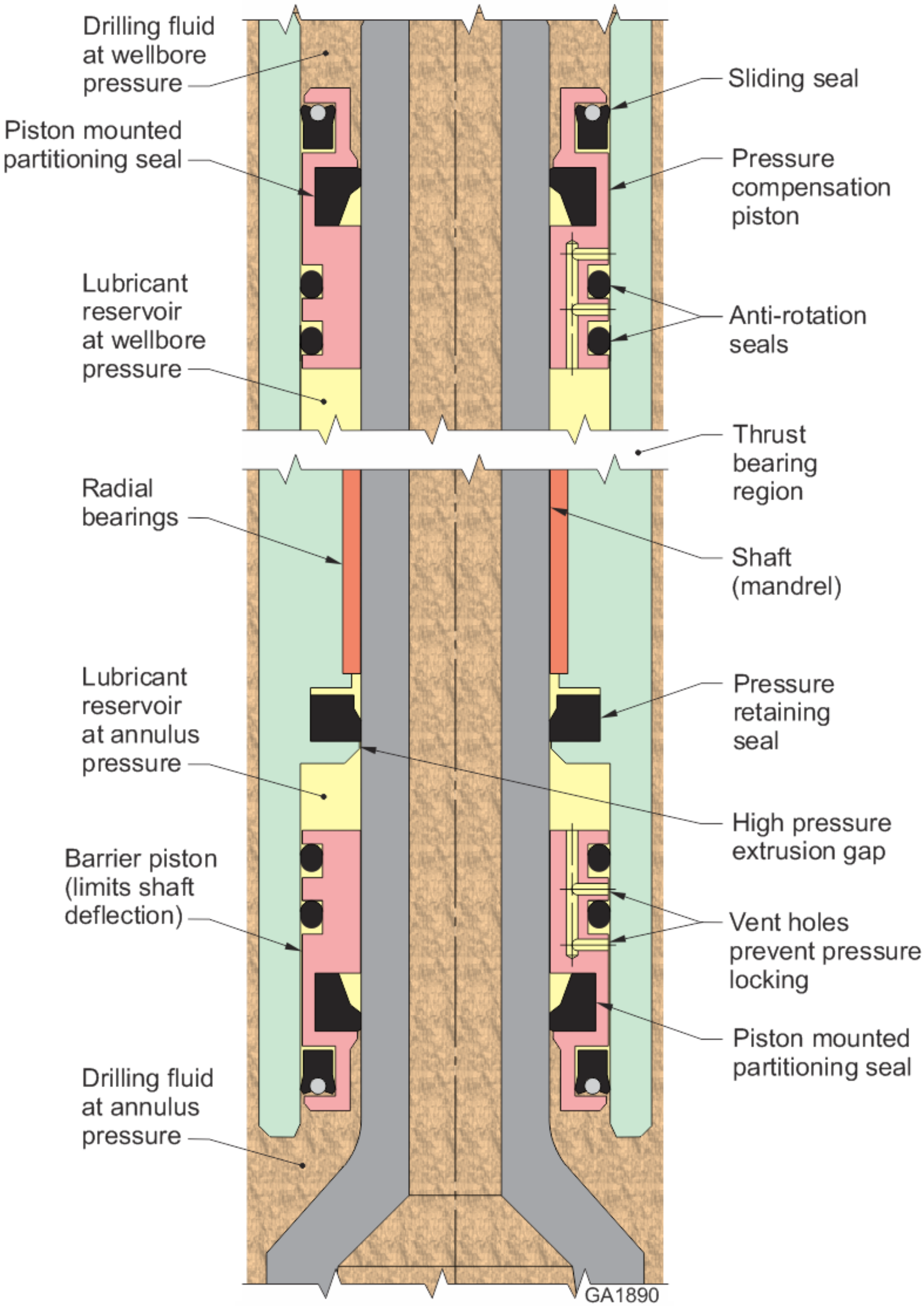

Figure 2

Schematic of a sealed bearing assembly with a deflection limiting barrier piston

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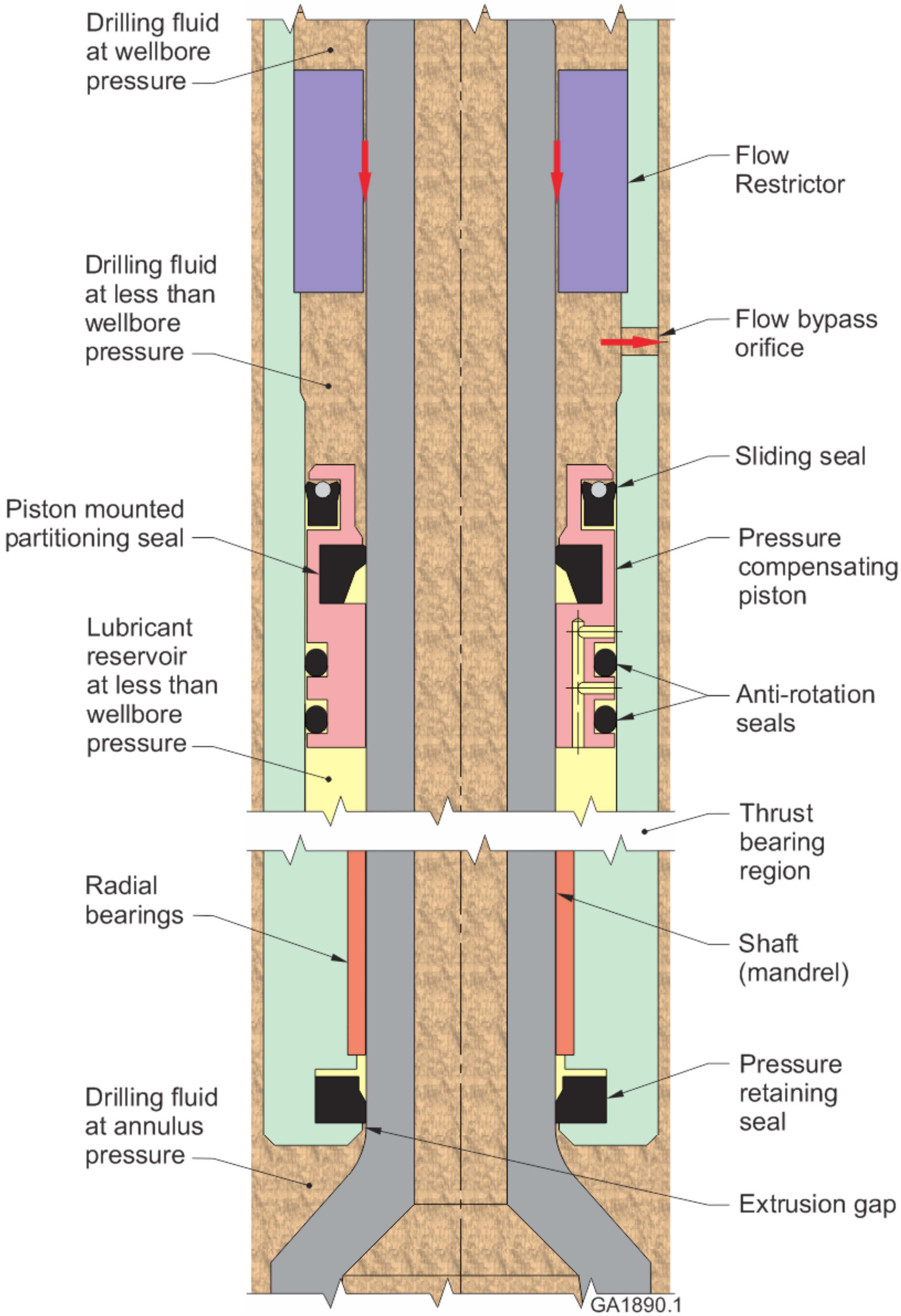

Figure 3

Schematic of a sealed bearing assembly with a flow restrictor and bypass orifice

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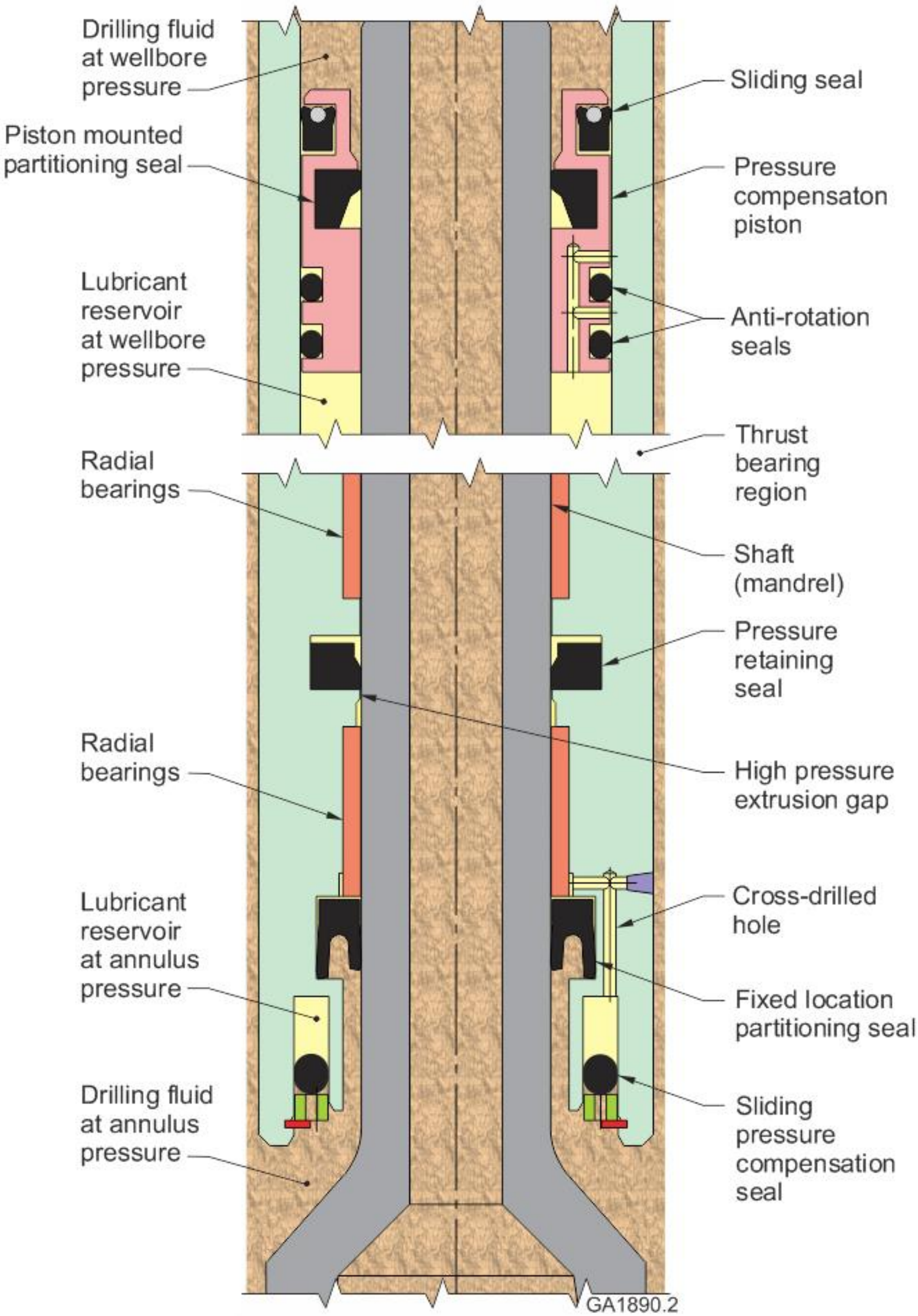

Figure 4

Concept: A bearing and lip seal outboard of the pressure retaining seal

(U.S. Patent 7,798,496, Kalsi Engineering, Inc.)

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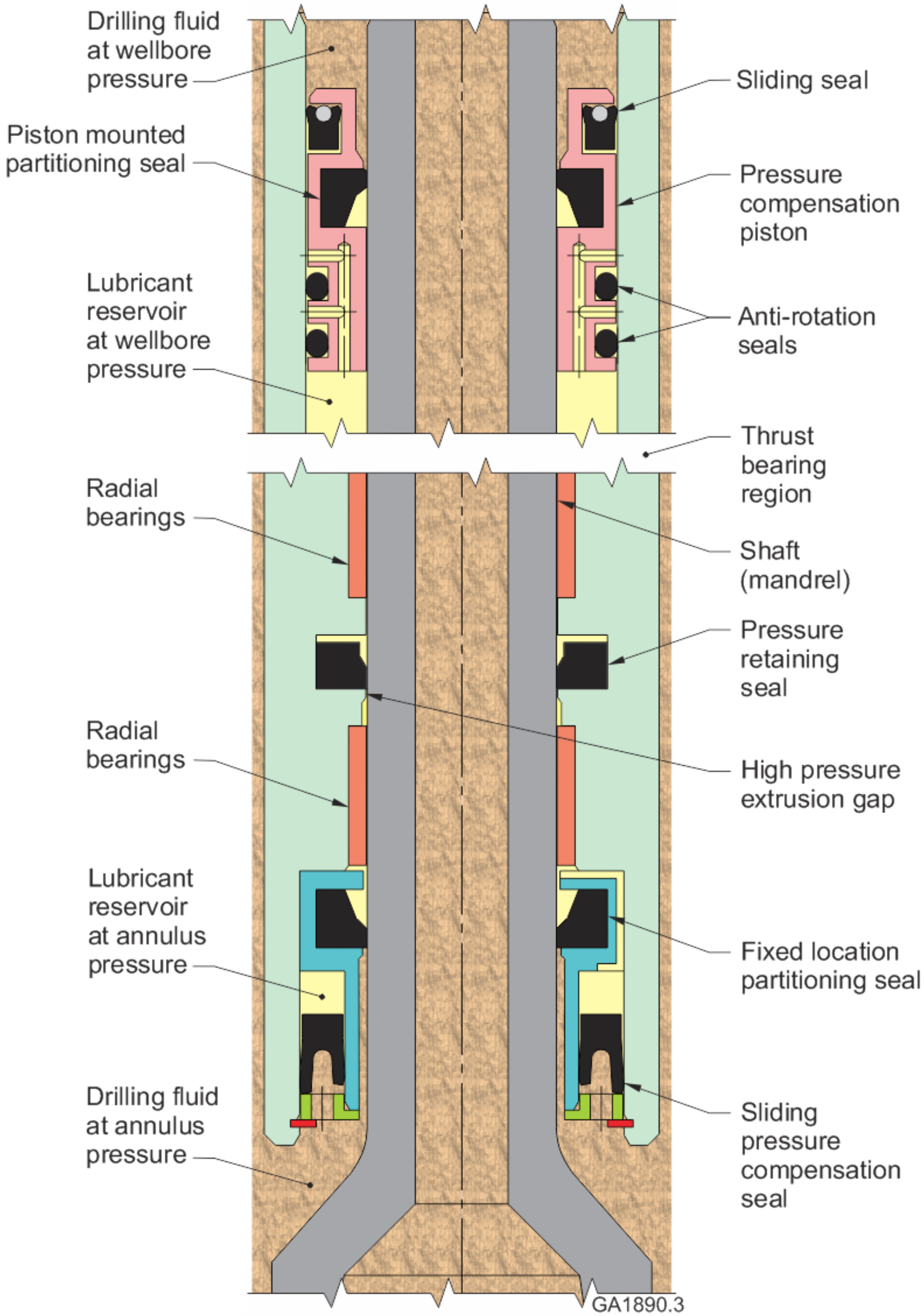

Figure 5

Concept: A bearing and compressive seal outboard of the pressure retaining seal

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