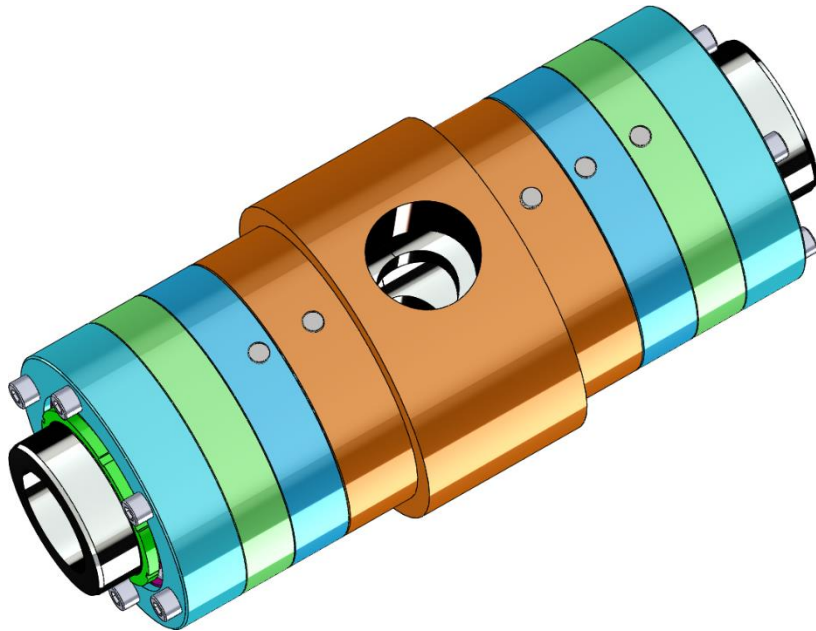


Chapter E5

Using Kalsi Seals in side port swivels



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1. Introduction

This chapter describes what we believe to be the best mechanical arrangement for using Kalsi Seals in a high pressure side port swivel. The basic design is patent pending, and available for licensing.

The design is robust, simple to manufacture, and easy to assemble and disassemble. The design eliminates all radial O-rings, which are difficult to deal with during assembly, and are also potential failure points. This design also eliminates all custom threads used in earlier designs, and minimizes the number of precision machined surfaces, making it economical to manufacture.

Licensing information

Kalsi Engineering prefers to license the patent pending features of the side port swivel design with a simple unilateral license, granting permission to a specific manufacturer to manufacture, use, and sell the design, if used with rotary seals that are purchased from Kalsi Engineering. When this type of license is used, the licensing fee can be included in the price of the seals, or as a per-seal or per-swivel licensing line item, or for a part number representing a combination that includes the seal and the hardware technology license. Bidirectional licenses are also possible, but less economical, because they inevitably involve extended and expensive interaction between legal departments, delaying project completion and inflating the licensing cost.

2. Overview of the swivel figures

Figures 1, 2, and 3 are section views of a high pressure side port swivel that are taken at different angles to show various internal details. Only one half of the swivel is shown, to allow the image to be larger. Figure 1 is a section view taken through the bulkhead housing bolts. Figure 2 is a section view taken through the screws that retain the bearing housing, and Figure 3 is a section view taken through the internal porting. Figures 4 and 5 are enlargements that show internal porting detail.

The housing assembly consists of housing components that are held together with axially oriented screws. The housing components shown herein include a radially ported center section, bulkhead housings bolted to the central housing, and a bearing housing bolted to the outboard bulkhead housing. The screws that retain the bulkhead housings thread into tapped holes in the radially ported central housing, and the screws that retain the bearing housing thread into tapped holes in the outboard bulkhead housing. If desired, the housings can be increased in diameter so that one set of screws retains the bearing

housing and the bulkhead housings, reducing the number of tapped holes that are required.

The housing assembly and the rotatable mandrel are located axially and radially with respect to one another by bearings. Although not shown, it will usually be desirable to protect the bearings, and retain the bearing lubricant, by providing lip type rotary shaft seal outboard of the bearings.

3. How the side port swivel operates

The general purpose and arrangement of a side port swivel

The purpose of a side port swivel is to conduct a flowing high pressure fluid through the radial port of the non-rotating central housing, and into the axial throughbore of the rotating mandrel, via one or more radial ports in the rotating mandrel. One common use of a side port swivel is for the cementing operation of an oil well.

In a high pressure cementing swivel, the mandrel is supported and rotated by the rig, and the housing assembly and associated cement supply line are supported by the bearings. In other types of applications, the housing assembly is supported by the structure of other associated machinery, and the rotating mandrel is supported by the swivel bearings.

Seal lubricant pressurization

The first stage seal lubricant is pressurized to a value equal to or greater than the pressure of the fluid being conducted by the swivel. In the simplest embodiment, the pressure of the first stage seal lubricant is balanced to the pressure of the conducted fluid by a diaphragm or bladder that also serves as a lubricant reservoir. When this method is used, the partitioning seal has to be axially spring loaded (not shown) using the method described in chapter D9, or has to be an Axially Constrained Kalsi Seal. The housing for the diaphragm or bladder can be connected to an opening opposite the radial flow port in the central housing (not shown), and if desired, the lubricant can be communicated to the region between the partitioning seal and the first stage pressure retaining seal by cross-drilled holes.

In lieu of using a diaphragm or bladder arrangement, the first stage lubricant pressure can be provided at a pressure greater than the pressure of the conducted fluid using a pressure amplification-type lubricant reservoir or a computer-controlled lubricant supply system, as described in chapter D11.

In the two stage arrangement illustrated in this chapter, the second stage seal lubricant pressure is provided at a pressure that is one half the pressure of the first stage seal

lubricant. This allows the first stage pressure to be shared by the first and second stage pressure retaining seals, with each seal being exposed to ½ of the differential pressure.

Floating backup rings allow the minimum possible extrusion gap clearance

The key to the side port swivel design of this chapter is the use of our patent pending floating backup rings. These rings, which are axially force balanced and radially pressure balanced, are guided radially by a journal bearing-type fit with the mandrel. This fit, and the radial pressure balance, allow the minimum possible extrusion gap to be used, thereby achieving maximum seal extrusion resistance. The axial force balance allows the backup ring to move laterally to accommodate mandrel runout and misalignment, while avoiding heavily loaded metal-to-metal contact at the extrusion gap, and the seal-damaging heat that such contact would produce.

Pressure capacity of each stage

We have performed two ~1,000 hour high pressure tests of 2.75” (69.85mm) seals with backup rings, with excellent results. The surface speed was 252 ft/minute (1.28 m/s), which is equal to 200 rpm on a 4.875” (123.83mm) OD mandrel. The lubricant temperature for these tests was 120 to 130°F (48.9 to 54.4°C).

In these tests, the interior of the mandrel was not pressurized. As a result, the extrusion gap clearance increased with increasing pressure, due to pressure-induced mandrel deflection. We estimate that the diametric extrusion gap clearance was in the range of about 0.002” at atmospheric pressure and about 0.005” at test pressure.

In one of the tests, a pair of -303 plastic lined Extra Wide Enhanced lubrication Kalsi Seals were tested on a floating mandrel using an ISO 150 viscosity grade synthetic hydrocarbon lubricant at 7,500 to 7,800 psi (51.71 to 53.78 MPa). In the other test, a pair of -106 Extra Wide Enhanced lubrication Kalsi Seals were tested on a shaft with 0.010” runout (FIM) using an ISO 320 viscosity grade synthetic hydrocarbon lubricant at 5,000 psi (34.47 MPa). In both tests, the rotary seals could have continued to run longer.

We believe these test results can be used as a general guide when designing pressure stages that have extrusion gap clearance in the same range as the tests. Larger extrusion gaps and higher temperature will reduce pressure capacity.

4. Mandrel design considerations

Mandrel diameters

The first step in mandrel design is selection of the bore diameter that is required to accommodate the desired volumetric flow rate of the conducted fluid. In cases where the conducted fluid is abrasive, the prudent designer will also consider the flow speed, and its

effect on mandrel erosion. Erosion can be managed by the use of a protective hard coating, and by utilizing a less erosive flow speed.

The second step in mandrel design is selecting the seal and bearing diameter. For optimum results, the wall thickness should be designed to minimize pressure-induced mandrel deflection at the seals and bearings, and to handle the mechanical loads the mandrel will experience. This is best accomplished through the use of finite element analysis. The analysis must include the pressures acting along the interior and exterior of the mandrel, including pressure changes associated with failure of any of the rotary seals.

In order to properly design the backup rings, a complete understanding of the dimensional effect pressure has on the sealing diameter and bearing mounting diameter is necessary and the bearing mounting surfaces. The designer should confirm that stress levels are at acceptable values for normal operating pressure and for proof testing.

Seal running surface and installation path

The mandrel surfaces contacted by the rotary seals and the backup rings are protected by a ground and polished tungsten carbide surface, per the normal practices described in chapter D2. The installation path for the Kalsi Seals includes an installation chamfer and a recess at the radial mandrel ports, to avoid cutting the seals during installation, as described in chapter D3. These mandrel features are preferably protected from corrosion to avoid corrosion related seal damage, as described in chapter D3.

Thrust shoulder and bearing nut

The mandrel provides an integral thrust shoulder for the thrust bearing set, as shown, that is designed to convey thrust from the mandrel to the bearings. The thickness is designed to handle anticipated thrust at acceptable stress levels. The shoulder diameter is per the bearing manufacturer's recommendations, and ideally will be at a diameter that allows the bearings to be removed by using a disassembly sleeve that bears against the inner race of the bearing.

The bearing nut and threads are designed to handle the anticipated thrust at acceptable stress levels. The bearing nut is designed to include a locking method (not shown), to prevent loosening in service. In the case of relatively small diameter swivels, it may be possible to use a standard AFBMA nut and the associated AFBMA deforming tang lock washer. In other cases, it will be necessary to custom design the bearing nut and locking arrangement.

5. Central housing design considerations

Overall size of the central housing

The central housing is sized to accommodate the required radial port size, internal clearance over the mandrel ports, and bolt engagement length at acceptable stress levels for mechanical, operating pressure, and proof pressure loads, while minimizing pressure induced deformation of the grooves for the rotary seals.

The groove for the partitioning seal

The groove for the partitioning seal is designed in accordance with chapter D5. If the partitioning seal is axially spring loaded, also refer to chapter D9. The extrusion gap that faces the conducted fluid is designed in accordance with the abrasive exclusion guidelines in chapter D7. Extrusion gap details are most clearly seen in Figures 4 and 5.

The recess for the first stage pressure retaining seal

The cylindrical bore and lubricant side wall of the recess for the first stage pressure retaining seal are designed in accordance with the instructions for the comparable seal groove surfaces in chapter D5.

Pressure communication porting

Referring to Figure 3 for an overall view of the pressure porting, the first stage lubricant pressure is communicated to a pressure distribution groove that is located near the groove for the partitioning seal. To minimize pressure lag, the axial length between the distribution groove and the seal groove should be minimized, as described and illustrated in chapter D11. Porting details are best understood by referencing the enlarged details of Figures 4 and 5.

The pressure porting also includes axial ports communicating balancing pressure to the back sides of the backup rings, to provide axial force balance. This pressure is communicated through housing interfaces via small diameter local O-rings. The use of small local O-rings minimizes hydraulic thrust area, and thereby minimizes the pressure load acting on the bulkhead screws.

The pressure porting also includes ports that communicate pressure to the outer surfaces of the backup rings, to provide radial pressure balance for dimensional stability.

6. Backup ring and bulkhead design

Detailed instructions for designing the backup rings are provided in chapter D17. As shown by Figure 2 of this chapter, the anti-rotation pin can be a loose pin that is inserted from the ID of the backup ring, and retained by the mandrel.

As a general overview, the axial depth of the recess that receives the backup ring is critical, because it controls the face-type extrusion gap clearance that the outer portion of the rotary seal is exposed to. The axial length of the backup ring and the depth of the recess are tightly toleranced, assuring that the backup ring is just a few thousandths of an inch shorter than the recess depth. These are two of the most critical dimensions of the assembly. The other two critical dimensions are the sealing diameter of the mandrel and the inner diameter (ID) of the backup ring.

The ID of the backup ring is designed to be as tight a fit as possible while still assuring freedom of relative rotation between the mandrel and the backup ring. Design of this fit requires the designer to consider tolerances, differential thermal expansion, and pressure induced deformation.

The diameter of the recess that receives the backup ring outer diameter (OD) is designed to accommodate the maximum anticipated lateral motion of the backup ring, in MMC tolerance conditions.

7. Housing interface design

Several of the housing end surfaces are sealing surfaces for face seals, and must have a suitably smooth surface finish—a 32 micro-inch AA or better surface finish is recommended. The end surfaces impact overall alignment of the housings, and must be machined parallel to one another, and square with critical bore surfaces.

The housings are aligned to one another by overlapping piloting surfaces near the OD of the housings. The MMC fit of the pilots should correspond to the MMC RC 3 class of fit, however the LMC fit can be looser than the LMC RC 3 class of fit, so that tolerances can be looser than RC 3 tolerances. The axial length of the overlap of the piloting surfaces should be kept short, to prevent the possibility of misalignment-related binding due to the “sticky drawer” effect. A concentricity tolerance should be provided between the housing pilots and critical interior surfaces, such as the extrusion gap bores, and the bore that receives the outer races of the bearings.

Preferably, the central housing and the bulkhead housings incorporate an indexing feature, to ensure that they are assembled in proper angular alignment. One practical indexing feature is the use of axially oriented indexing pins, and corresponding recesses in the mating parts. Another practical indexing feature is to use a slightly uneven angular spacing of the hole pattern for the bulkhead screws, so that the holes only line up when the housings are in the correct angular orientation.

8. Bearing housing design

The bearing housing should be designed with sufficient strength and bolt retention to handle anticipated mechanical loads. The bore that receives the outer bearing races should be designed with both the bearing manufacturer's recommended fit and any anticipated pressure related expansion of the inner races in mind.

In many cases, it will be desirable to incorporate a lip-type rotary shaft seal in the bearing housing, to retain a bearing lubricant. In such cases, the bearing cavity should be vented, to prevent pressure buildup that could damage the lip seals.

9. Miscellaneous considerations

Avoid thermal binding of the bearings

The images included in this chapter only show the end of the swivel that incorporates the thrust bearings. At the opposite end of the swivel, the bearings must be free to slide axially, per normal bearing fitting practices, to accommodate axial differential thermal expansion between the mandrel and the housing assembly. Do not incorporate bearing thrust shoulders at the opposite end of the swivel!

Optional barrier seals

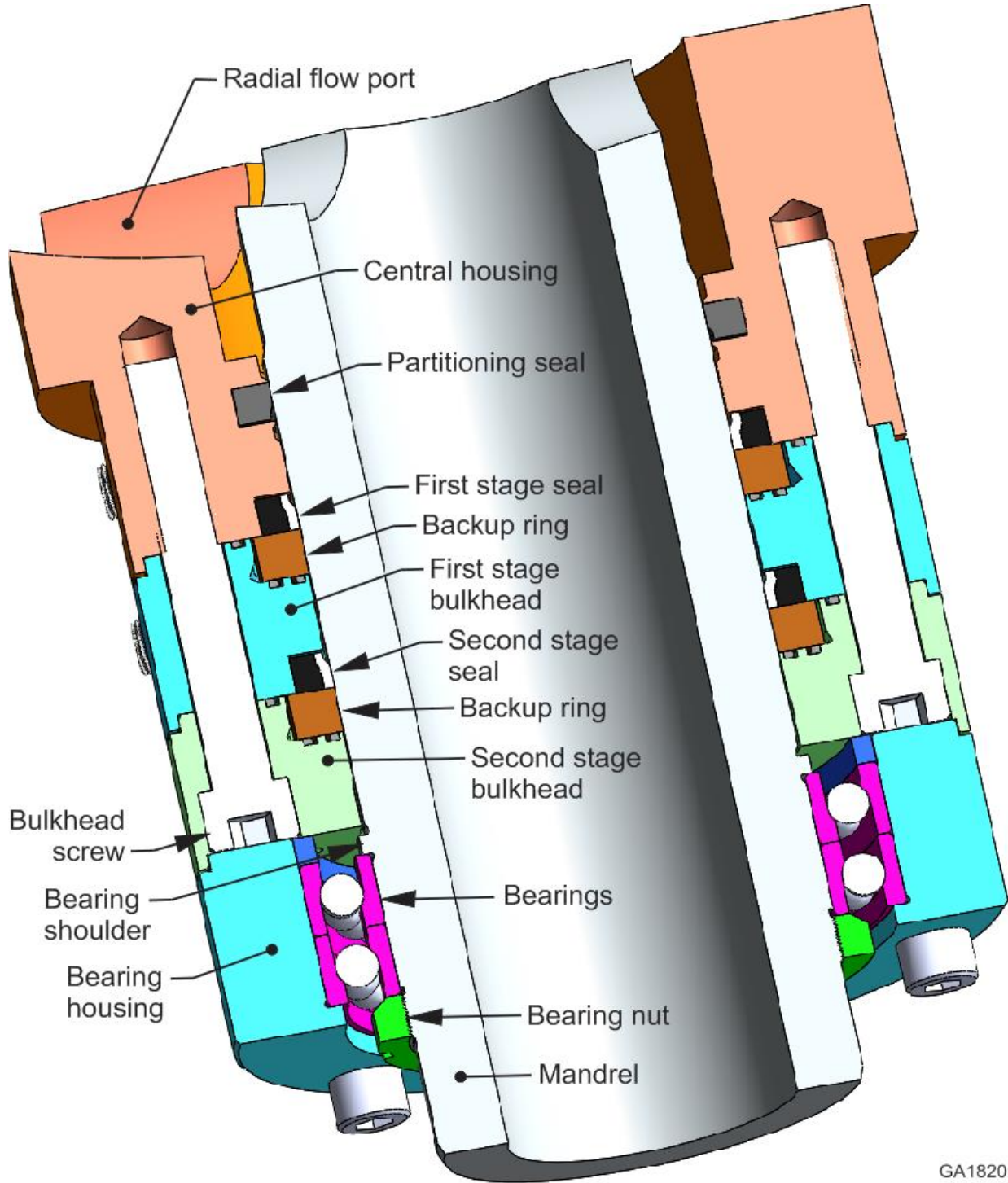
In applications where the conducted fluid may contain a wide variety of chemicals, it may be desirable to incorporate a lip-type barrier seal that faces the conducted fluid. This barrier seal can be made of a different material than the partitioning seal, to resist a different spectrum of chemicals. When a barrier seal is used, the region between the barrier seal and the partitioning seal is filled with lubricant, and pressure balanced to the conducted fluid. Information on the use of barrier seals is provided in chapter D10.

Integral first stage lubricant supply

In some cases, it may be desirable to mount the first stage lubricant supply directly to the central housing. For example, a lubricant supply in the form of a piston-type amplifier or a diaphragm and its related housing could be radially mounted to the central housing at a radial port that is oriented 180° from the radial flow port. This arrangement would allow the lubricant supply to be pressurized by the conducted fluid, and also admits to the use of manifold-type cross-drilled holes to conduct the first stage lubricant to the region between the partitioning seal and the first stage pressure retaining seal. Information on lubricant supplies is provided in chapter D11.

Second stage lubricant pressure

The general concept of pressure staging is described in chapter D18, along with descriptions of methods for supplying staged lubricant pressure.



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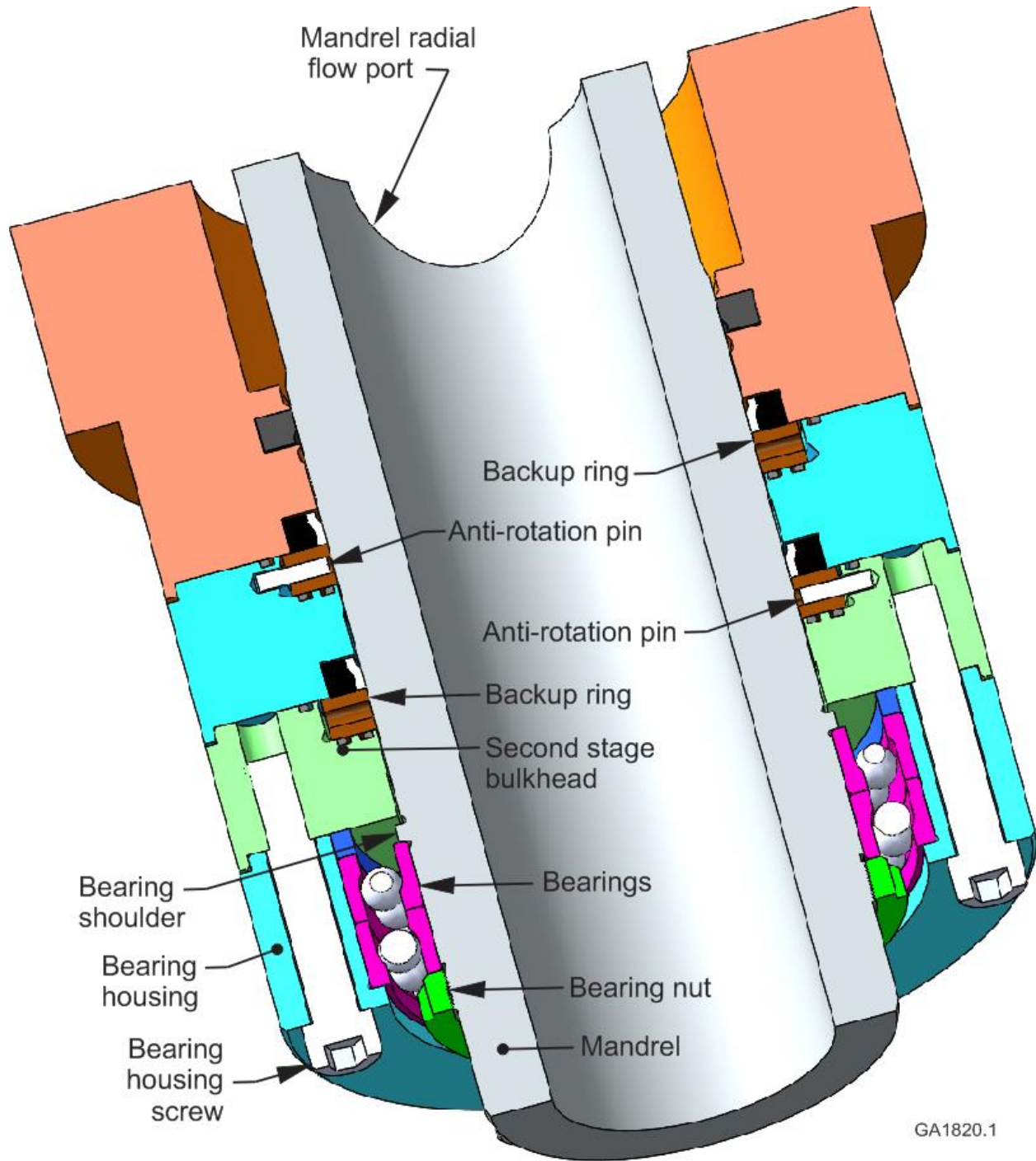
Figure 1

Side port swivel sectioned through the pressure retaining bolts

This view is taken through two of the socket head cap screws that retain the bulkhead housings to the central housing. These screws are sized to withstand the hydraulic force generated by the first stage pressure acting over the sealed area between the central housing and the first stage bulkhead housing, and the mechanical load acting through the thrust bearings.

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Figure 2

Side port swivel sectioned through the bearing housing bolts

This view is taken through the screws that retain the bearing housing to the second stage bulkhead housing. These screws are sized to withstand the mechanical forces acting through the thrust bearings. This view also shows the backup ring anti-rotation pins, which are inserted from the inside of the backup ring, and captured by the mandrel.

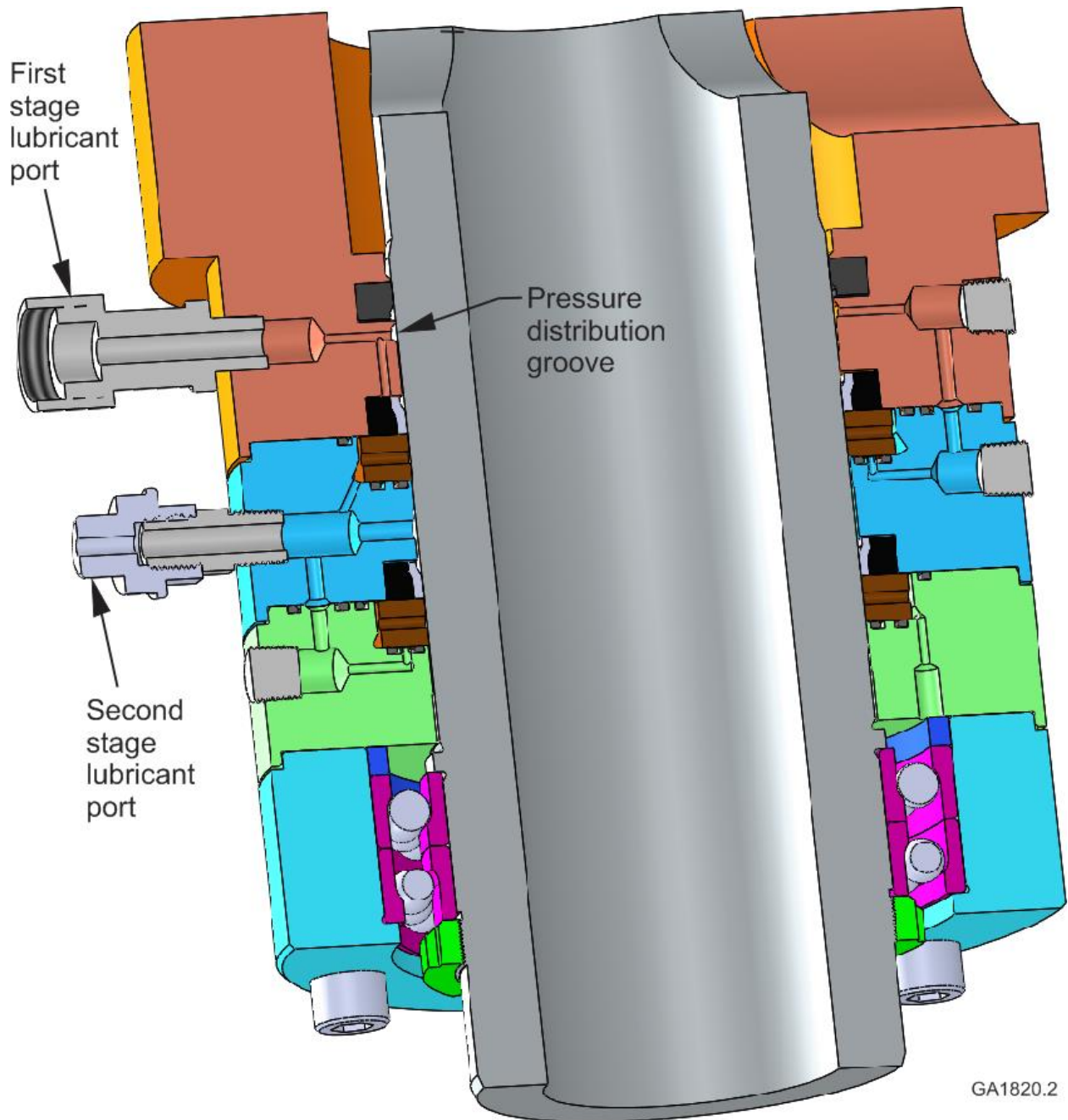


Figure 3

Side port swivel sectioned through the internal lubricant porting

This section view is taken through the porting that provides lubricant pressure to the seals and backup rings. The porting details are shown more clearly in the enlargements of Figures 4 and 5.

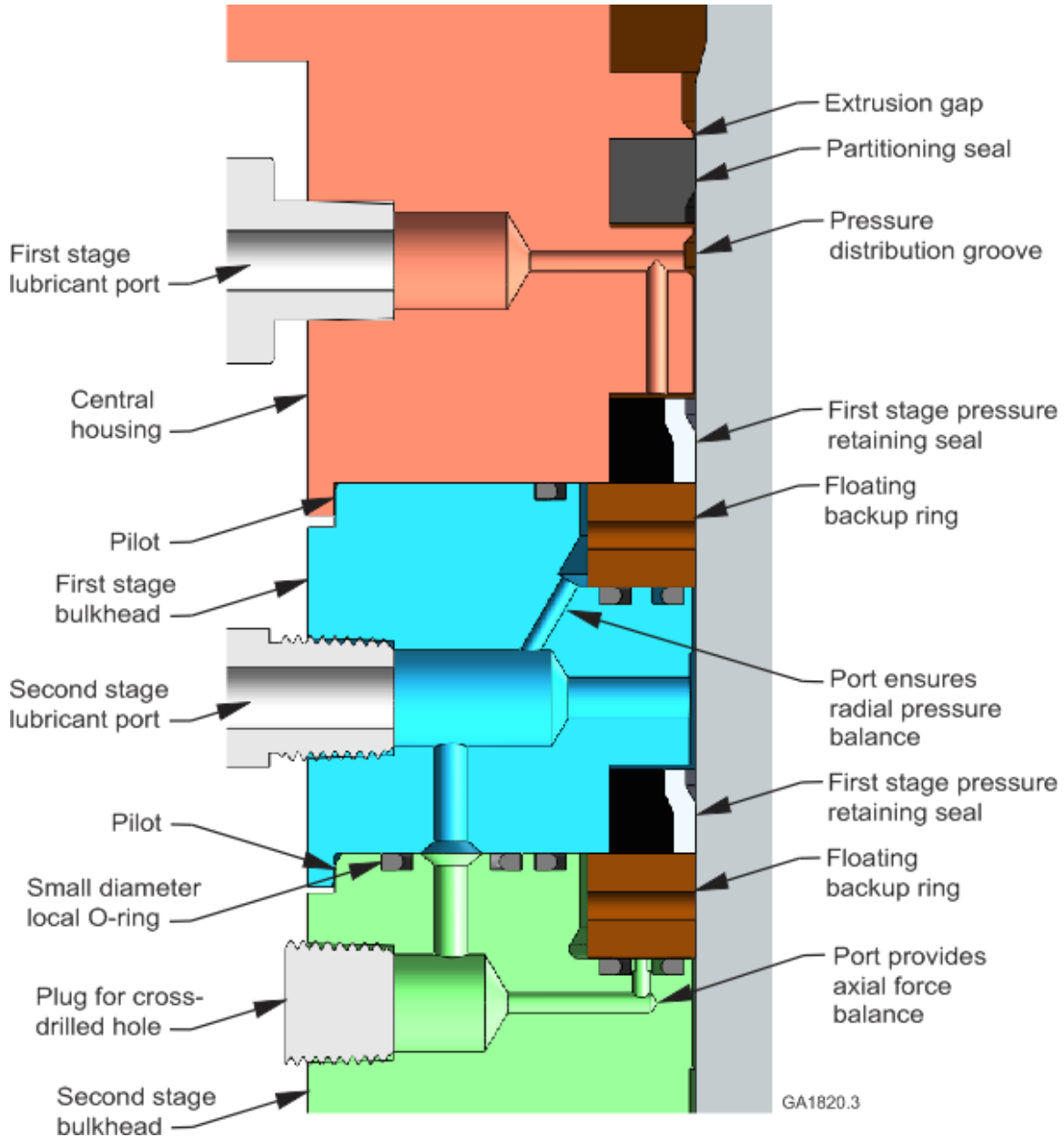


Figure 4

Enlargement of side port swivel sectioned through the lubricant inlet ports

This view shows the cross-drilled hole arrangement that provides second stage lubricant pressure to the back side of the second stage backup ring, creating axial force balance. It also shows the porting that assures radial pressure balance of the first stage backup ring. The annular pressure distribution groove, which assures rapid pressure transmission of the first stage lubricant pressure to the partitioning seal is also shown.

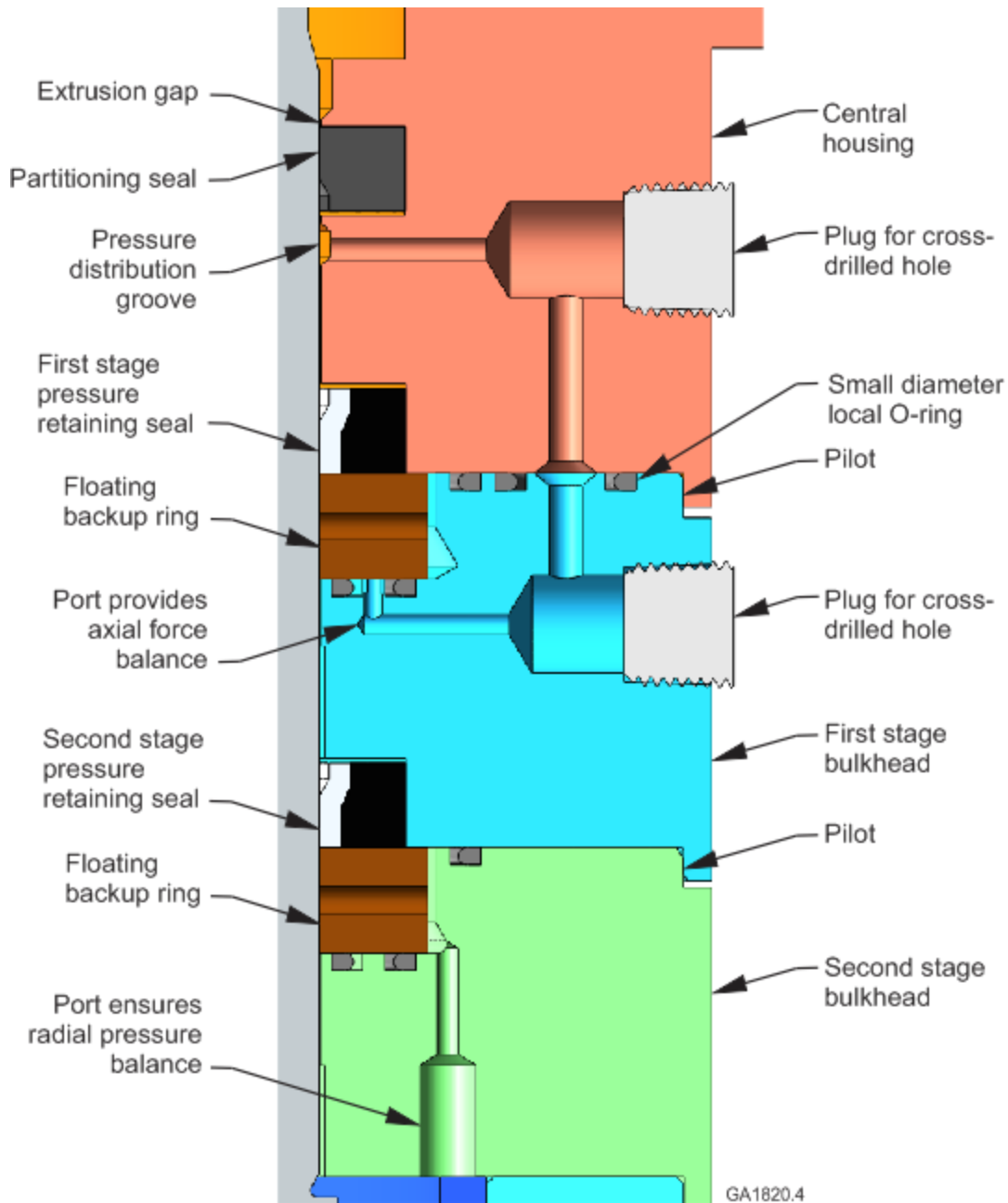


Figure 5

Enlargement of side port swivel sectioned through the plugged ports

This view shows the cross-drilled hole arrangement that provides first stage lubricant pressure to the back side of the first stage backup ring. It also shows the porting that assures radial pressure balance of the second stage backup ring.