Development of A High-Load Capacity Test Rig to Evaluate the Static Performance of Process Fluid-Lubricated Thrust Bearings

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Abstract. Electrical submersible pumps (ESPs) are artificial lift devices widely used in the oil and gas industry to increase production. For unconventional wells, which includes subsea applications, installation cost is an order of magnitude higher than the cost of the typical ESP, thus pump reliability is paramount. Contamination of the thrust bearing lubricant oil by process fluid due to seal failures is one of the leading causes of thrust bearing failure. While improving seal reliability can prevent thrust bearing failures, an alternative approach is to develop process-fluid lubricated thrust bearings. This research aims to characterize and study the performance of thrust bearings operating with process fluid lubrication. A component level-level test rig for testing the performance of thrust bearings in any mixture of water, sand, and air is designed, built, and commissioned. Initial tests of the PEEK and Tungsten Carbide (WC) bearings include a load capacity test to determine the hydrodynamic limit of the bearing, a hydrodynamic test to corroborate test rig stability, and a wear test to determine if the bearings can operate feasibly past hydrodynamic limits in a mixed lubrication regime. Initial results show that lubrication starvation can occur in an unpressurized chamber for low viscosity fluids that are not supplied directly to the bearing. Furthermore, the film thickness, temperature rise in the pads, and load capacity all are lower for water lubricated bearings when compared to oil lubricated bearings. Results suggest that liner material affects wear rate but not load capacity; however, operating even the WC bearings past hydrodynamic limits causes substantial wear, and is not recommended as an operating mode. Polycrystalline Diamond (PCD) coated mixed lubrication bearings were also tested. These bearings, although requiring higher drive torque, sustained higher loads than the tilting pad bearings without significant wear.

1. Introduction

Electric Submersible Pumps (ESPs) are predominantly used as an artificial lift mechanism for downhole operations in the oil and gas industry [1]. Seal failure, resulting in contamination of the bearing lubricant by process fluid, leads to erosion, abrasion, and seizure of the bearing, and is a major source of failure for these machines [2]. These failures can be particularly costly, especially in subsea applications, where the cost of replacing a failed ESP can be higher than the cost of the machine itself [3]. Two approaches to address this failure mode include improving the seals or designing process fluid-lubricated thrust bearings. This study focuses on the latter. The use of properly qualified process fluid lubricated thrust bearings can reduce maintenance costs and increase the longevity of ESPs downhole operation even when there is a seal failure. A test rig was designed to evaluate a variety of thrust bearing designs in process fluid lubrication to assess their viability for this application.

A brief review of the existing literature on test rig construction for thrust bearings follows. Russell [4] presents a test rig to evaluate the performance of ball bearings, button insert PCD bearings, and hydrodynamic bearings up to 7000 lbf (31.1 kN) and 1000 RPM. The rig is vertically mounted, with an accessible test chamber that is flooded with lubricant during testing, and
hydraulic pistons supplying axial force. A rig developed by Harika [5] to test water contamination, similar in design to one developed by Pasanen [6] to test hydrodynamic lubrication, is horizontally mounted, makes use of a pneumatic jack to provide axial load, and employs a belt for power transmission. Another notable design was created by Gregory and Kingsbury [7], which makes use of a sliding housing for test bearing access. The test rig is capable of large loads of 22,000 lbs (97.8 kN) and speeds of up to 11,000 RPM but can only test one bearing at a time and uses clean oil lubricant.

2. Test Facilities

The test rig was designed to meet the following requirements and functions:

- Allow for testing of thrust bearings in water, air, and sand slurry conditions.
- Provide quick access to the test bearing for inspection and replacement.
- Have the ability to test bearings to failure/seizure without damaging the rig.
- Maintain high unit loads for extended periods of time in order to observe bearing wear.
- Accommodate a variety of bearing sizes and configurations for testing (up to 8”~200mm).

The final design of the rig, shown in Figure 1, draws from Russell’s test rig design by implementing a vertically mounted configuration and a flooded test bearing chamber. Its three major subassemblies are the load driving assembly, slave bearing spindle assembly, and support structure assembly. The load driving assembly is completely removable to allow for quick test bearing access, and the test chamber is sealed to isolate the slave bearings from the process lubrication present in the test bearing area. Three hydraulic cylinders apply a maximum axial force of 30,000 lbf (133.4 kN) onto the test bearing. A torque-limiting coupling protects the motor from overload in case of test bearing seizure. The test chamber features three load cells, one for each hydraulic piston, three water submersible proximity probes to measure bearing deflection and hydrodynamic film thickness, and ten temperature sensors, thermocouples or RTDs depending on the application, to monitor bearing temperature. A tandem torque meter/tachometer installed between the coupling and the motor provides the system torque and shaft speed.

![Figure 1. Section cut view of final test rig design with components labeled.](image-url)
Figure 2 shows the components of the load driving and slave bearing spindle assemblies. The end flange in the load driving assembly seals the test chamber from the bottom, while the axial location of the test bearing is controlled by the spacer plate, load cell plate, and bearing plate, along with the hydraulic pistons. Three slip fit locator pins prevent rotation of the bearing while testing is conducted. The bearing plate was sized to prevent significant deformation while transferring the load from the hydraulic cylinders through the load cells to the bearing. In the spindle assembly, a set of tandem tapered roller bearings react the load applied to the test bearing, while the top single row bearing holds the weight of the spindle assembly. Locking nuts hold and preload the bearings, and both sets of bearings align the spindle rotor with its casing. The blind flange and mechanical seal separate the slave bearing chamber from the test bearing chamber, and through bolts secure it and the spindle casing to the support structure. A splash guard was installed between the blind flange and spindle rotor casing to shield the tandem slave bearings from any leakage through the mechanical seal, and a relief channel and hole provide an exit path to the leakage flow.

![Diagram of spindle assemblies](image)

**Figure 2.** Slave bearing spindle assembly (left) and load driving assembly (right).

Figure 3 depicts the external flow loop schematic of the test rig. Water (lubricant) flows from a reservoir through a heat exchanger designed to maintain a steady fluid temperature. A diaphragm pump operating with an accumulator and a check valve provides pressurized water to the test chamber. A pressurized air line connected at the entrance of the test chamber injects air to the inlet flow for multiphase testing. Shop air feeds the diaphragm pump, accumulator, and supplies the process lubrication mixture through a mass flowmeter and electronically actuated ball valve. The mixture exits the test chamber and returns to the reservoir. A sand auger feeds sand to the test chamber inlet flow at a constant rate and a filter in the reservoir return line captures sand particles past the test chamber.
3. Commissioning and Initial Results

After acquisition and assembly of all requisite components, testing was performed on two bearings to demonstrate test rig operability. Figure 4 shows the two test bearings comprising six tungsten carbide (WC) and PEEK tilting pads, respectively, with a stainless steel support structure and an effective outer diameter of 3.5" (8.89 cm). Table 1 lists the main bearing and pad dimensions, which are typical for an ESP thrust bearing. The thrust runner is made of stainless steel with a WC coating. The runner mating surface total runout and roughness are 0.008" (0.2 mm) and 8 μin (0.2 μm), respectively. All initial testing was performed with single-phase, water lubrication at 40 psig (2.75 bar).

Figure 3. External flow loop of test rig.

Figure 4. Tilting pad test bearings in the full, six pad orientation.

| Table 1 Test Bearing Dimensions |
|--------------------------------|
| Inner Diameter | 1.6” (4.06 cm) |
| Outer Diameter | 3.5” (8.89 cm) |
| Pad Length     | 0.97” (2.46 cm) |
| Pad Width      | 0.97” (2.46 cm) |
| Arc Length     | 1.14” (2.90 cm or 39˚) |
Preliminary testing revealed premature bearing failures even at no load condition. It was later determined that the bearings experienced immediate fluid starvation, and neither bearing was able to hold a film at any load condition. Rubbing began immediately following the application of load, causing the PEEK bearing to fail completely, while the WC bearing experienced significant wear on the leading edge of the pads, as can be seen in Figure 5. The fluid starvation failures were attributed to a lack of an unrestricted flow path to the inner diameter of the bearing. This condition resulted from a combination of multiple factors, including the outward-pumping action of the bearing during operation, the limited space between the shaft and the bearing inner diameter, and the lubricant being supplied from the bearing outer perimeter.

In order to confirm fluid starvation as the cause of these results, three pads were removed from a fresh set of bearings. While this decreases the theoretical absolute load capacity of the bearing, it provides a flow path for lubrication to the inner diameter of the bearing that was absent in the six-pad configuration. Bearing performance tests were conducted over a range of operating speeds on both PEEK and WC bearings in this configuration. The WC bearing was able to hold a hydrodynamic film up to 210 psi (~15 bar) unit load at 900 rpm, 250 psi (~17 bar) unit load at 1800 rpm, and 350 psi (~24 bar) unit load at 3600 rpm. After reaching these values, film rupture occurred as evidenced by the spikes in torque and pad temperature, as shown in Figure 6. The pad average temperature decreased briefly when the unit load increased from 480 to 550 psi (33 to 38 bar). This trend may be related to a slight tilt in the bearing orientation due to pressure redistribution in the hydraulic cylinders. Past the hydrodynamic limit of the bearing, contact between the runner and bearing pads began to occur, leading to the wear pattern on the WC bearing pad shown in Figure 7. As with the six-pad testing, the wear is predominantly located at the leading edge (LE) of the bearing pads. The PEEK bearing was able to perform similarly but was not run past the hydrodynamic limit to avoid inducing the same bearing failure produced in the six-pad configuration. This resulted in pads that show minimal wear, most likely caused by asperity contact abrasion due to the hydrodynamic film thickness being as low as 0.1 mils (2.54 µm) with water lubrication.

Figure 8 depicts the friction coefficient from testing at 3,600 rpm. The friction coefficient is calculated by dividing the torque over applied load and mean radius of the bearing. The graph resembles a Stribeck curve showing three distinctive trends that are qualitatively identified as a hydrodynamic, mixed and boundary lubrication regions. In the hydrodynamic region the friction coefficient decreases as the load increases, which indicates torque is relatively constant and the bearing is operating without contact between runner and pads. The mixed lubrication region displays a modest increase in friction coefficient as the unit load increases, which is associated
with a torque increase due to initial asperity contact between the stationary and rotating surfaces. The final region is characterized by a relatively constant friction coefficient as the load was continually applied even after contact occurred, coinciding with an increase of the input torque and pad surface temperature (i.e. film rupture).

**Figure 6.** WC bearing torque vs. unit load for 900 RPM (top left), 1800 RPM (top right), 3600 RPM (bottom left), and temperature vs. unit load for 3600 RPM (bottom right).

**Figure 7.** WC bearing pad (left) and PEEK bearing pad (right) wear patterns after testing.
Figure 8. Friction coefficient vs unit load for WC wafer at 3600 RPM. The hydrodynamic, mixed, and boundary lubrication regions are marked green, blue, and red respectively.

A five-hour wear test was also performed on a fresh WC bearing at 3600 rpm and identical lubricant flow conditions. Axial loading was set at 365 psi unit load, just above the hydrodynamic load capacity of the bearing determined in previous testing, in order to induce bearing/runner contact. As seen in Figure 9, significant wear occurred on both the runner and bearing. The extreme amount of wear present on the runner is attributed to the fact that this component was made out of 17-4 stainless steel substrate with a WC coating. After the coating was removed by wear, the softer substrate was exposed to the bearing's WC wafer pads and began to wear at a much faster rate.

Figure 9. WC thrust bearing (Left) and thrust runner (Right) after wear test.

4. Flow Path Modification and Polycrystalline Diamond Bearing Testing
A number of modifications were made to components within the test chamber in order to permanently resolve the fluid starvation issue experienced during initial testing. The chamber's lubrication inlet was rerouted from the side of the test chamber casing to the bottom of the end flange. This change, along with the addition of an internal pipe through the bottom spacer, allowed
redirecting the inlet flow to the inner diameter of the bearing. The spindle rotor was also truncated in order to prevent it from obstructing the rerouted flow.

Following the completion of these modifications, testing began on Polycrystalline Diamond (PCD) coated mixed lubrication bearings. The geometry of these bearings and runners consists of a discontinuous set of PCD inserts in a circular pattern. Work done by Sexton and Cooley found that these bearings undergo a run-in process of up to 100 hours before their optimum load-carrying capacity and friction coefficients are reached [8]. The performance of the PCD bearing in process lubrication showed signs of this process. Figure 10 shows a comparison of torque and friction performances between tests conducted over the first fifteen hours of bearing operation. Each test was performed at 1000 RPM and a chamber pressure of 25 psig. Between each measurement, the rig ran under steady state conditions while holding a high load to accelerate the run-in process. PCD bearing performance consistently improves with run time, as expected. These bearings, which operate in a mixed lubrication regime, were able to sustain higher loads than the tilting pad bearings without significant wear.

![Figure 10. PCD bearing experimental results during run-in process: (a) friction torque vs. axial load and (b) friction coefficient vs. axial load.](image)

**Conclusions**

A test rig was successfully developed in order to test a variety of thrust bearing designs in process fluid lubrication. During the rig commissioning process, a fluid starvation issue was identified and resolved. This was first accomplished by removing half of the pads of the tilting pad bearings being tested to create a flow path the inner diameter of the bearing, and later by directly rerouting the lubricant flow to the bearing's inner diameter. For hydrodynamic tilting pad trust bearings, pad surface material was found to have little affect below the hydrodynamic lubrication limit. WC performed better than the softer PEEK bearing above the hydrodynamic limit, but with wear significant enough to recommend avoiding any operation of the bearing past that limit. PCD bearings, which operate in boundary and mixed mode lubrication performed at higher loads than the hydrodynamic bearing without significant wear. The rig is currently being modified for multi-phase lubrication testing on larger tilting pad bearings.

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