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SUPERCRITICAL CARBON DIOXIDE EXTERNALLY PRESSURIZED POROUS MEDIA BEARINGS

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ABSTRACT
The Brayton Loop facility at Sandia National Laboratories (SNL) has been instrumental in identifying technology gaps and developing solutions critical for the commercialization of supercritical carbon dioxide (sCO2) power cycles. The benefits afforded by these new conversion technologies are required to fully leverage the potential from the next generation of nuclear reactors: increased efficiency, smaller footprint, and the ability to use dry cooling. While the recompression closed-loop Brayton Cycle (RCBC) test loop test program has been successful, its full potential has been stymied by the poor reliability of the turbine-alternator-compressor (TAC) bearings. While many of the early events were due to poor controllability issues, resulting from the first-generation power electronics, it demonstrated their lack of robustness necessary for these high-power density machines.

Despite contactless operation at speed, foil air bearings are in contact with the shaft at all speeds that are below that required for lift-off; for the foil bearings in the TAC it's around 8,000 RPM. Therefore, surface to surface contact occurs during start and shutdown, leading to wear. These unpreventable events contribute to loss of low friction surface treatment and are life limiting. While fully developed foil bearings are commercially applied to air cycle machines and small gas turbines (microturbines), rotors of low mass, this failure mechanism is amplified for large commercial sCO2 systems with heavy rotors and larger loads, rendering them unsuitable for future commercial small modular reactors (SMRs). It was therefore decided to identify and test more robust bearing solutions that would not only improve run time accumulation and facilitate performance and cycle configuration testing, but also demonstrate solutions which are scalable and better suited for large commercial systems.

As part of the TAC improvement program, and after the control system upgrade, the bearings were identified as the main technology to increase TAC reliability and efficiency. TACs currently employ aerodynamic foil type bearings. This report considers the retrofit design, acceptance testing, and the initial testing of a TAC retrofitted with externally pressurized porous (EPP) gas bearings. Both radial and axial constraint are required to support the rotor and react the axial thrust load. This was accomplished by replacing the compressor journal and thrust bearings with a single EPP bearing assembly, and a simple replacement of the journal on the turbine side.

NOMENCLATURE

AC	alternating current
BNI	Barber Nichols Incorporated
CDP	compressor discharge pressure
CMC	ceramic matrix composite
DC	direct current
DOE	Department of Energy
EPP	externally pressurized porous
KEB	KEB America, Inc.
NE	nuclear energy
PMG	permanent magnet generator
RCBC	recompression closed-loop Brayton Cycle
rpm	revolutions per minute
sCO2	supercritical carbon dioxide
SCFM	standard cubic feet per minute
SNL	Sandia National Laboratories
SMR	small modular reactor
TAC	turbine-alternator-compressor
TRL	technology readiness level
V	volt

1. INTRODUCTION

The recompression closed-loop Brayton Cycle (RCBC) test loop at SNL has suffered from reliability and functionality issues which has impaired development progress of the RCBC cycle and limited its endurance capabilities for validating components. With the addition of the new Power Electronics, functionality, and control issues of the turbine alternator compressors (TACs) has been resolved, and the platform can now be leveraged to

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understand cycle control and for verification of low TRL components. such as bearings and seals.

Bearings are a critical component of any high-speed machine, and development of bearing systems for sCO2 power conversion technologies is one of NE's top priorities to support commercialization goals. While rolling element and other oil lubricated bearings are abundant in all commercially available equipment, sCO2 is a solvent, which prohibits the use of oil within the power cycle boundary. This issue is typically solved by adding seals and buffer cavities. This adds complexity, cost, and greatly increases the axial distance between supports. The increased bearing span not only reduces the shaft first bending mode but can also affect turbomachinery performance.

Gas foil bearings are used in commercial applications for air-cycle machines and microturbines and were implemented in the first-generation TAC's due to their ability to run without oil lubrication. Unfortunately, they are limited in application to lower power machines, and are particularly sensitive to start/stop cycle wear and fine particulate contamination. This led to premature catastrophic failure of the TACs with little warning. Experience with an EPP thrust bearing on another test platform at Sandia has provided insights to their robustness and suitability for sCO2 applications. Their suitability for further investigation was also presented in the conclusions by Ertas[2]. EPP bearings are simple, non-contacting, become self-lubricated bearing under loss of supply pressure, and with low-flow characteristics they can also function as a seal. To test these bearings in the TAC, Newway Air Bearings was tasked with designing a set of EPP bearings for retrofitting in the TAC, while foil air bearings permit very tight packaging, with some compromise, a suitable arrangement was developed that maintained the critical rotor geometry. This paper discusses the design, acceptance test results, and the initial test results of the retrofitted EPP bearings in the modified TAC in the sCO2 loop.

2. EPP Bearings

Aerostatic bearings are universal outside of the turbo machinery industries. Commonly referred to as air bearings, they are widely employed in the machine tool, metrology, semiconductor, and medical industries. There are tens of thousands of high-speed spindles using aerostatic air bearing technology. Besides metrology applications, higher load applications include drilling, milling and precision grinding. Stiffness, damping, and load capacity are important factors in cutting technologies, and these same attributes have recently encouraged acceptance in several sCO2 turbomachinery applications. The most common compensation techniques used in aerostatic gas bearings are orifice and porous compensation. Porous bearings have an advantage over orifice-type bearings because they achieve lift at lower air gaps without experiencing collapse (see Figure 1). Collapse occurs when the gap becomes so restrictive it starves the surface of flow. An orifice type bearing requires a high percentage of the design pressure to achieve lift-off. This is because an orifice bearing has limiting features, comprising the orifice and any air distribution grooves, to establish the lift area. Alternatively, a porous bearing has the pressure distributed across the entire surface, with the flow distributed uniformly across the bearing flowing evenly through its pores.

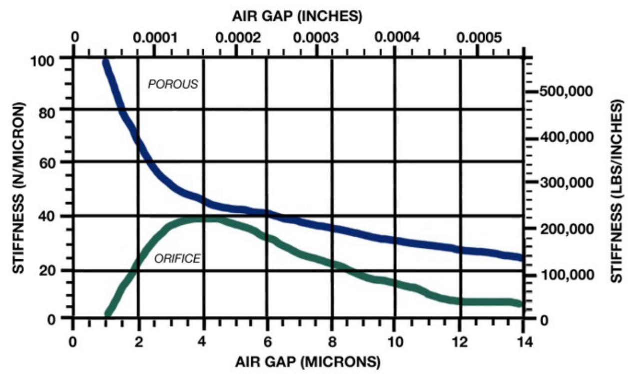

Figure 1: Comparison between orifice type and EPP bearings.

The ideal gas bearing supplies pressure equally across the whole wetted surface of the bearing, and automatically restricts and dampens the flow of gas at the same time. EPP bearings achieve this by diffusing the gas throughout a fine porous matrix and delivering it across the entire face of the bearing. Both graphite and carbons are naturally porous and are common material choices for the bearing matrix. The stability found in EPP bearings is largely due to the damping effect from the torturous passageways the gas must flow through to reach the face. This damping effect makes it difficult for the volume of air in the gap to change quickly, resulting in a naturally stable gas film. In the case where there is contact, both graphite and carbon are good plain bearing materials, widely used in industry, and add to their resiliency and robustness.

Figure 2: Tilt pad PMB system applied to a large rotor.

The small rotor diameter of the TAC and the retrofit requirement prevented the use of the preferred multi-pad journal EPP bearings shown in Figure 2, and fully cylindrical liners were employed for the TAC upgrade, as shown in Figure 3. A detailed description of the TAC hardware can be found in numerous

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published papers, including, (Darryn add the wright paper ref, and at the end in references) While this has the advantage of simplicity, it lacks several features that add additional value to tilt pad EPP bearings,namely, the spring-loaded pads, whose clearance is very small, and can be actively managed during operation, and the ability to use the pads spring loaded supports during transportation or when stationary. If desired, they can also be used as a brake to prevent overspeed or reduce low speed spooling during a controlled shut down. This tilting pad arrangement is suitable for large sCO2 systems that will power conversion systems for the next generation of large commercial reactors, which will extend into the hundreds of MWe.

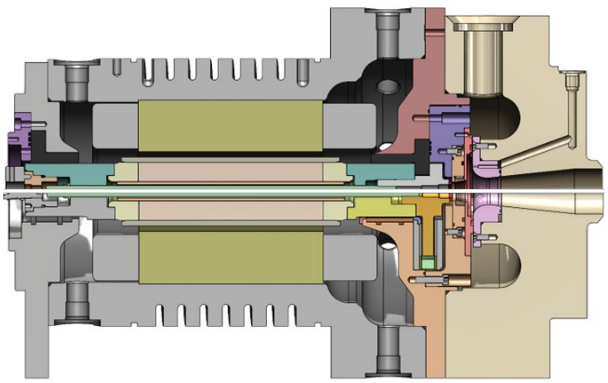

Figure 3: TAC cross section with EPP bearing in upper section, and foil bearing below.

EPP bearings also have high temperature capability. While the Graphite used in these test bearings can operate at 500°C continuously and is suitable for all but the most advanced high temperature reactors, the next generation of bearings made from Ceramic Matrix Composite’s (CMC) are expected to be able to operate up to temperatures of 850°C, well above the 750°C maximum temperature being discussed for today’s sCO2 power conversion cycles. Locating the bearings closer to the turbomachinery means that the losses associated with radial and axial tip clearances can be dramatically reduced. In addition, the reduced need for a parasitic cooling supply, and lower supply flows, further improves thermodynamic efficiency. When compared to oil lubricated bearings, additional efficiency gains come from the elimination of oil shear losses, and the parasitic power draw of the auxiliary oil systems to heat, pump and scavenge the oil.

2.1 EPP Bearing Theory

EPP bearings are an aerostatic type bearing which utilize compressed gas flowing through porous media to provide frictionless motion to various types of machinery. In the case of rotating machinery, these bearings can be in the form of Journals or Tilting Pivot pads to support a rotor. The load capacity of such bearings is close to that of Hydrostatic Oil bearings as the static working principle is similar. The stiffness of EPP bearings can vary with supply pressure and is able to adjust during machine operation. By increasing the supply pressure, the stiffness in the bearing gap increases and exerts a higher force on the rotor. With EPP journal bearings, the bearing gap is fixed and is decided during the design phase to conform with gap thermal transients. A very narrow gap will yield high stiffness, low eccentricity, and high damping. A larger gap (<0.025mm) allows for a larger range of stiffnesses, damping, and eccentricity, as seen in Figure 4. While eccentricity may be higher, the ability to vary the stiffness and damping though the machine operation is crucial to safe operation and machine longevity.

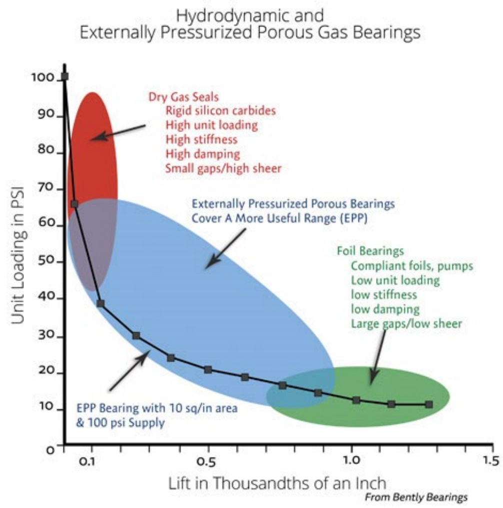

Figure 4: Air Bearing Lift Characteristics. [Ref: Newway]

The viscosity of the pressurized gasses is very low compared to oil. This reduces the shear power loss and achieves very low starting resistance in these bearings. Since the shear loss is low, there is little heat buildup compared to oil bearings. This can be seen in Figure 5 below.

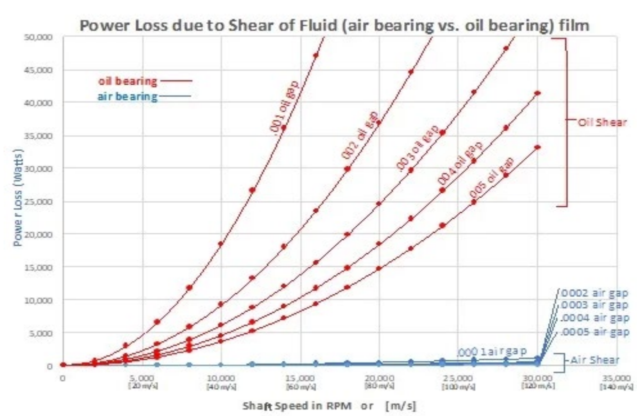

Figure 5: Power loss due to shear [Ref: Newway].

The theoretical impact of damping and stiffness is shown in Figure 6 and Figure 7.

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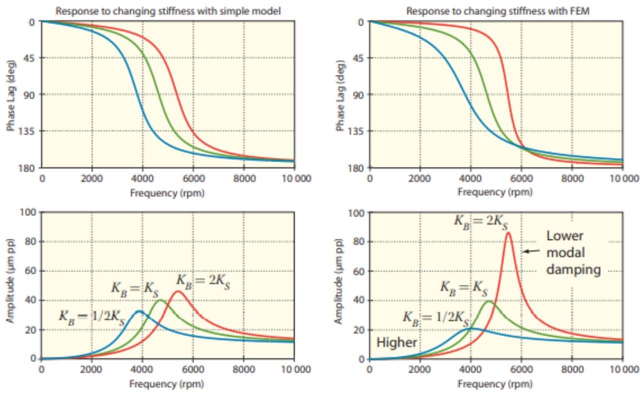

Figure 6: Impact of damping on vibration response. [Ref: Bently, Fundamentals of rotating machine diagnostics]

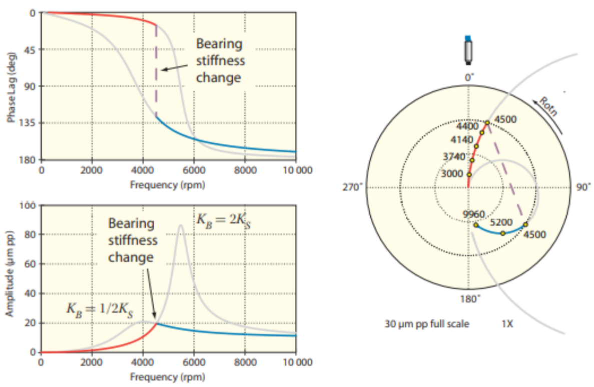

Figure 7: Impact of stiffness on vibration response. [Ref: Bently, Fundamentals of rotating machine diagnostics]

EPP bearings can also function as a seal by creating a narrow, high-pressure region between the high and low-pressure regions on either side of the bearing and are very easy to implement into most existing machines. They take only a slightly larger envelope than foil bearings, and only need a single gas feed line routed to the bearing.

2.2 EPP Bearing Acceptance Testing

A rig to characterize the TAC bearings ahead of delivery to Sandia was built using a simple rotor and housing construction. To eliminate the complexity of the PMG, the shaft was driven externally by a quill shaft, see Figure 8. The design permitted easy component modification and bearing change out. During tests the bearings were supplied with compressed Nitrogen over a range from 0 to 1500 psig. The bearings consisted of a 35.5mm journal on the Turbine/Non-Drive End and a combination 38mm journal and thrust flange on the Compressor/Drive End. These bearings are rigidly pressed into the housings to aid in studying the direct relationship between the bearing gap, pressure, and stiffness.

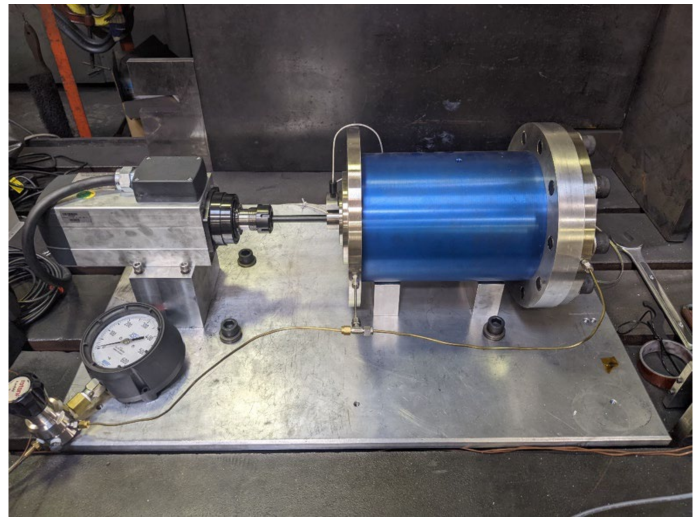

Figure 8: Acceptance test rig.

Flow testing was performed using a bank of high-pressure nitrogen tanks. total flow rates were recorded over a range of pressures from 20 to 1500 psig. The corresponding leakage flows are shown in Figure 9. The maximum flow observed was 60 standard cubic feet per minute (SCFM) at 1500 psig; at 2200 psig it is estimated to be 115 SCFM. With a 60 psi pressure difference required for operation, the flow to atmosphere on both ends was around 1.0 SCFM.

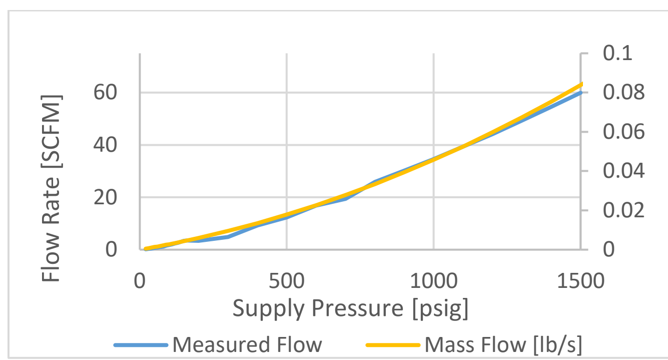

Figure 9: Bearing leakage measurements.

Venting rates were measured from 50 to 320 psig and are shown in Figure 10. An average 50% venting rate was observed through the motor cavity. The pressure in the vent rose less than 5 psi through the test when venting to atmosphere through a 1/8 in diameter tube. This is a significant decrease from estimated labyrinth seal leakage rate of 123 SCFM. [From Sandia Report SAND2010-0171].

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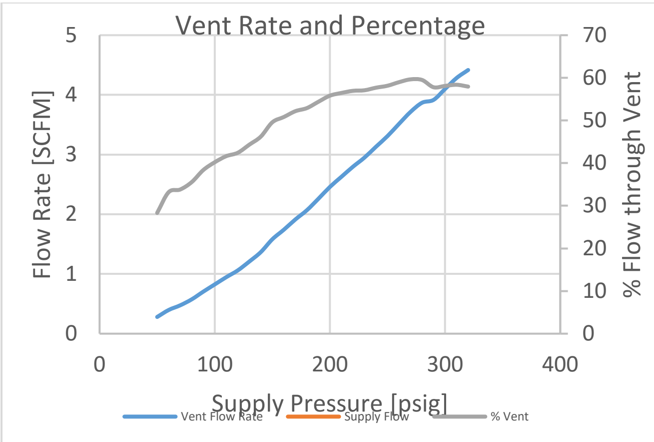

Figure 10: Vent flow rates.

Axial load testing was done with an air cylinder employing an air bearing pushing against the end of the shaft, see Figure 11.

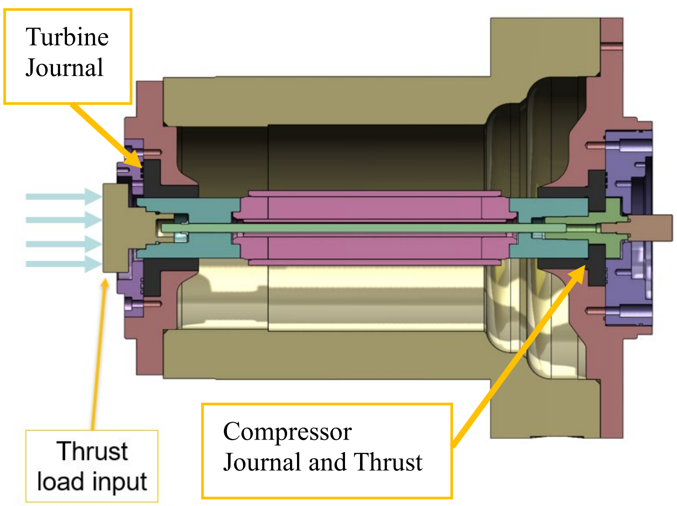

Figure 11: Test rig showing bearings and thrust input pad.

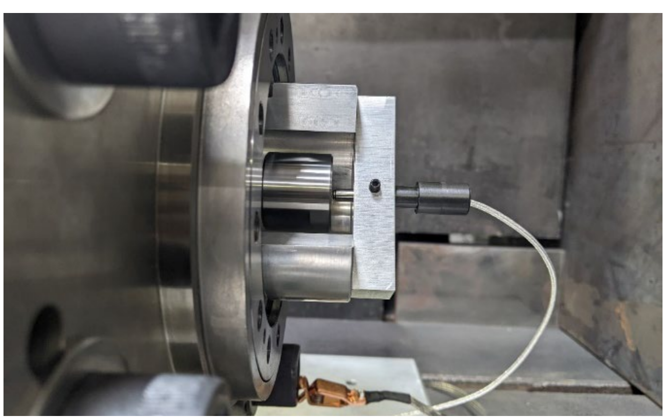

Figure 12: Capacitance probe mounting.

Axial load testing was done with an air cylinder and air bearing pushing against the end of the shaft. A capacitance probe, Figure 12, on the opposite end of the shaft measured the displacement of the shaft axially with varying loads.

The air cylinder has a piston diameter of 1.77 in and was pressurized up to 30 psi resulting in 53 lbf load [236N]. This test was performed at multiple pressures from 300 to 600 psig. The results are shown below in Figure 13.

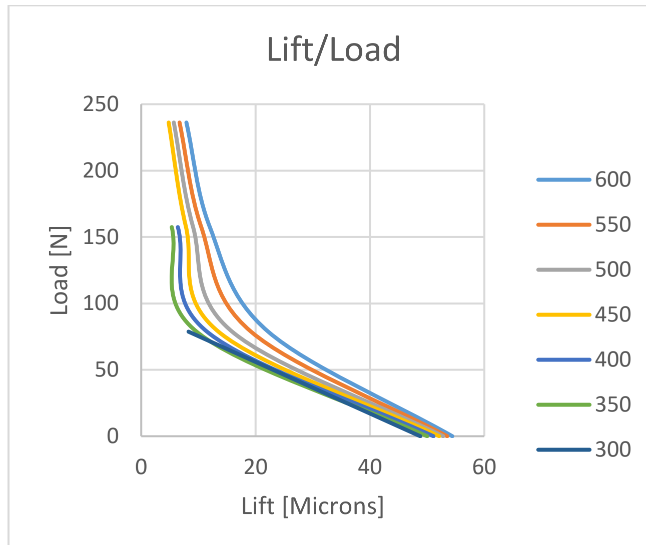

Figure 13: Thrust load test results.

Radial load testing was done by applying a mass load to the drive end of the shaft and measuring the lift at the turbine bearing once pressure was supplied to both bearings. The displacement was measured with a capacitance probe at the top of the shaft and is shown in Figure 14. The corresponding test results are shown in Figure 15.

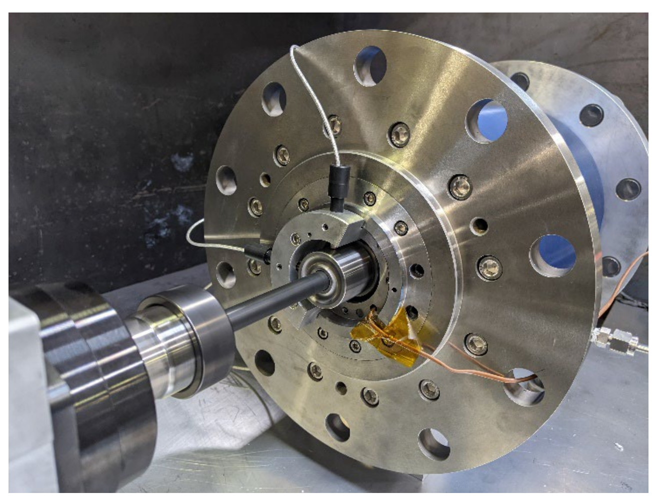

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Figure 14: Prox probe mounts for radial measurements.

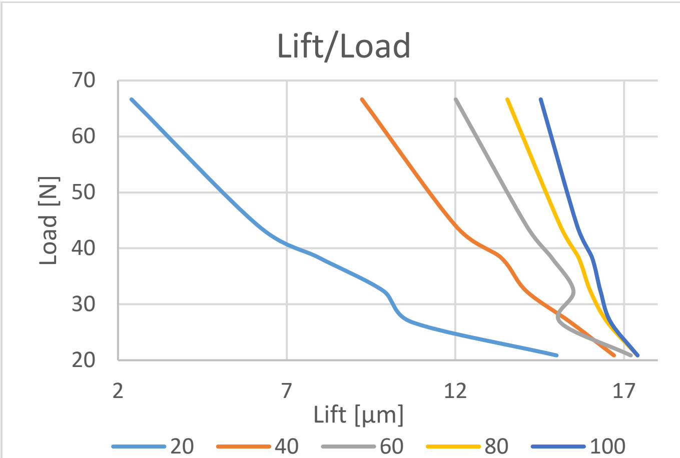

Figure 15: Static radial load tests.

High speed stiffness tests were performed on the rotor at 9,000 rpm. This included taking baseline measurements with the balanced rotor at 50 to 600 psig (see Figure 16).

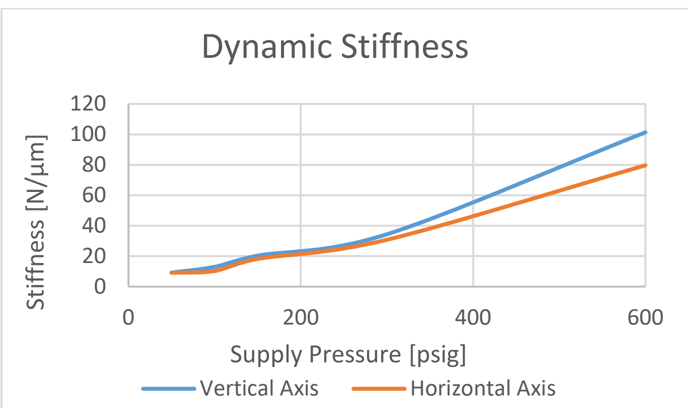

Figure 16: Dynamic Stiffness Measurements.

The Orbit Plots shown below in Figure 17 present the changes in orbit at 50, 150, and 600 psig. before and after adding an offset mass to induce synchronous imbalance forces. Measurements were retaken and compared to the baseline to quantify the stiffness and how it changes with increased supply pressure.

Note the change in orbit size and shape as the supply pressure increases. The increased pressure in the journal increases the stiffness in the bearing gap and reduces the eccentricity of the rotor due to the additional mass, offering a simple means to dynamically control eccentricity during operation.

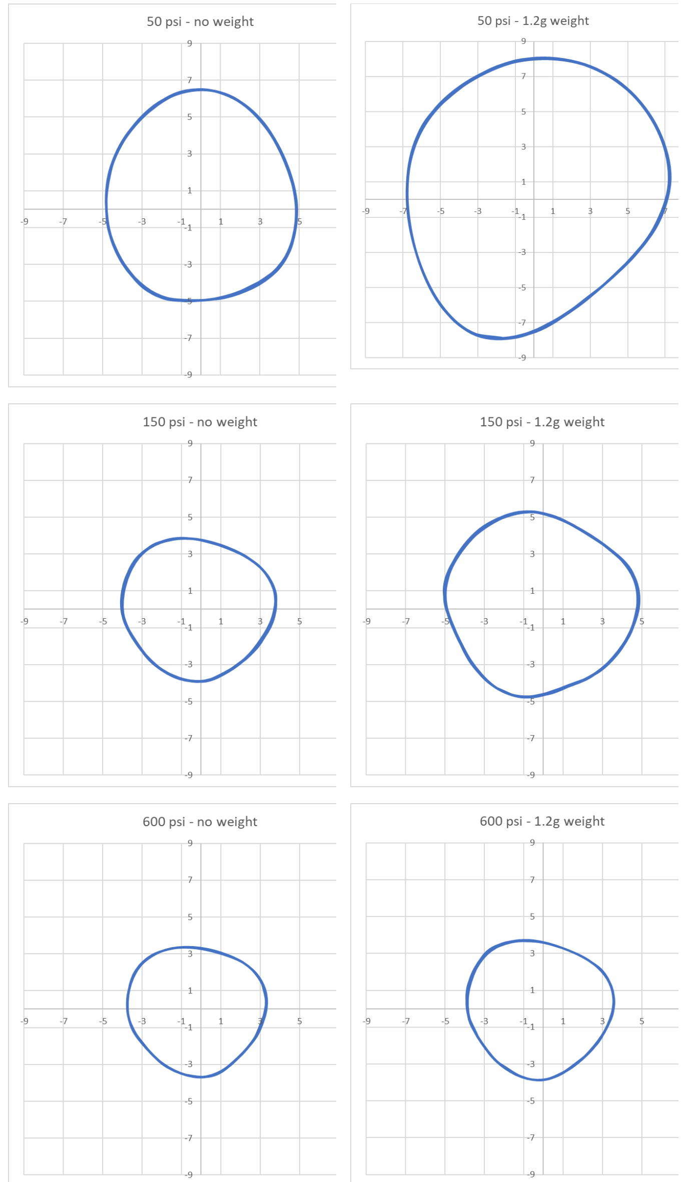

Figure 17: Orbit plots for balanced and offset mass.

Thermal tests to study heat buildup during sustained high speed were also performed. The system was run to 18,000 rpm with incrementally reduced supply pressure while monitoring the orbit plot and temperature. The results shown below in Figure 18, represent conditions after 30 minutes with a bearing supply of 100 psig. At 60 psig supply, there was no noticeable temperature rise after 40 minutes of continuous run time.

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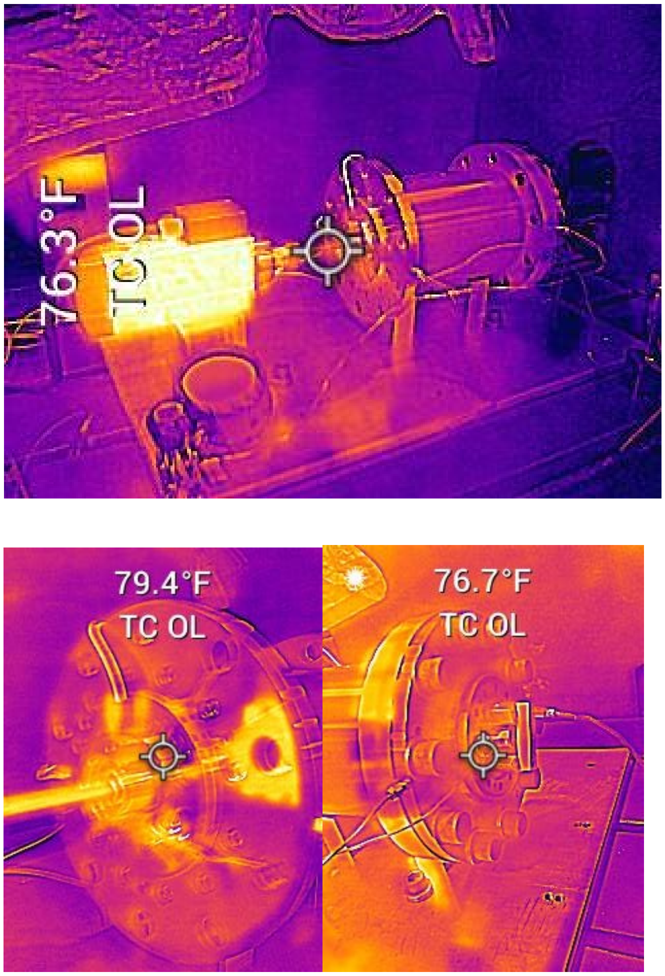

Figure 18: Thermal stability tests.

2.3 EPP Bearing Test Loop

The PMB test loop illustrated in Figure 19 is a closed loop system intended to maintain stable temperature and pressure supply conditions. The bearing leakage flow is captured and reclaimed through a recirculation loop using a hydropac compressor. The bearing supply pressure and the differential pressure across the bearings are independently controlled, while the rotor speed is also commanded independently. In an operational TAC the bearing supply would transition from an external supply to compressor delivery at a predetermined speed or pressure once the supply pressure at compressor discharge pressure (CDP) is sufficient and steady acceleration has been achieved.

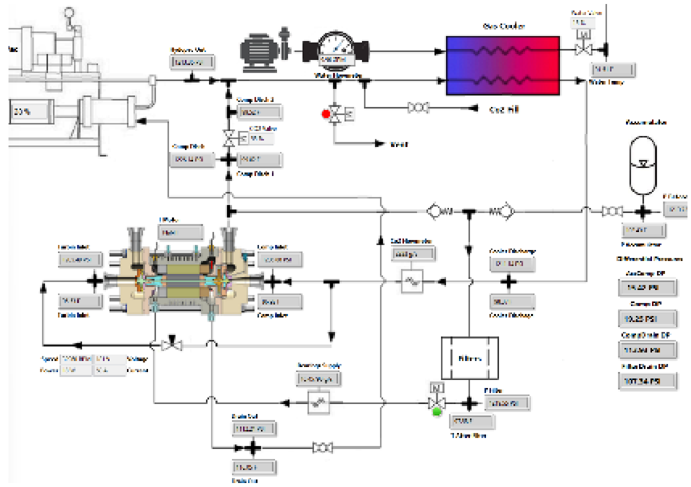

Figure 19: Bearing test rig P&ID.

3. EPP bearing initial test results

The initial porous media bearings test consisted of six main segments (as shown in Figure 20) labelled A through F:

1. Segment A –System startup to 30,000 rpm
2. Segment B – Maintaining 30,000 rpm
3. Segment C – Maintaining 33,000 rpm
4. Segment D – Maintaining 36,000 rpm
5. Segment E – Maintaining 40,000 rpm
6. Segment F – System Shutdown

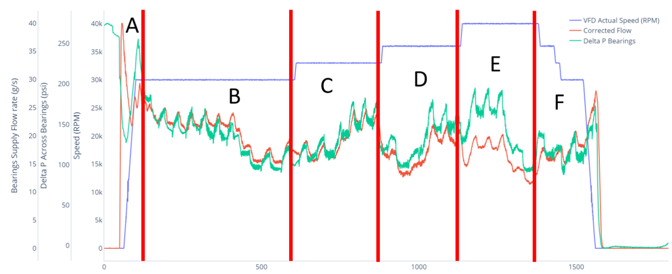

Figure 20: PMB test segments.

Figure 21 presents segment 'A' which demonstrates how the bearing flow was initiated before commanding the KEB motor controller to accelerate the bearings rig to 30,0000 rpm. The ability to pressurize and levitate the shaft while at rest overcomes one of the main concerns of the foil bearings; surface to surface contact until self-levitating speed is attained. In the trace below, the oscillation at approximately 10,000 rpm is due to the controller transitioning from speed signal to sensorless operation. Once the bearings rig was steadily spinning at 30,000 rpm the supplied bearing flow rate became more stable.

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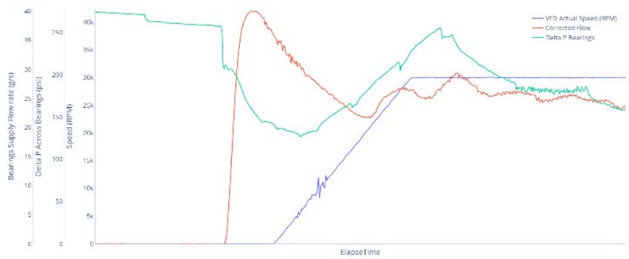

Figure 21: Segment A – system start up to 30,000 rpm.

In segment 'B,' the speed was maintained at 30,000 rpm. The bearings rig internal cavity pressure was reduced incrementally to determine flow rates for the bearing as a function of pressure differential. The differential pressure ranged from 175 psid to 100 psid and the results are depicted in Figure 22. The data quality throughout the tests were impacted by the pressure and flow oscillations caused by a positive displacement pump used to reintroduce the cavity leakage back into the circuit. However, it remains evident that the bearings supply flow is proportional to the differential pressure.

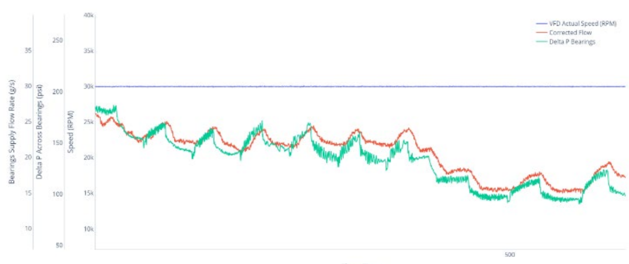

Figure 22: Segment B – Maintaining 30,000 rpm.

Segment 'C' consisted of keeping the differential pressure and flow rate the same while increasing speed from 30,000 rpm to 33,000 rpm. It is important to note during this transition to higher speed the bearings flow had no perturbations, which informs us that there was no contact to the bearing during this acceleration. While maintaining 33,000 rpm, flow measurements were documented by varying the pressure differential from 100 psid to 175 psid and are captured in Figure 23 below.

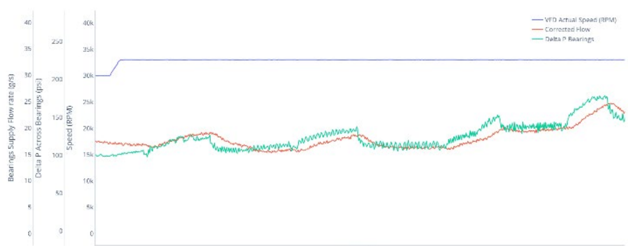

Figure 23: Segment C – Maintaining 33,000 rpm.

In segment 'D' the differential pressure and flow rate the same as before while the speed was increased from 33,000 rpm to 36,000 rpm. It was also observed that there was very little change in flow rate. While maintaining 36,000 rpm, the pressure differential was varied from 100 psid to 175 psid and the bearing flow rate recorded as shown in Figure 24. In previous turbomachinery tests there is a distinct audible signature change at 35,000 rpm which is believed to be a critical speed; no such indication was present for this test.

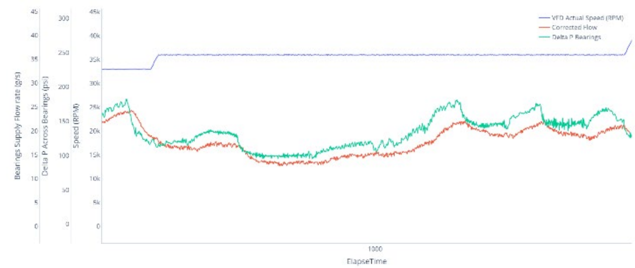

Figure 24: Segment D – Maintaining 36,000 rpm.

Segment 'E' consisted of keeping the differential pressure and flow rate the same as before while speed increased from 36,000 rpm to 40,000 rpm and is recorded in Figure 25.

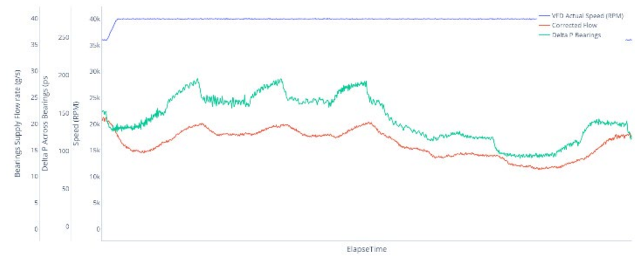

Figure 25: Segment E – Maintaining 40,000 rpm.

In segment 'F,' Figure 26, speed is ramped down to obtain data points at 36,000, 33,000 and 30,000 rpm. It was paused to ensure bearing temperatures were acceptable before commanding the machine to stop. Once the speed was steady at 30,000 rpm the test was ended and the rig commanded to stop. During the coast down to 0 rpm, flow to the bearing was maintained to ensure bearing robustness.

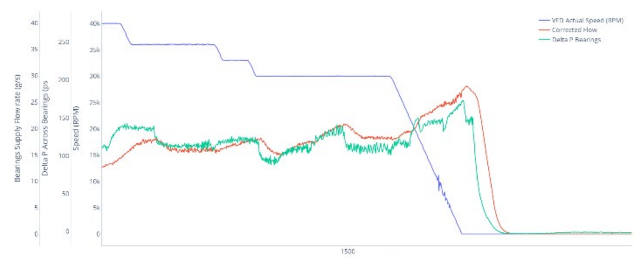

Figure 26: Segment F – System shutdown.

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4. CONCLUSION

The initial tests of the EPP bearing retrofitted into the TAC confirmed the technology is suitable for high speed sCO2 applications. During the acceleration and ramps to steady speeds, the rotor was well-behaved and exhibited no vibration or instability. The test also confirmed the control procedure required to ensure suitable conditions for reliability.

Of particular interest was the leakage flow. Over the range of pressure deltas during the test, the mass flow from both sets of bearings ranged from 10 g/s to a maximum of 25 g/s. The flow rate is controlled by the exit anulus from the bearing face and typically operates choked. The mass flow is therefore a function of the pressure and temperature only and at steady state conditions is insensitive to dP. This was a marked improvement over the labyrinth teeth seals, typically employed for high-speed rotating equipment.

While these tests demonstrate the performance of the EPP bearings for thrust and journals, they also demonstrated their capability to function as a seal with very low leakage losses. This is extremely important for small machines with integral motors, because windage losses around the motor are significant unless their cavities are maintained at low pressure.

Even with the poor flow regime from the hydropac, the EPP bearings performed well, demonstrated tolerance to instabilities, and allowed for smooth running and friction-free start stops. In these initial tests EPPs verified their potential commercial potential, providing support for further detailed characterization of the technology and the investigation of how their other benefits can be utilized.

Before continuing with testing the hydropac pulsation issue must be resolved. It is intended to suppress the pulsations by adding an accumulator to the supply line. This will permit a more rigorous analysis of the EPP bearing performance and provide better correlation of flow and pressure relationships.

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