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An Approach to Turbomachinery for Supercritical Brayton Space Power Cycles

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Abstract. Closed Brayton cycles using supercritical working fluids have the potential to reach high efficiencies at lower turbine inlet temperatures, using extremely compact turbomachinery. This general concept has many potential applications, including light-weight energy-dense electrical power systems for space propulsion and life-support. Hardware development for this technology has been led by Sandia National Laboratories, where supercritical CO_2 turbomachinery running on gas foil thrust and journal bearings is being operated. One of the primary challenges in supercritical fluid turbomachinery design has involved the development of high pressure strategies for bearings and seals; the bulk of this paper outlines a strategy that has been employed to address these issues, leading to successful testing of S-CO_2 power turbines. These turbines are hermitically-sealed and sized at 125kWe, therefore the approaches taken are commensurate with design concerns for hardware for space power systems.

Space power systems have different requirements than Earth-based systems. In particular, this is apparent in fact that CO_2 is favored as a working fluid in Earth's atmosphere due the proximity of its critical temperature (304K) to ambient conditions. In space, fluids with critical temperatures near 400K or higher are favored in order to maximize the average heat rejection temperature and minimize radiator mass. Still, the lessons learned from supercritical CO_2 apply, and designs developed are a good starting point for working with other supercritical fluids. Experience with CO_2 at Sandia has been generalized to show that seals and bearings performance depends on fluid pressure and density, above all. This study will summarize designs for bearings and seals technology for the Sandia supercritical CO_2 turbine, and in doing so, identify and propose solutions to the challenges for supercritical working fluids for advanced space power systems.

1. INTRODUCTION

The supercritical CO_2 (S-CO_2) closed Brayton cycle, which uses high pressure CO_2 as its working fluid, is an attractive alternative to conventional steam plants, offering the potential for better overall economics due to a higher electrical conversion efficiency and lower capital cost [Dostal, 2005]. Sandia National Laboratories (Albuquerque, NM, US) and the US Department of Energy (DOE/NE) are operating an S-CO_2 power cycle using two custom-built 125kWe microturbines [Conboy, 2012]. This test facility can be counted among the first and only power producing S-CO_2 Brayton cycles anywhere in the world.

Similarly, supercritical fluid power systems have the potential to make a large impact in space power and propulsion. The current state-of-the-art reference design for relatively large, high efficiency power systems (>100kWe) is the He-Xe Brayton cycle [Mason, 2001], [Middleton, 2013]. However, if designed appropriately, a supercritical fluid power cycle can best an ideal-gas Brayton cycle for a given turbine inlet temperature [Wright, 2007]. This is due to a higher density fluid inlet condition to the compressor, and lower resultant power consumption to achieve a given pressure ratio. Supercritical fluid turbomachinery is also typically much more lightweight. An equally attractive case for the supercritical fluid Brayton cycle is the prospect that good efficiencies of the high temperature He-Xe cycle can be matched with reduced turbine inlet temperatures. This may allow use of more standard, reliable materials in construction rather than experimental alloys. Sandia has proposed the supercritical C_4F_8 power cycle, which can attain efficiency greater than 29% with a turbine inlet temperature of only 825K, and a compressor inlet temperature of 390K [Wright, 2007]. C_4F_8 has a critical point of 388K, 2.78MPa. Hexane, with a critical point of 506K, 3.06MPa may be another candidate fluid for space applications.

In the course of the ongoing hardware development program for Sandia's supercritical CO_2 (S-CO_2) turbine, which first began in 2007, thrust bearings, radial bearings, and shaft seals have demanded a disproportionate amount of the overall effort. The high energy density of the S-CO_2 power cycle dictated that at 125kWe, each turbine and compressor wheel would be mere centimeters diameter. These small sizes in turn dictated that the turbine and compressor wheels would need to rotate at high speed, on the order of 75,000rpm, in order to attain good efficiencies (Figure 1).

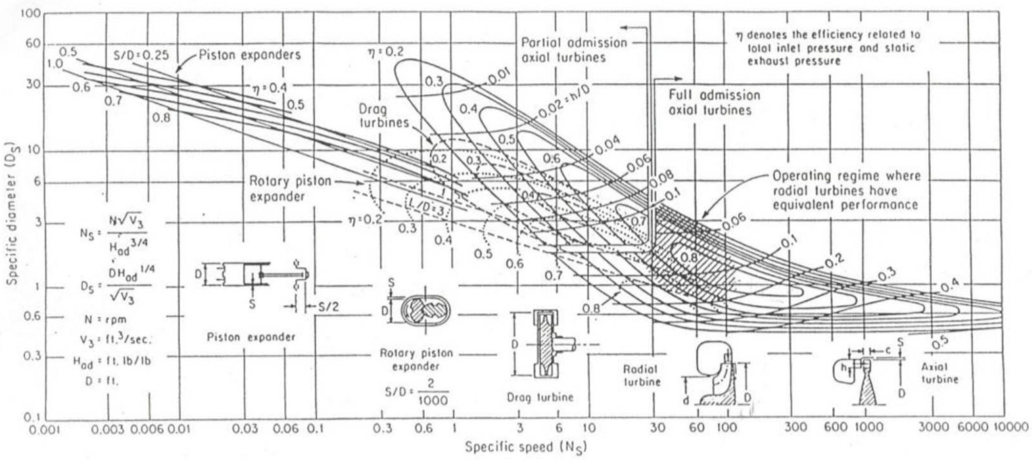

Figure 1: Ns-Ds Chart for Turbine Design

While this high speed requirement is not unusual for commercial microturbines which operate on air at ambient pressure, the S-CO_2 cycle poses another challenge in that the entire cycle operates at pressures ranging from 7-20MPa. Because the cycle also operates in the vicinity of the CO_2 saturation curve, its working fluid attains densities that are a significant fraction of the density of water (200-600kg/m^{3}s). This environment is prohibitive to the use of high speed rotating machinery because excessive irreversible losses that would be incurred. Therefore rotating seals are necessary to isolate the bearings and alternator from the rest of the system to reduce friction.

It is difficult to separate discussion of the high pressure seals from the thrust and journal bearings. Because the use of rotating seals in supercritical fluid environments is still an emerging technology, the approach taken for the Sandia S-CO_2 turbine was to build a fully hermetically-sealed system, utilizing the working fluid itself for bearings lubrication and cooling in an isolated cavity operating at reduced pressure. This requirement is reminiscent of power cycle hardware configuration for space applications, where shaft sealing to the environment is not feasible.

This design was achieved by building the compressor and turbine on a single shaft, with the permanent magnet rotor/alternator and bearings positioned in the low pressure region between them (Figure 2). The boundary of the low pressure cavity volume is defined by shaft seals at the back of the turbine and compressor wheels. Its reduced pressure is a function of effectiveness of the seals and the stroke rate of a hydraulic pump used to evacuate the cavity of CO_2. This seal leakage flow is then pumped back into the high pressure bulk flow at another point downstream.

The high working pressure, and small projected areas of the turbomachinery wheels, creates the potential for large thrust loads. Preliminary calculations have revealed that the compressor and turbine may have net forces exceeding ~4000N (1000 lbs-f) acting on each individually, but must come close to balancing (~400N/100lbs-f) in order to minimize demands on the thrust bearing. In addition to the steady state load demands, the thrust load applied by the turbomachinery has a high sensitivity to transient conditions at the turbomachinery, such as small spikes in pressure. These transients must be avoided or supported by the bearings. In short, the high speeds, extreme fluid conditions, and heavy loads have posed design challenges for small-scale supercritical fluid turbomachinery.

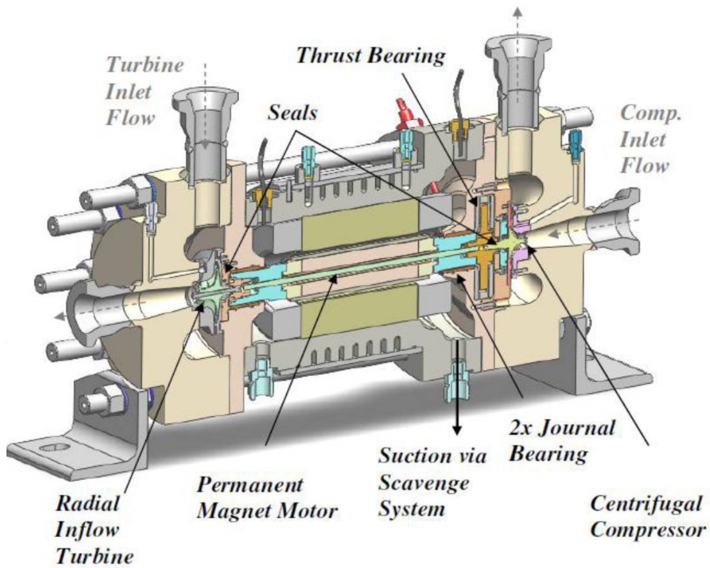

Figure 2: Schematic of the turbo-alternator-compressor.

2. SUPERCRITICAL FLUID FOIL BEARINGS

The unique challenges presented in thrust and radial load management and rotordynamic stability for the S-CO_2 turbine demanded a thorough consideration of all bearings options, including ball bearings, gas foil bearings, hydrostatic bearings, and even magnetic bearings. Primarily, it was clear that bearings would have to operate within a CO_2 environment – adequate rotating seals for S-CO_2 simply did not exist to allow bearings to operate outside the pressurized system. This eliminated more traditional approaches that would have isolated bearings from the process fluid, and used oil-lubricated bearings. In some sense, this simplified the system from dependency on cumbersome ancillary equipment for pumping, storing, cooling, sealing, plumbing, and filtering the oil lubricant. Oil also cannot sustainably lubricate a system at high temperatures, due to thermal breakdown.

Magnetic bearings would have required a more complex development effort, and were temporarily set aside from the decision. Ball bearings had been tested in S-CO_2 during the development process, but didn't perform well, requiring frequent replacement. Over time, the poor viscosity of CO_2 allowed the rolling elements to come into more aggressive contact than they would in oil, wear quickly and deform. It is also known from theory that their capacity would decrease with increases in operating speed.

This left gas foil bearings, which make use of the process working fluid to generate a hydrodynamic load capacity that increases with speed, eliminating system contamination from oil and enabling higher temperature operation. Compliance in the structural foils allows the bearings to deform under load for increased capacity, and to compensate for misalignment and distortion. Because of these features, foil bearings have proven to be an enabling technology for small to medium-scale advanced turbomachinery systems. Microturbines require high rotational speeds to attain good efficiencies, resulting in wear conditions that rolling element bearings cannot survive long-term.

In a typical configuration, gas foil bearings are composed of a smooth top foil and a corrugated bottom foil or 'bump foil'. Each is made from a thin sheet of compliant metal, typically a nickel-based alloy such as X750. The top foil is affixed to the bearing housing at a point, and is initially allowed to sit to the height of its bump understructure. During loading and running, the foils are perturbed by the film pressure profile and resultant deflection of the foil bearing. Foil bearings have dual mechanisms for imparting stiffness and damping to a rotordynamic system. The first is of course based in hydrodynamics, resulting from the fluid film between the runner and top foil and its converging profile. Unique to compliant bearings, a second mechanism results from the structural shape and materials properties of the top and bottom foils. These combine to provide a spring support. Because of this compliance, foil bearings are non-linear structurally and hydrodynamically, and modeling requires the coupling of multiple systems of equations in order to evaluate these simultaneous solutions [Conboy, 2012].

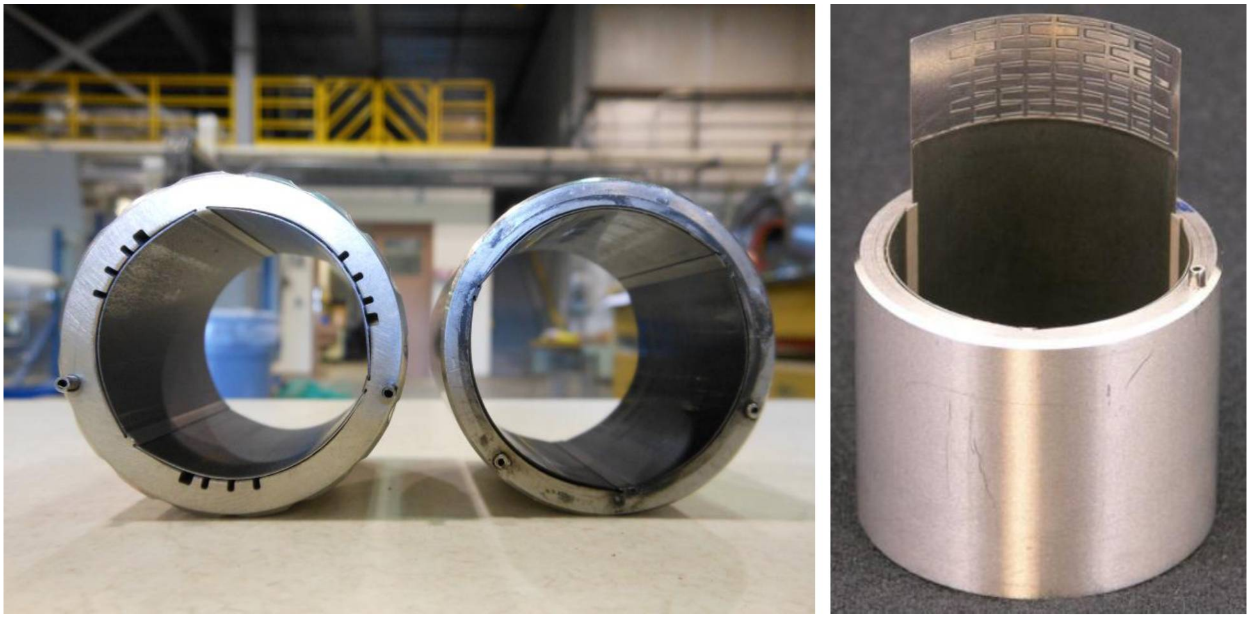

Figure 3: (**L**) High and low temperature radial bearings used for the S-CO_2 turbine, (**R**) Understructure of a radial bearing.

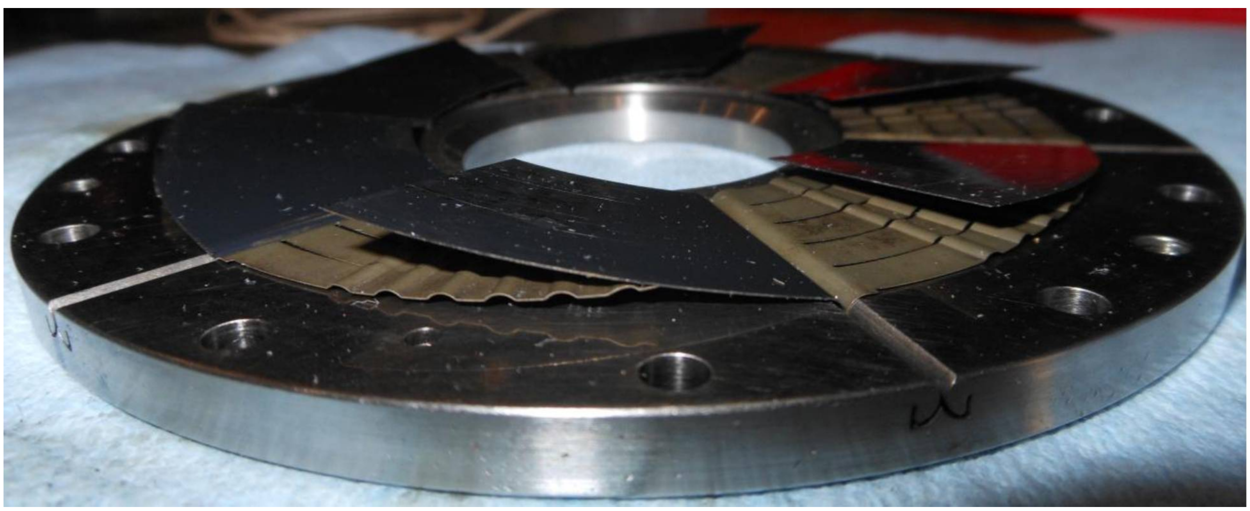

Figure 4: Photo of the gas foil thrust bearing used for S-CO_2 turbine.

The coupling of gas bearings with turbomachinery is but an emerging practice: only a small amount of performance data is available in literature. Industry has been slow to broadly adopt this technology due to the relatively small number of manufacturers and therefore a high perception of risk. However, in recent years, NASA Glenn Research Center had sought to encourage more R&D by conducting studies to design and test open-source gas foil thrust and journal bearings [DellaCorte, 2007], [Dykas, 2010]. The limited experience for foil bearings was by and large only in ambient pressure air.

DellaCorte and colleagues also established a rule of thumb for foil bearings support in air: approximately 1 lb/in^2 of projected area, per inch of diameter, per 1000 rpm [DellaCorte, 2000]. Therefore the S-CO_2 turbine bearings diameter and length were initially sized based on projections for air. The higher density of CO_2 at elevated pressures meant that this is would likely an underestimate of load capacity performance; still, these estimates were used for a first-order design. The journal bearings were obtained off-the-shelf from Capstone, pictured in Figure 3, though explicit dimensions are subject to export control. The right-hand side of provides a look at the 'finger-like' bump understructure used in both journal bearing types.

The bearing in the left hand image of Figure 3.6 is Capstone's 'high temperature' foil journal bearing, and the right hand image is the 'medium' temperature bearing. They differ mainly in the solid lubricant coating used on the foil pads. A good solid lubricant layer is critical for protecting foil bearings in startup and shutdown, when the turbine crosses through speeds too low to support a hydrodynamic CO_2 film. The high temperature turbine side journal bearing uses a patented Capstone coating with a temperature limit near 1000°F; this is required as the turbine end leakage flow may reach similarly high temperatures when the system reaches its high temperature design point. The compressor end journal bearing uses a Teflon-based coating, capable of reaching temperatures of 450°F. This is more in line with the low temperatures expected at the compressor end leakage flow.

The design uncertainties presented for foil journal bearings were minor in compared to the foil thrust bearings. No commercial manufacturer could be found to deliver foil thrust bearings; therefore they were designed and built in-house at Barber-Nichols from open source information provided by NASA Glenn. Based on NASA's experience, a foil thrust bearing was developed featuring 6 pads (Figure 4). Pads were separated from one another by 15 degree gaps for enhanced thermal management. Details of the geometry are considered export controlled information. Foils were made from Inconel X750, and a coating of Teflon lubricated the top pads for smooth transition to high speeds. Two foil assemblies were created with mirror-image geometry, and sandwiched a thrust runner disk, to complete a bi-directional thrust load assembly.

Theory for gas foil thrust bearing operation is complicated and not widely applied beyond a few experts in the field. Very little performance data is available that can be attributed to a known geometry. The exception to this comes from experience at NASA Glenn in ambient pressure air, not dense CO_2. However, a first-order design was put together by again applying the radial air bearings rule of thumb (1 pound per inch-squared of projected area, per thousand rpm).

After assembly of the turbine with foil bearings, extensive testing was carried out in a high pressure CO_2 environment. The most significant difference between air and high pressure gas was found to be the presence of turbulence in lubricating films at high pressures. Density for CO_2 at 1.4MPa (200psi) is still an order of magnitude higher than for air at ambient conditions, and viscosity is measurably lower. This combines to give higher Reynolds numbers for a given rotation rate and film thickness. Turbulence gives rise to increased load capacity, but also higher frictional losses. Friction is largely concentrated at the thrust bearing assembly, giving rise to high fluid temperatures and a need for several measures of thermal management, including bypass cooling flow and advanced thrust pad geometry. The presence of turbulence also yields a strong sensitivity of power loss to speed changes or environmental changes within the turbine housing. A suite of codes and tools, validated by performance data, has been developed by Sandia National Labs to assess bearings performance and turbine thrust loading [Conboy, 2012].

3. SUPERCRITICAL FLUID SHAFT SEALS

Supercritical fluid seals exist for commercial scale systems, with suppliers including John Crane, Inc., for example. These are typically film-riding face seals such as dry-gas liftoff seals, and include multiple stages and buffer gas management systems. However for small scale high speed turbines, supercritical fluid seals did not exist prior to this program – this technology required development from the ground up.

Furthermore, it was planned to operate the bearings and alternator within a reduced pressure cavity internal to the turbine, isolated from the supercritical pressures within the main flow paths of the system. Shaft seals would limit CO_2 flow into this cavity, which would be pumped back into the bulk system flow further downstream. This seals leakage flow would provide lubrication and cooling to the gas thrust and journal bearings (Figure 2). Designs sought to limit this rotor cavity pressure to 1.4MPa (200psi) despite pressures of 7.6MPa (1100psi) and 13.8MPa (2000psi) facing the compressor and turbine wheels respectively. An equally important objective was to limit the leakage flow to less than 0.5% of the total system flow, in order to maximize motive power of the CO_2 through the turbine. The seals also needed to be able to withstand temperatures of 340K/150°F on the compressor side and 810K/1000°F on the turbine side.

Various seals types of shaft seals were considered from a perspective of minimizing leakage rates while simplifying the technology development path. Gas lift-off seals are in general much better at limiting flow than labyrinth seals, though are very complex and haven't been commercially developed for small high speed systems. Ultimately labyrinth seals were selected, which are much simpler in terms of their geometry and from the standpoint of integration into the system. The remaining effort was concentrated on selection of materials and geometry to maximize performance.

The earliest design attempt featured a stepped labyrinth seal with a smooth rotor, and toothed bronze stator. The seal was designed to give a small clearance between the rotor and stator, but after several cycles, this was worn down to several times the initial separation. Leakage through the seal was much larger than expected, on the order of 0.1kg/sec, and the pressure downstream of the seal was in excess of 4.8MPa (700psi) for a supercritical condition upstream (Figure 3). At this gas pressure, the friction generated at only 25,000rpm was immense, and temperature sensors mounted in the fluid near the bearings recorded high temperatures.

The design shifted to abradable materials and geometry. This is the concept of configuring the rotor and stator with essentially zero clearance, and allowing the rotation of the toothed shaft to wear custom channels into the smooth stator. This allows for minimal flow area in sealing. Various abradable materials were initially tested, including Fluorosint 500, Graphitar 80, and Graphitar 86. Fluorosint is a robust plastic material which showed signs of promise in early testing. At low speeds, the leakage flow rate had been decreased by more than an order of magnitude and low pressures achieved downstream of the seal. However as speeds approached 50,000rpm, the Flurosint melted and disintegrated from excessive seal-face friction. Graphitar became the next candidate material; it has improved solid lubricant characteristics and is rated for higher temperatures. Graphitar 80 and 86 both showed promise, but vibrations in the bearings housing caused the more brittle graphite based seals to chip and crack during operation, destroying the fragile foil bearings. This occurred in subsequent tests during which the graphitar material was allowed to freely rotate within its housing, and a later test when the seal was tightly fixed within its housing. A more ductile polyimide material (Vespel-SP22, 40% graphite) was also tried, but partially melted and otherwise showed little promise.

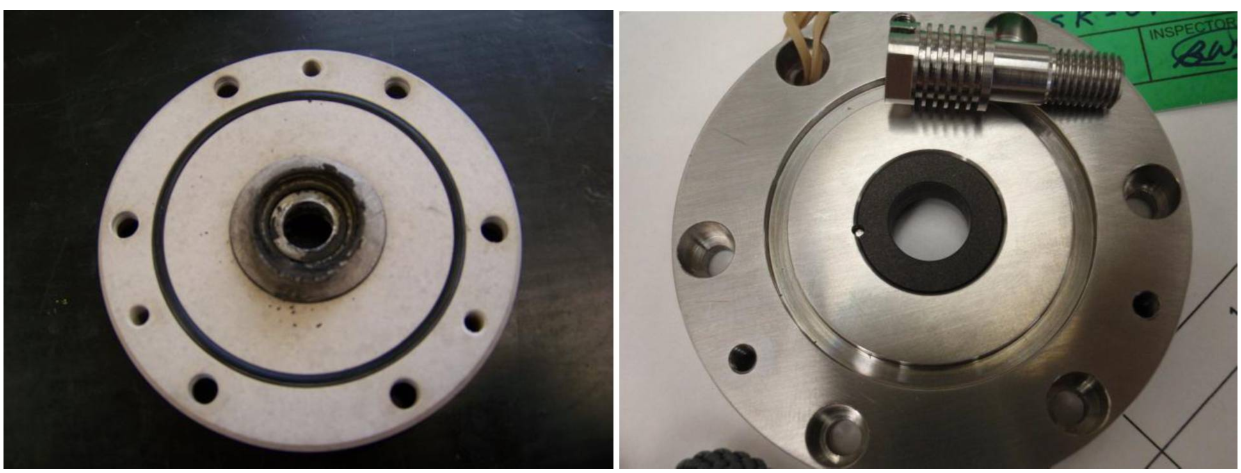

Figure 5(a) Early Fluorosint seal concept, destroyed during testing **(b):** Final version of the 'Floating' Graphitar seals technology, fixed by a single pin.

The final innovation was a return to the abradable Graphitar material, but using a single pin to fix the seal within the housing. This 'floating' seal concept allowed a degree of freedom during operation about the fixed point, but prevented the stator from adhering to the seal and causing it to rotate and vibrate in its housing. A more robust high temperature graphite material, Graphitar 3030 was also used, and a slight change in geometry to a stepped seal. The resulting seal has been spun to high speeds in excess of 65,000rpm, at turbine inlet temperatures approaching 810K/1000F. More importantly, this design achieves compatibility with foil bearings, reduces seal leakage to 0.01kg/s and can drop pressure downstream of the seals to 1.4MPa (200psi), minimizing frictional losses (Figure 4). This makes for a stark comparison with the original non-abradable seals design shown in Figure 3.

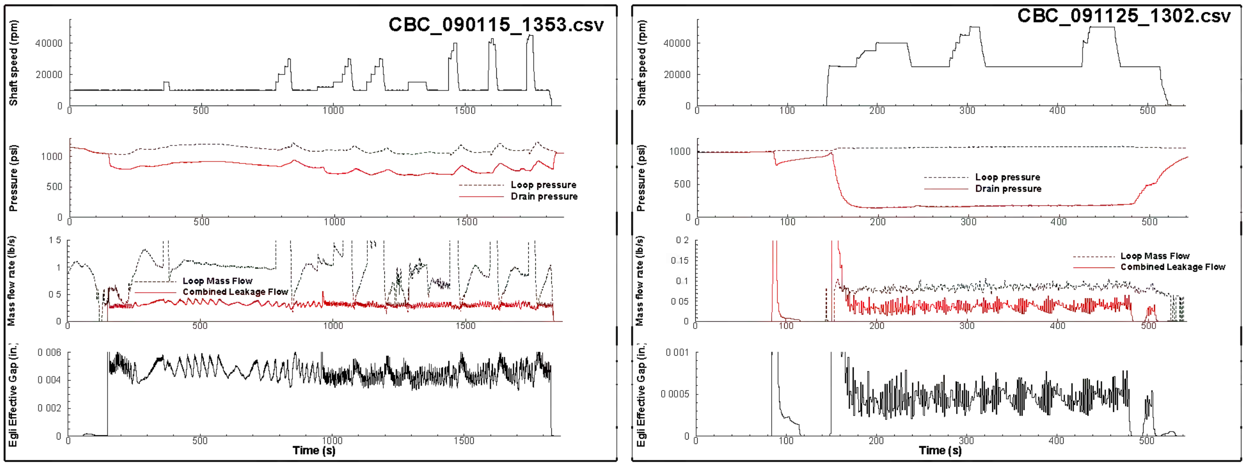

Figure 6: Labyrinth seals performance data, initial design: bronze stepped labyrinth seals, non-abradable, straight rotor, toothed stator)

In the end, at least 16 different seals geometries and materials were tested before a satisfactory approach was found, ultimately using high temperature abradable stepped labyrinth seals to both limit leakage flow and to allow compatibility with gas foil bearings in a high pressure CO_2 environment. A patent application was filed for this new technology [Wright et al, 2011].

For other supercritical fluids, including C_4F_8, this final seals design directly applies. Leakage is correlated with upstream density, seals flow area, and the pressure difference across the seal. Any supercritical fluid power cycle is sure to have densities facing the seals of several hundred kg/m^3, therefore minimizing flow area by using abradable seals is a necessity.

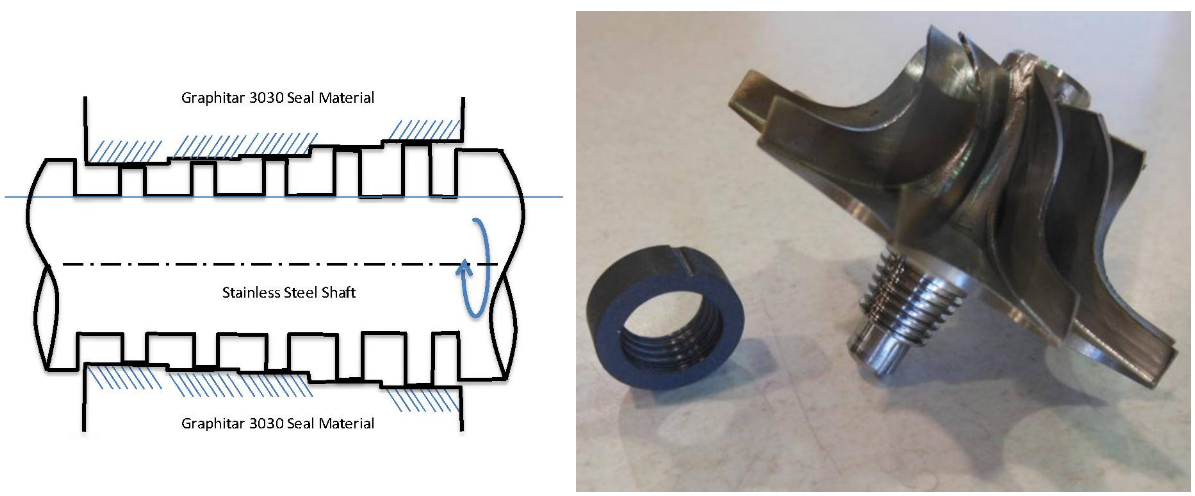

Figure 8 (a) Schematic of the stepped labyrinth seal concept. **(b):** Abradable graphitar 3030 seal, pictured with a 125kWe S-CO_2 turbine wheel.

4. CONCLUSIONS

In summary, the primary challenges to progress in S-CO_2 turbomachinery have been in bearings and seals technology. This required devising high speed supercritical fluid shaft seals to allow the bearings to operate in a reduced pressure environment, then creating compatible bearings capable of adequate load capacity with minimal frictional losses. Acceptable seals performance was achieved using abradable Graphitar 3030 stepped labyrinth seals in a semi-fixed 'floating' housing. This configuration combined survivability at high temperatures, good seal solid lubricity at the seal face, compatibility with fragile foil bearings and minimized seals leakage flow area. This system utilizes two commercially-available radial bearings; one near the turbine-end rated for high temperatures, and one at the compressor-end. The difference in their temperature rating is predominantly the solid lubricant coating used on the top foil surface to protect the bearing during start-up and shutdown. A gas foil thrust bearing was assembled based on experience in ambient pressure air.

Extensive testing of the foil bearings assembly in high pressure CO_2 has yielded a wealth of information and scaling law for performance. Most critically, sensitivity of load capacity and power loss to fluid conditions at the bearings have been established. This bearings and seals approach is the heart of the design for a supercritical CO_2 turbine operating at Sandia which can reduce the pressure in the bearings environment below 1.4MPa (200psi), and has recently achieved speeds exceeding 65,000 rpm without excessive power losses or thermal damage to the foil bearings.

The lessons can be generally applied to any supercritical fluid power cycle. In particular, the small scale, closed-cycle, hermitically-sealed turbine design presented here is ideal for space power systems. For space applications, the fluid C_4F_8 is predicted to outperform many other power cycle concepts, including the He-Xe Brayton cycle. For a generalized supercritical fluid turbine performance of seals is based on their ability to minimize leakage flow; performance of bearings is dictated by their load capacity, stiffness, and power loss. These characteristics are largely tied to density of the working fluid. For seals operating in any supercritical fluid (ex. C_4F_8) high densities at the compressor and turbine inlet (200-600kg/m^3) impact high flow rates if leakage area is not minimized. Downstream of the seals, elevated operating pressures persist on the order of 25-50kg/m^3, resulting in turbulence within the lubrication layer for small high speed turbomachinery, and sharply increasing sensitivity of load capacity and power loss to environmental changes.

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