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TURBOMACHINERY OVERVIEW FOR SUPERCRITICAL CO_2 POWER CYCLES

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ABSTRACT

Supercritical CO_2 (sCO_2) power cycles offer high efficiency and power density relative to the incumbent Steam Rankine and Air Brayton cycles for power generation over a wide range of applications, including waste heat recovery, concentrating solar power, nuclear, and fossil energy. One significant advantage of sCO_2 cycles over other Brayton cycles is that the high fluid density results in very compact turbomachinery for compression and expansion. This compactness reduces material costs and is also beneficial in low-space and potentially low-weight applications. The combinations of pressure, temperature, and density in sCO_2 power cycles are outside the experience base of existing turbomachines such as gas turbines, steam turbines, and even high-pressure gas compressors, and sCO_2 turbomachinery design is a significant challenge for realizing these cycles. This tutorial provides a brief overview of sCO_2 cycles and describes the resulting operating requirements and design concepts for the pumps, compressors, and expanders required for various cycle configurations. An overview of existing prototype turbomachinery in various laboratory facilities is provided along with a review of turbomachinery configurations and designs in the literature for various applications. Details regarding challenges common to most sCO_2 turbomachines including rotordynamics, pressure containment, sealing, and transient/off-design operation are presented with a description of specific components including bearings and seals. Turbine- and compressor-specific challenges including thermal management, overspeed risk, aerodynamic performance, and range requirements are also discussed.

INTRODUCTION

Supercritical CO_2 (sCO_2) cycles are proposed as a potential alternative to the steam Rankine cycle, which dominates electric power generation throughout the world due to a number of favorable characteristics including >100 years' experience, mature equipment designs, water availability and favorable fluid properties. Typical steam Rankine cycles have a thermodynamic efficiency near 40%, although this is dependent on the maximum cycle temperature and can increase significantly for state-of-the-art ultra-supercritical steam cycles. Steam cycles are used in a wide variety of power generation applications, including fossil fuels, nuclear, solar, and waste heat recovery. Analysis of various sCO_2 cycles indicate that they can outperform steam Rankine cycles by ~5-10 points in thermal efficiency (see Figure 1) for all of these application areas, particularly at heat source temperatures above approximately 450 °C for an sCO_2 Recompression Brayton Cycle (RCBC) (DOE, 2015). The RCBC or other sCO_2 cycle implementations can be advantageous over steam at even lower heat source temperatures depending on heat rejection temperature and overall cycle duty.

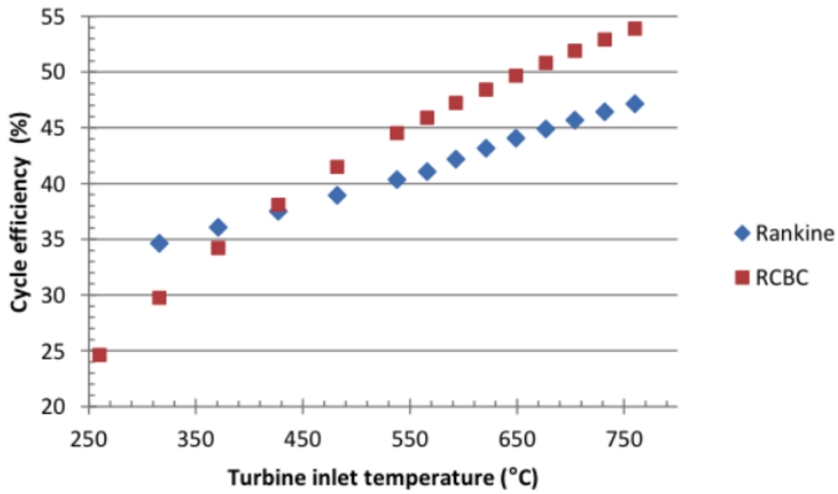

Figure 1. Efficiency Comparison of Steam Rankine and sCO_2 Recompression Brayton (RCBC) Cycles (DOE, 2015)

In addition to improved efficiency, sCO_2 cycles offer a significantly more compact power block than the steam Rankine cycle (Brun *et al.*, 2017). A size comparison of similarly rated steam and sCO_2 turbines is shown in Figure 2. In addition to compact turbomachinery, high-performance compact heat exchangers are also an enabling technology for compact sCO_2 power cycles. Although they are not discussed in detail in this paper, compact heat exchanger designs typically considered for sCO_2 applications include printed circuit, micro-tube, compact plate-fin, and other emerging design concepts. The smaller equipment sizes result in a smaller plant footprint and potentially a lower capital cost than a comparable steam Rankine cycle.

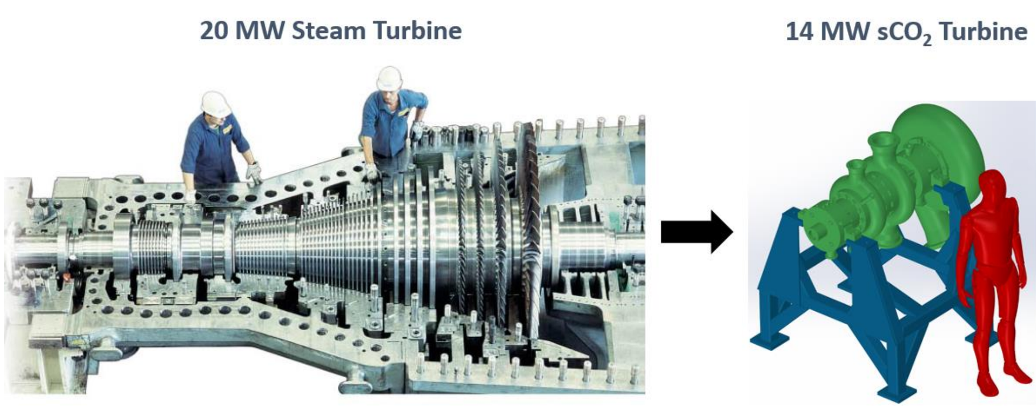

Figure 2. Size Comparison of 20 MW Steam Turbine with 14 MW sCO_2 Turbine [Adapted from (Brun *et al.*, 2017)]

The realization of robust, high-efficiency sCO_2 power cycles relies on efficient turbine and compressor designs that can operate reliably in a compact, high power density, high-pressure environment. This tutorial paper provides an overview of turbomachinery design attributes and challenges for sCO_2 cycles, and is organized into five primary sections. The first section introduces properties of supercritical CO_2 and the various cycle architectures most commonly considered in the literature. Next, existing and proposed sCO_2 machinery configurations are reviewed, followed by a discussion of attributes and components that are common to most sCO_2 compressor and turbine designs. The final two sections describe design considerations that are specific to sCO_2 compressors/pumps and turbines, respectively.

SCO_2 PROPERTIES AND CYCLES

An sCO_2 power cycle is any power cycle that utilizes CO_2 as working fluid at conditions above the critical pressure (73.9 bara/1071 psia) and temperature (31.1 °C/88 °F). Above these conditions, CO_2 is a supercritical fluid having gas-like behavior but with high fluid densities. Some sCO_2 power cycles operate entirely above the critical point, avoiding the two-phase region of the fluid, although significant variation in fluid properties will still occur with small pressure and temperature perturbations when operating at conditions near the critical point. Other condensing sCO_2 cycles operate both above and below the critical point. sCO_2 is a good candidate for a closed loop power cycle due to its wide availability, inertness, and near-ambient critical temperature for heat rejection near the critical point.

The most basic sCO_2 cycle with moderate efficiency is the simple recuperated Brayton cycle, shown in Figure 3, which involves a compressor, turbine, heater, and cooler as well as a recuperator for transferring turbine exhaust heat to the compressor exit stream. Compression takes place near the critical temperature, where fluid properties allow compression work with minimal work input. The temperature-entropy diagram illustrates several characteristics that are common across many sCO_2 power cycles: (1) the high amount of recuperated heat transfer in the cycle, (2) the high turbine exit temperature, and (3) the close proximity of the compressor inlet to the vapor dome.

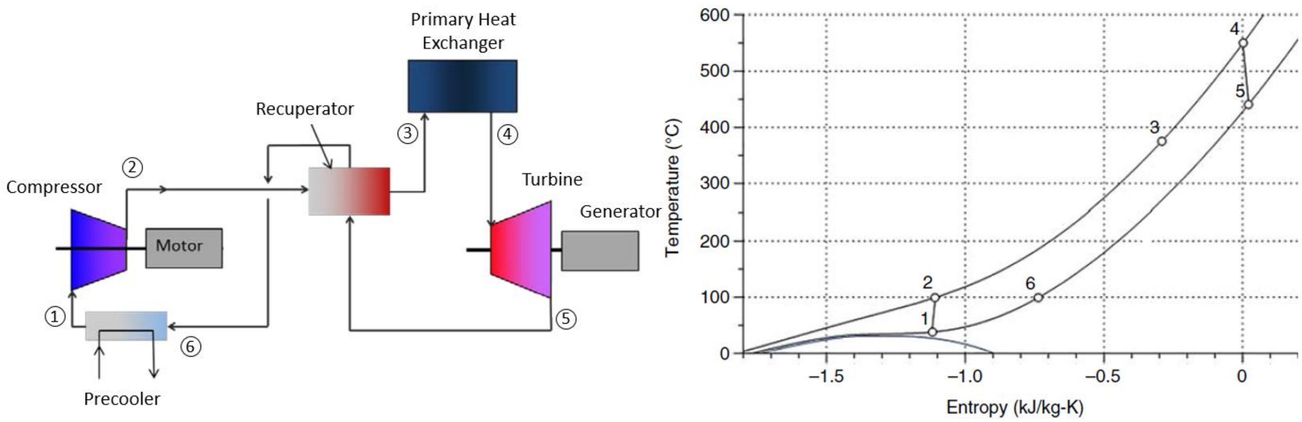

Figure 3. Simple Recuperated sCO_2 Power Cycle

Due to the relatively low efficiency of the simple sCO_2 Brayton cycle, most studies focus on the recompression Brayton cycle, which maximizes cycle efficiency at moderate temperature by introducing a second recuperator and compressor in order to minimize the detrimental effects of differences in heat capacity in the recuperator fluid streams. Thermodynamic efficiencies of an sCO_2 recompression cycle can meet or exceed 50% depending on the heat rejection temperature and component performance. An example high-temperature recompression cycle and representative conditions are shown in Figure 4 and Table 1, respectively.

Other cycle variations also exist for tailoring cycle performance to different applications and conditions. These include condensing cycles, which reduce the low-side pressure to below the critical pressure and incorporate a condensing precooler. Condensing cycles typically have lower efficiency than the recompression cycle but also lower mass flows and significantly reduced heat transfer duty and potentially lower overall cost relative to the recompression cycle. For WHR, more complex partial heating or cascaded cycles are often considered that maximize power generation for the lower heat source temperatures. Other work includes mixtures of sCO_2 with other fluids in order to shift the critical point and potentially improve efficiency.

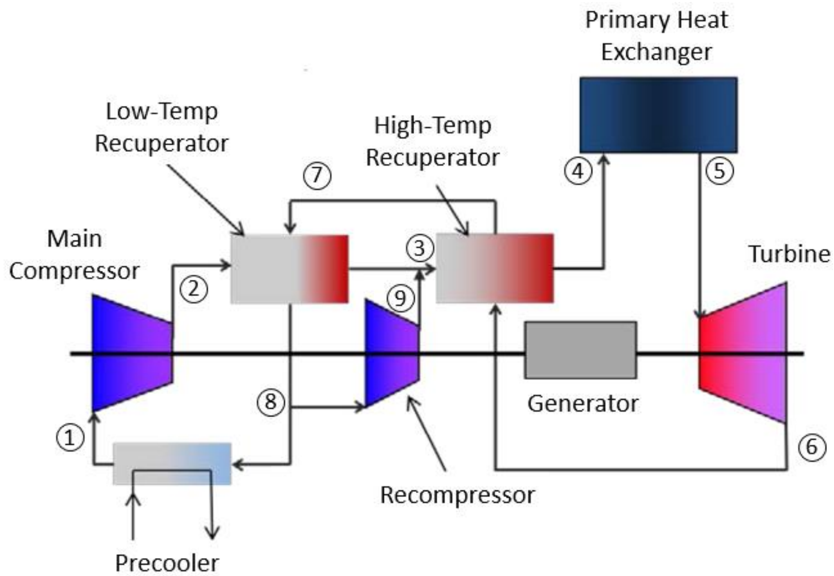

Figure 4. sCO_2 Recompression Power Cycle

Table 1. Example Cycle Conditions for a High-Temperature sCO_2 Recompression Cycle

Location	ID	Temperature (°C)	Pressure (bara)
Main Compressor Inlet	1	35	85.5
Main Compressor Exit	2	78	241.3
High-Temperature Recuperator HP Inlet	3	194	239.9
Primary Heat Exchanger Inlet	4	533	238.6
Turbine Inlet	5	700	237.2
Turbine Exit	6	581	89.5
High-Temperature Recuperator LP Exit	7	204	88.3
Recompressor / Precooler Inlet	8	88	86.9
Recompressor Exit	9	194	239.9

SCO_2 MACHINERY CONFIGURATIONS

Due to the high fluid density, sCO_2 turbomachinery designs are generally small and operate at high rotational speeds, particularly at relatively small power block scales. Designs have been presented in the literature for a variety of applications ranging from tens of kW to 1000 MW-scale power blocks. A detailed review of existing turbomachinery prototypes and designs has been published by the authors in Brun *et. al.* (2017). A summary of published sCO_2 turbomachinery speeds and scales from various sources in the literature is provided in Figure 5. Cases shown at 3 MWe and below incorporate high-speed turbines, compressors, and generators that generator operate at high speeds where power electronics are required to convert the power to grid frequency. Higher-power cases between approximately 7-50 MWe all use gearboxes between the power turbine and generator, and cases above 50 MWe have a synchronous power turbine. In most cases (where green and red circles overlap), a single turbine drives both compressors and the generator, but other layouts (with non-overlapping green and red circles) incorporate a split-shaft design with a separate higher-speed turbine to drive the compressor.

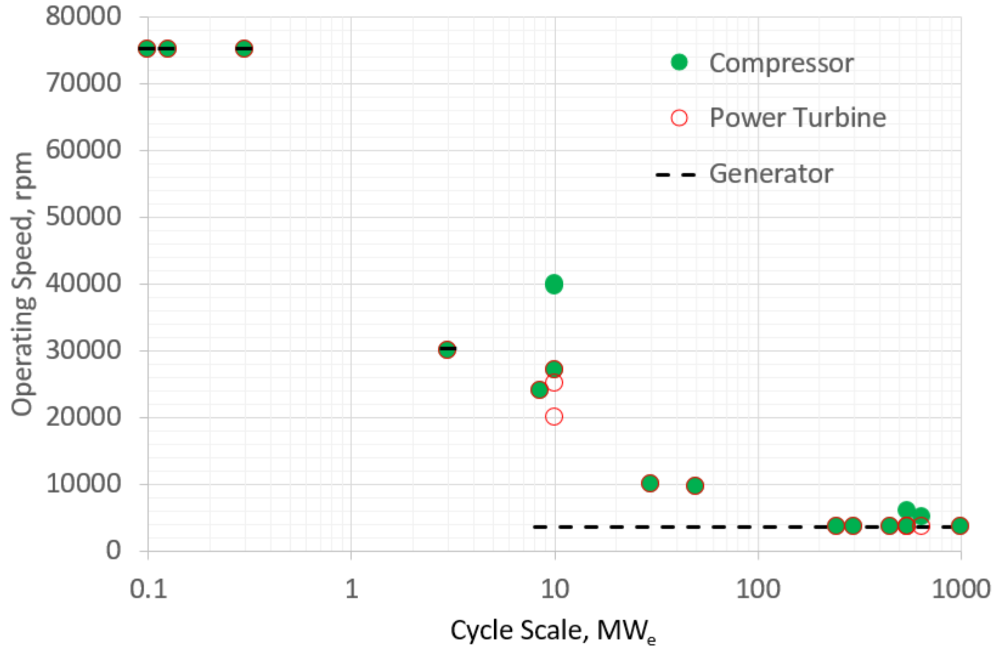

Figure 5. Summary of Published sCO_2 Turbomachinery Speeds and Scales [Adapted from (Brun *et al.*, 2017)]

At most scales, the compressors in sCO_2 turbomachinery designs are centrifugal compressors, although there is some overlap with axial compressor designs at the multi-hundred MWe scale. Note that this paper uses the terminology "compressor" to represent both pumps and compressors. The distinction between pumps and compressor in sCO_2 applications is not distinct since the fluid densities are high for liquid, gas, and supercritical fluid states. In all cases, the efficiency requirements and operating speeds are typically higher than grid frequencies (for < 100 MWe scales). Turbines are typically radial designs below 10 MWe and axial designs above 50 MWe, with overlapping designs in the 10-50 MWe range. The range of sizes and speeds for sCO_2 centrifugal compressors and radial turbines up to the 20 MWe scale is highlighted in example calculations presented by (Musgrove *et. al.*, 2013) in Figure 6.

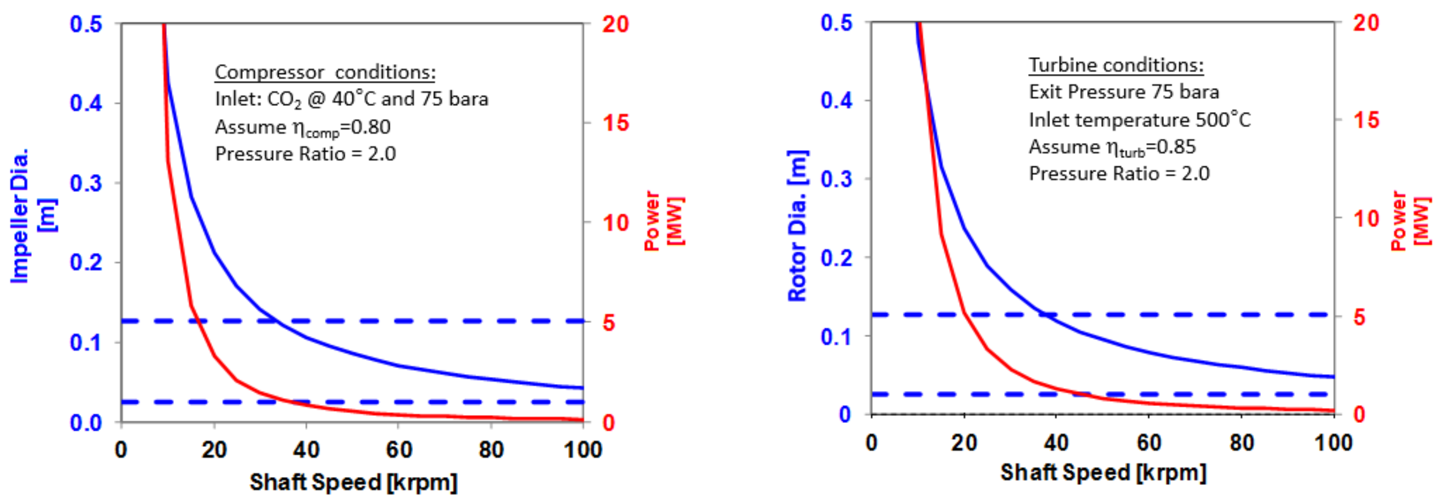

Figure 6. Example Sizes and Speeds for Radial sCO_2 Turbomachinery (Musgrove *et al., 2013*)

Several shaft and sealing configurations exist for transmitting shaft power between the turbomachinery and any generators/motors in an sCO_2 cycle, and also between turbine(s) and compressor(s). In theory, the simplest and perhaps most elegant solution is to package all elements within the same high-pressure casing and operate the generators/motors at high pressure. This solution is compact and eliminates the need for shaft end seals and their associated leakage. In fact, this layout was applied to initial sCO_2 research prototypes operated at Bettis Atomic Power Laboratory and Sandia National Laboratories. However, the high fluid density in these units has resulted in prohibitively high windage losses in the motor cavity, as noted by Wright *et al.* (2010) and Kimball and Clementoni (2012). These machines seek to reduce the motor cavity pressure via an internal seal and scavenge pump/compressor, but the packaging of these seals inside the pressure vessel is a challenge and power consumption by the scavenge compressor significantly reduces cycle efficiency. A hermetically-sealed machine concept also requires bearings that may be placed in the process environment (e.g. gas bearings or magnetic bearings) rather than oil-film bearings that are used on nearly all large industrial machinery. Finally, since a gearbox cannot be incorporated within the high-pressure casing, a nonsynchronous alternator speed may be required at many scales and costly power electronics are required to match grid frequency. Due to these limitations, most sCO_2 machinery configurations at high power levels isolate the bearings and, potentially, gearboxes from the turbomachinery via shaft end seals.

The choice to include a gearbox depends on the scale, efficient turbomachinery speeds, and other layout options including the sealing configuration or single- or dual-shaft configurations. Epicyclic (planetary) gearboxes are typically preferred for sCO_2 machines due to their smaller size and lower power consumption. Epicyclic gearboxes are available at shaft power ratings up to approximately 60 MW (GE Oil & Gas, 2016) and are often considered for coupling the turbine to a synchronous generator at 1800 or 3600 rpm. A gearbox may also be used to separate turbine and compressor speeds. Gearbox losses reduce the overall system efficiency (Beckman and Patel (2000) report gearbox losses near 1.5%), but this penalty may be eliminated by improved compressor efficiency at higher speeds.

Dual-shaft machinery layouts incorporate separate expanders for driving the compressor and generator, similar to split-shaft gas turbines. This layout enables higher-speed operation that may (depending on cycle conditions and scale) allow for more efficient compressor and possibly high-speed turbine performance. Dual-shaft layouts are also attractive for versatility during startup and shutdown since the turbine-compressor unit can be operated independently of the generator set. Disadvantages of dual-shaft turbomachinery include the requirement of a separate costly pressure casing for each of the two machines. Increasing the number of machines increases the cost of piping and ancillary equipment such as valves, lubrication oil systems, and sealing systems. In addition, the amount of CO_2 lost from the system (or cost of a leakage recompression system) increases proportionally to the number of shaft end seals, which may be doubled for a dual-shaft configuration.

COMMON DESIGN ATTRIBUTES AND COMPONENTS

This section describes a number of turbomachinery components that are common to both sCO_2 compressors and turbines. Because general details regarding these components are available from a variety of standard industry literature, this tutorial focuses on sCO_2-specific challenges and solutions.

Bearings

Bearings can be categorized in 4 basic groups: rolling element, sliding element, magnetic, and fluid-film. A high-level comparison of these bearing types is given in Table 2.

Table 2. Bearing Type Summary [Adapted from Brun *et al.* (2017)]

	Rolling Element	Sliding Element	Fluid-Film	Magnetic
Working medium	Gas/oil	Working fluid	Gas/oil	Working fluid
Shaft support	Rolling contact/ hydrodynamic lift	Sliding contact	Hydrodynamic/ Hydrostatic lift	Electromagnetic fields
Stiffness	High	Low	High	Medium
Damping	Low	Low	High	High
Load capacity	Medium	Low	High	Medium
Control	Passive	Passive	Passive	Active
Contacting	At low speed &	Always	At low speed	Never

	Excursions			
Cost	Low	Low	Medium	High
Drag torque	Low	Medium	Low-medium	Very low

Bearing selection is rather complex and depends on many factors such as cost, duty cycle, load, speed, size/weight, efficiency, and dynamic performance. When selecting a bearing for sCO_2 applications a number of factors must be considered. The bearing must have good load carrying capability, good stability characteristics, and allow for high surface speed. A summary of the different bearing types proposed for use in sCO_2 applications and their relevant power ranges is given in Figure 7. Note that although this figure proposes a suitable range for a given bearing type, these are only approximate and bearing manufacturers may provide solutions that exceed the indicated range.

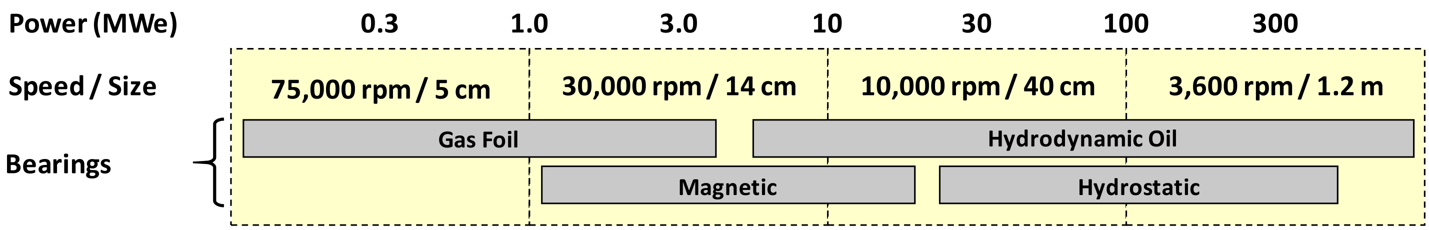

Figure 7. Bearing Type as a Function of Rotor Power [adapted from (Seinicki *et al.*, 2011)]

Although Figure 7 references a number of bearing types, in industrial scale land-based sCO_2 applications, the fluid-film oil bearing is expected to be the most prevalent. Fluid-film bearings bring many advantages over other bearings types, where the most profound advantage is durability due to non-contacting operation. Additionally, for oil-lubricated bearings, the damping provided by the fluid-film relative to its rolling element bearing and magnetic bearing counterparts improves machine vibration characteristics and stability.

Supercritical CO_2 (sCO_2) turbomachinery poses unique challenges to bearing support systems; these challenges stem from the working fluid properties of sCO_2. When compared to other conventional working fluids, sCO_2 possesses a high fluid density while retaining low viscosity. As a working fluid in a power cycle, this combination of properties result in a compact, power-dense, turbomachine. Therefore, bearing applications in sCO_2 turbomachinery face unique challenges: high bearing surface speed and high unit load.

High Surface Speeds

Increasing torque at a given shaft speed requires larger shaft diameters for safe torque transmission. Since bearing surface speed is proportional to the product of shaft radius and speed, increased torque for a given rotating speed results in higher rotor surface velocities. For oil fluid-film bearings, most manufacturers attempt to keep bearing surface velocities below 300 ft/s (91 m/s) (Nicholas, 2013). Despite some experimental work with higher surface velocities, most manufacturers tend to stay within these limits; otherwise, heat generation in the fluid film becomes problematic. For smaller scale turbomachinery, surface speed limitations in oil-type fluid-film bearings can be overcome by the use of gas bearings; these bearings are often limited to a load carrying capacity of 15 psi, which can be problematic given the high unit loading expected in these applications as noted in numerous thrust bearing failures by Kimball and Clementoni (2012).

High Unit Loading

Unit loading in a shaft results from numerous sources. In most applications, radial bearing unit loads result primarily from the weight of the shaft due to gravity. For sCO_2 turbomachinery, these loads are not expected to differ from other large land-based power cycle applications. In sCO_2 turbomachinery, high unit loads are predicted to result from asymmetric pressure differences across volutes and scrolls at off-design operation, lateral forces transmitted across gear teeth in gearboxes and integrally geared machinery, and transient loads due to upset conditions. For most machinery, asymmetric pressure gradients at off-design operation are not a great concern; however, the wide range in compressor operation required in sCO_2 applications having a variable compressor inlet temperature near the critical point could result in significant off-design operation of the volute.

Although radial bearing unit loads in sCO_2 applications cause concern, predicting and managing thrust loads is also important. As turbomachine gas density and pressure increase, the distribution of pressures around each turbomachinery stage has a more severe impact on thrust. Even if thrust at design is well predicted, these pressure distributions will change substantially as the machine operates off design, causing thrust. Accommodating additional thrust on a shaft requires a larger thrust bearing, which stresses the thrust bearing even further., increasing the probability of having hot spots in the bearing and increased power loss.

Although few relevant works exist regarding foil bearings in sCO_2 conditions, Conboy *et al.* (2012) and Kimball and Clementoni (2012) discuss some of the challenges faced in using high speed sCO_2 lubricated foil bearings in their bench-top scale test facilities. Notable among their observations were a number of thrust bearing failures. While radial gas foil bearings are not explicitly mentioned in the papers above, the authors have experience with implementing radial gas foil bearings in a 3200 psi downhole compressor-expander. For this application, achieving stable full-speed operation has been a challenge due to the high critical speed ratios inherent in gas foil bearing-supported rotors (Wilkes *et al.*, 2016b).

Rotordynamics

The combination of high power density, dense working fluid, and high temperatures present in sCO_2 turbines makes rotordynamic design challenging. Although each of these challenges has been successfully solved individually in other applications, the unique combination of these poses a challenge in sCO_2 applications. This section focuses on lateral rotordynamics challenges common to most sCO_2 turbomachines, including lateral instability, cross-coupling, and shaft elements that increase axial length.

Supercritical CO_2 compressors and turbines have similar rotordynamic challenges as reinjection gas compressors. In this application, pressures often exceed 5000-10000 psi, resulting in gas densities approaching water. With this high density, a number of strategies have been developed to improve stability. The most common approach to improve stability is to incorporate damper bearings or damper seals (i.e. hole-pattern seals) to improve the damping ratio of the shaft's lowest mode. In sCO_2 turbomachinery this approach may not be enough. The source of instability for sCO_2 turbomachinery is cross-coupled stiffness induced by swirling high-density gas around the wheels, seals, and shaft cavities. These sources can be mitigated by reducing swirl velocity using swirl brakes.

Cross-Coupling in Annular Seals and Secondary Flow Passages

Annular seals reduce leakage between regions of high and low pressure within a turbomachine. This is often accomplished with toothed labyrinth seals. Since turbomachines impart or extract energy into the flow through swirl (usually in the direction of rotation), this swirling flow leaks through the secondary cavities and enters the annular seals. This degrades stability. Additionally, the secondary cavities themselves can generate cross-coupling. Reducing swirl in these cavities and entering the seals is critical to improving stability.

To perform a stability analysis on sCO_2 machinery, SwRI recommends the work performed by Moore *et al.* (2007). With this newest method, CFD is used to calculate a dimensionless coefficient, which predicts an aerodynamic cross-coupling term for all similar impellers at varying operating conditions. This coefficient is applied in the equation below to calculate the cross-coupled stiffness of a single impeller using the following equation:

$$K_{XY} = \frac{C_{mr}\rho_d U^2 L_{shr}}{Q/Q_{design}}$$

In the above equation, C_{mr} is a CFD-determined coefficient (dimensionless), U is the wheel tip speed, L_{shr} is the axial length of the shroud from eye seal to tip (in), and Q/Q_{design} is the ratio of volume flow relative to design volume flow. This approach was validated with several test cases on unstable compressors and has a physics based derivation. Note that the cross-coupling is proportional to gas density.

The natural gas reinjection compressor shown in Figure 8 features oil-lubricated, tilting-pad radial and thrust bearings and dry gas seals. This 9:1 compressor has a 262 bar (3800 psi) case pressure which about double the requirement predicted for most sCO_2 recompression cycles. For sCO_2 applications, the main compressor and recompressor could be put on the same shaft in a back-to-back arrangement. Since both suction and discharge pressures are nearly identical, resulting in a low pressure differential across the division wall seal, the seal would provide little benefit in terms of stiffness and damping. The impeller diameter on the compressor will be smaller than the recompressor due to its higher inlet density, which will create a thrust imbalance. In this case, a balance piston would be required to minimize thrust, and could be equipped with a damper seal.

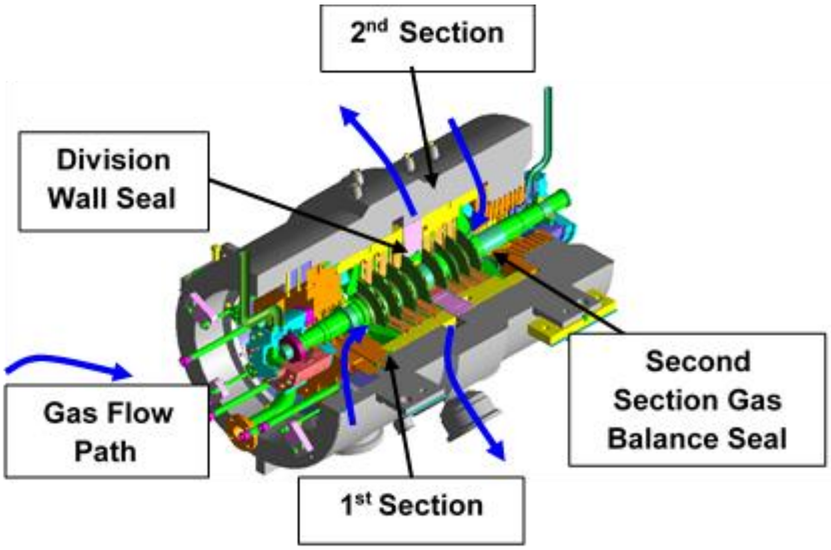

Figure 8. Isometric View of High-Pressure Compressor

Hole pattern seals have been used successfully by many researchers such as Moore (2002, 2008) to stabilize high-pressure re-injection compressors. Hole pattern seals provide stabilizing damping that can result in increasing stability with increasing discharge pressure if properly applied. Damper seals contain an array of cylindrical or hexagonal holes as shown in Figure 9, and typically operate against a smooth rotor.

Figure 9: Example Hole Pattern Damper Seal (Moore et al.¸ 2002)

Figure 10 shows measured and predicted effective stiffness and damping for a divergent hole pattern seal taken from the testing done at Texas A&M University (Smalley *et al.*, 2004). Note that there is a regime at low frequency where these seals are highly destabilizing. In general, these seals become destabilizing when running around 4 time the machines first critical speed. With that in mind, as shaft sizes decrease and lengths increase as is the case in sCO_2 applications, there will be a limit as to the effectiveness of a hole pattern seal to stabilize a machine. Also, it should be noted that negative effective stiffness at low frequencies can lower the first natural frequency, causing further concern (see (Camatti *et al.*, 2003), (Moore *et al.*, 2006), and (Eldridge and Soulas, 2005)).

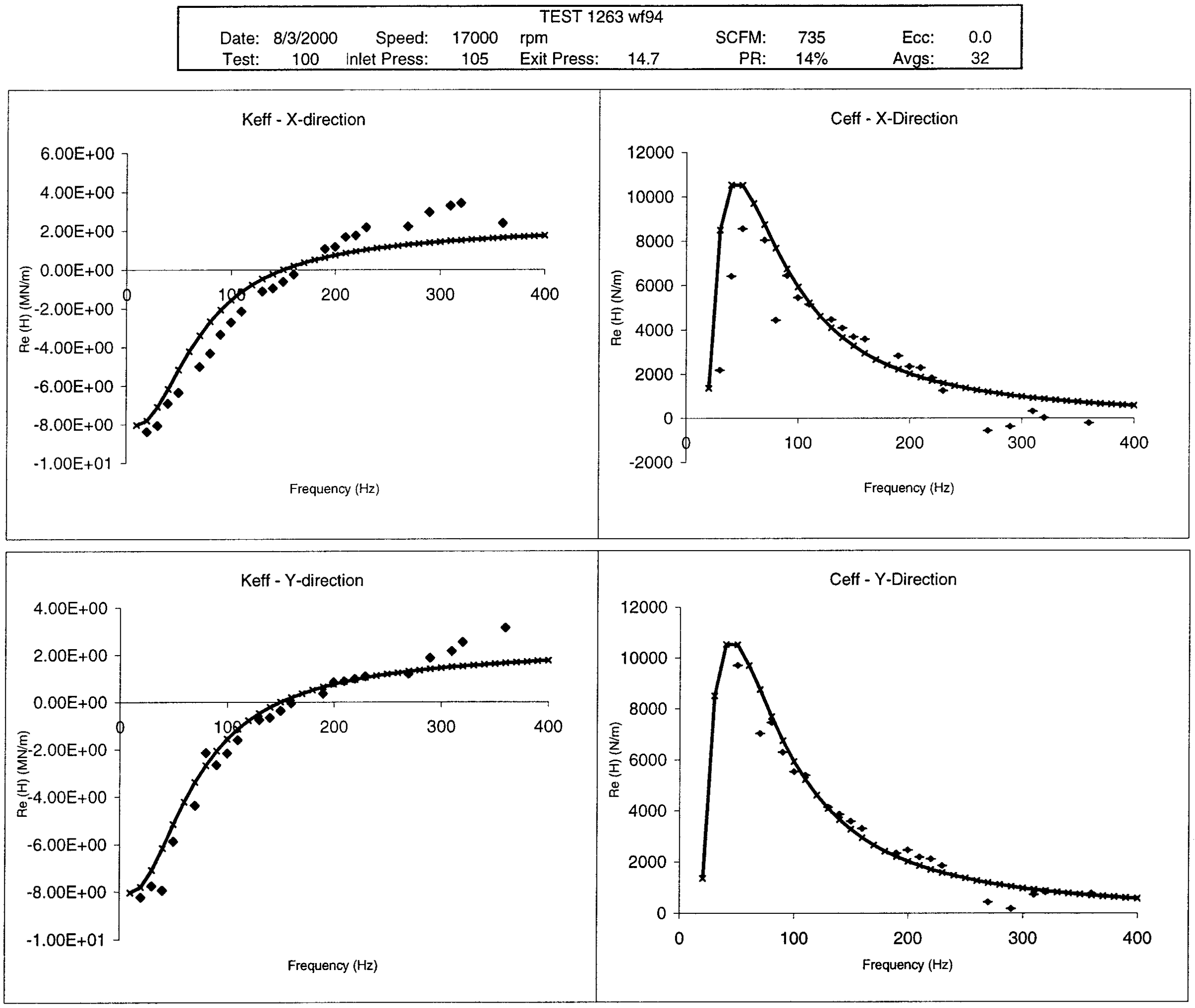

Figure 10. Measured and Predicted Effective Stiffness and Damping for Hole Pattern Damper Seal (Smalley *et al.*, 2004)

Brown and Childs (2012) showed that hole pattern-seals damping and stiffness vary greatly with inlet pre-swirl. Therefore, swirl brakes are used upstream of critical seals inside the compressor. Figure 11 shows one example of a swirl brake. This anti-swirl device can actually create negative swirl in the seal due to counter-rotating vortices at the inlet of the seal lands. A properly designed swirl brake transforms a normally destabilizing seal into a stabilizing one with little detriment to the leakage and the performance of the turbomachine.

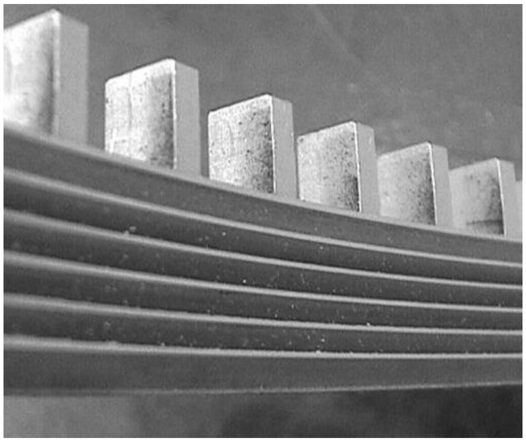

Figure 11. Swirl Brake Installation at Impeller Eye Location (Moore *et al.*, 2002)

Shaft Axial Length

sCO_2 rotor-bearing systems are prone to have high flexibility ratios because of the considerable length requirements, this is especially true in sCO_2 turbines. High temperatures in turbine expanders require thermal management systems to protect end seals and minimize shaft stresses. This thermal management system requires additional axial length compared to conventional high-pressure machinery. The reason for increased length in the thermal management section is due to thermal stresses caused by temperature gradients. During a recent project, Wilkes et al. (2016) determined that the length of a thermal transition region needed to be at least 1.75 times the shaft diameter to stay below recommended stresses for the case of a 500°C temperature drop in Inconel 740. For the case of a 4 inch centrally supported rotor, this may require as much as 14 additional inches of shaft length, which lowers the fundamental natural frequency of the machine.

Rotordynamics Case Study: 20 MWe sCO_2 *Expander*

In their book, Brun *et al.* (2017) provide an example rotordynamic study for a 20 MW turbine having three stages. Although a comprehensive review will not be provided here, it is sufficient to state that when plotted on a Fulton chart, the experience of the major compressor manufacturers does not encompass sCO_2 turbines as shown in Figure 12.

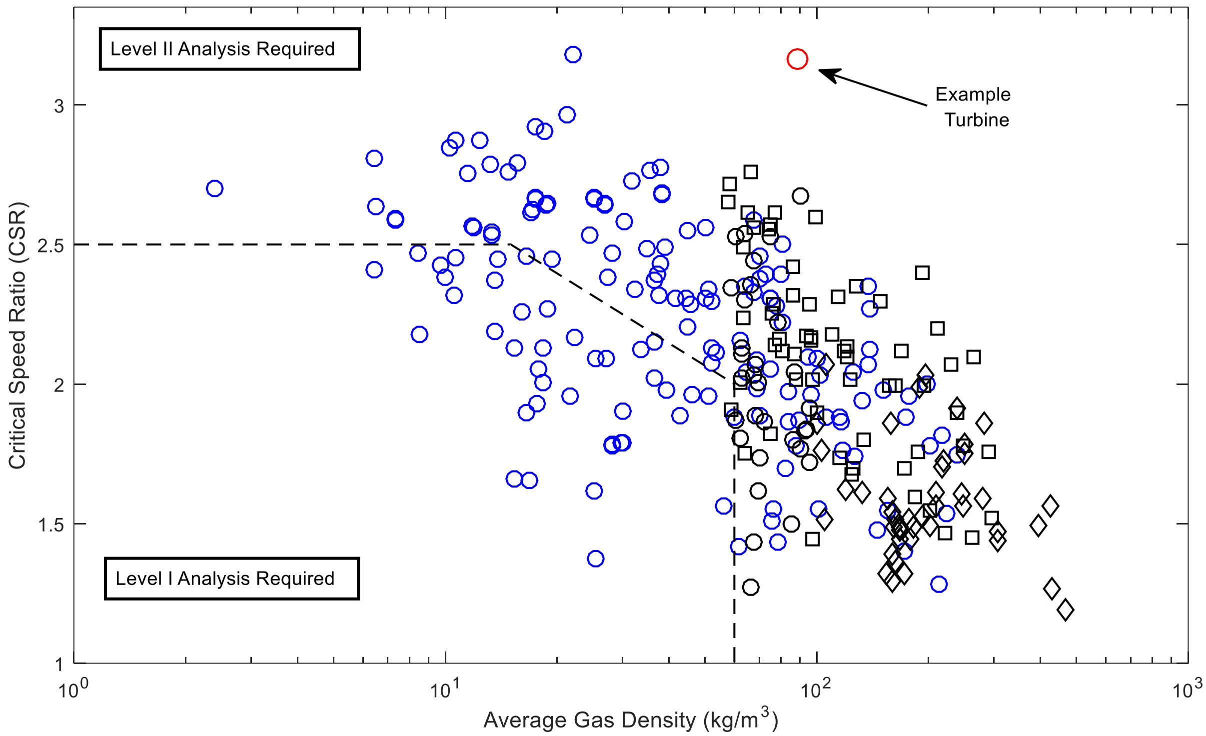

Figure 12: Fulton Experience Chart for Gas Reinjection Compressors [Adapted from (Brun *et al.*, 2017)]

Shaft End Seals

Existing dry gas seal (DGS) technology is designed to reduce shaft seal leakage to the atmosphere to an acceptable level. Though a variety of DGS configurations exist (single seals, tandem seals, and tandem seals with an intermediate labyrinth seal). Tandem seals with an intermediate labyrinth are generally used in natural gas applications due to the minimal leakage to atmosphere and their perceived robustness. However, for non-flammable applications like sCO_2, a single seal is adequate.

For compressors (and turbines prior to warm up), CO_2 can create dry ice across the seal face and potentially clog the seal vents. To mitigate this risk, the seal gas supply is usually heated to 80-100 °C to avoid multi-phase dome and dry-ice regimes. On the other side of the spectrum, dry gas seals contain temperature-limited elastomeric or polymer seals between the seal cartridge and the casing and between the shaft sleeve and shaft. With this constraint, most dry gas seals have a maximum operating temperature of 350 °F (177 °C) (inlet gas temperature). This poses a number of risks for the DGS in the event of a shutdown or failure for hot sealing applications (200°C -700°C). Since seal gas serves to cool the seal, it must be supplied even after a hot shutdown until the casing has cooled below this maximum temperature. In the event of a seal failure, hot gas will overwhelm the buffer supply. This requires that internal components should at least be made of a stainless alloy to avoid heat damage until the loop can be blown down. Inherent here is that excessive backpressure at the seal vent will cause hot gas to enter the bearing cavity. This would pose a fire risk, and must be mitigated by the inclusion of an adequate vent line.
In sCO_2 systems, the DGS supply is typically slightly above the compressor inlet pressure. Since most cycles operate near the critical point for the compressor inlet, real gas properties must be considered. One publication addressing this is Thatte *et al.* (2016), who describe a multi-scale coupled physics approach for fluid-structure-thermal interactions in an sCO_2 DGS.

Existing DGS are also applied to shaft sizes that are typically 4 to 6 inches in diameter (Bidkar *et al.*, 2016c) and potentially up to 13.75 inches (John Crane, 2015). Utility-scale turbomachinery is predicted to require, a commercially available dry gas seals that are not currently available. Bidkar *et al.* (2016c) describe the analysis and design of a hydrodynamic face seal that is approximately 24 inches in diameter for a 450 MWe sCO_2 turbine (Bidkar *et al.*, 2016a). This paper notes specific design challenges with high heat generation, thermal-induced coning, and the need for computationally intensive coupled fluid-structure-thermal analysis supported by experimental testing.

Pressure Containment / Materials

One of the primary challenges with casing design for sCO_2 turbines is containing a combination of high temperature and pressure. While inlet pressure and temperature are similar to ultra-supercritical steam turbines, the exhaust pressure and temperature are far greater. Additionally, the need for dry gas shaft end seals requires a sharp temperature gradient in the pressure containment near the ends of the casing, which is unique to an sCO_2 turbine. In general, sCO_2 compressor operating pressures and temperatures are well within experience limits for CO_2 from the oil and gas industry.

Due to the complex nature of typical turbomachinery case geometries, finite element analysis (FEA) is commonly employed to design for pressure containment. ASME Boiler and Pressure Vessel Code (BPVC), Section VIII, Division 2, Part 5 is recommended for these analyses, where components may be evaluated for plastic collapse, local failure, buckling, and cyclic loading. Thermal loads (heat transfer boundary conditions, etc.) in addition to all relevant mechanical loads (internal pressures, contact forces, constraints, etc.) may be necessary to determine the temperature distribution and resulting thermal stresses within the part (i.e., not just considering the maximum temperature applied uniformly throughout).

Static Seals

Care must be made in the selection of the static seals in sCO_2 turbomachines. In many machines, various elastomer seal materials such as Viton®, Aflas®, or Kalrez® provide good options up to 400/450/600 °F. However, particularly in sCO_2 service, all of these materials are at risk of explosive decompression when gas that has slowly absorbed into the material during pressurized operation expands within the material during a decompression event. This results in material failure. Most manufacturers provide allowable depressurization rates for each material; however, the likelihood of explosive decompression worsens with increases in pressure or temperature and decreases with elastomer hardness. Due to the rapid pressure swings predicted due to transient or upset conditions, it is likely that depressurization rates for elastomers will be exceeded in industrial sCO_2 applications. Clementoni and Cox (2014) note problems with explosive decompression in Viton® seals in their sCO_2 loop. Thus, elastomer seals are generally not preferred for sCO_2 service.

Polymer seals are not subject to this issue and are the seal of choice for sCO_2 static seals. Polymer seals (most are PTFE alloys) are more challenging to implement, as they have lower elasticity. C-shaped seals with a metallic spring and pressure energized shape help overcome some of these shortcomings, but still require good surface finish for leak-tight operation.

In the hot sections of sCO_2 turbomachines, metallic seals are often required. These too are usually pressure energized C-seals but have little elasticity so they must be applied to an axial face for leak tight operation. A good surface finish (16 uin or better) is required in this instance, and most of these seals are silver or gold plated nickel alloys. Figure 13 shows an example of these seals.

Figure 13. Metallic C-Seals for High Temperature

COMPRESSOR DESIGN CONSIDERATIONS

Compressors in sCO_2 applications operate at similar pressures as existing CO_2 compressors in sequestration and reinjection applications. However, the compressor inlet temperatures near the critical pressure are intentionally kept high enough to avoid the range and operability challenges caused by high sensitivity of gas properties to variations in temperature near the critical point. However, most sCO_2 power cycles are configured with the compressor operating at lower inlet temperatures very near the top or top-left of the vapor dome in order to take advantage of compression power savings and resulting higher cycle efficiencies with the cooler fluid. Figure 14 compares existing sCO_2 compressor operating regions with proposed sCO_2 operating regions on a pressure-enthalpy diagram. Operation near the critical point results in several potential design challenges including the impeller mechanical design, consideration of real gas properties, and potential for condensation or cavitation at the compressor inlet.

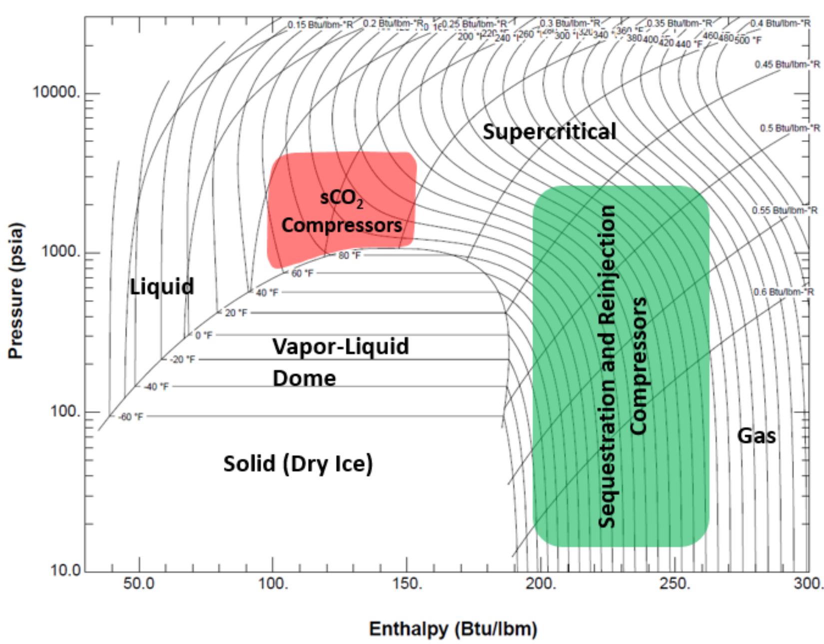

Figure 14. Compressor Operating Regions for Existing Applications and sCO_2 Cycles [Adapted from (Allison *et al.*, 2016)]

Impeller Mechanical Design

Most sCO_2 compressor impeller designs incorporate closed or covered compressor impellers due to relatively low tip speed requirements, high blade loading, and insensitivity of eye seal leakage to thermal growth. Covered impellers are generally less prone to high-cycle fatigue failures than open impellers due to a stiff blade support configuration, but multiple failure cases have been reported in the literature by Kushner *et al.* (2000) and White *et al.* (2011). The high fluid density in sCO_2 impellers will also affect the natural frequencies of blade-dominant modes (Gill *et al.*, 1999) and generates relatively high aerodynamic loading amplitudes. Thus, impeller designs should consider the dynamic stresses resulting from wake excitation from upstream and downstream stator components such as inlet guide vanes, diffuser vanes, and struts that apply periodic excitation to the blades. This approach is not unique to sCO_2 applications and various analysis/design methods have been described by Kushner *et al.* (2000) and Lerche *et al.* (2012).

Real Gas Properties & Range

Optimal sCO_2 cycle designs typically result in a compressor operating relatively close to the critical point, where fluid properties change rapidly and ideal gas assumptions are extremely inaccurate. Thus, all phases of the design should be performed with an accurate equation of state. The NIST REFPROP software produces accurate properties for pure CO_2 near the critical point and can be coupled directly to many compressor design codes via direct call of REFPROP functions or through property lookup tables. Either approach increases solution times relative to ideal gas law solutions, and care must be taken for simulation of operation very near the two-phase region in order to avoid convergence issues due to simulated phase change.

The rapidly changing properties of sCO_2 near the compressor inlet result in significant changes in fluid density for small variations in inlet temperature. This variation can result in large fluctuations in volume flow, particularly for air-cooled cycles. Pelton *et al.* (2017) presented the analysis results in Figure 15 that show a 45% change in volume flow for an sCO_2 compressor with nominal inlet conditions of 37 °C and 85.2 bara. With additional range margin needed for off-design cycle operation plus surge and stall margin for operability, range requirements for an sCO_2 compressor can reach or exceed 70%. Pelton *et al.* (2017) also present a novel partially-shrouded stage configuration coupled with a casing treatment for shroud bleed and other features resulting in a wide range, high-efficiency sCO_2 compressor stage.

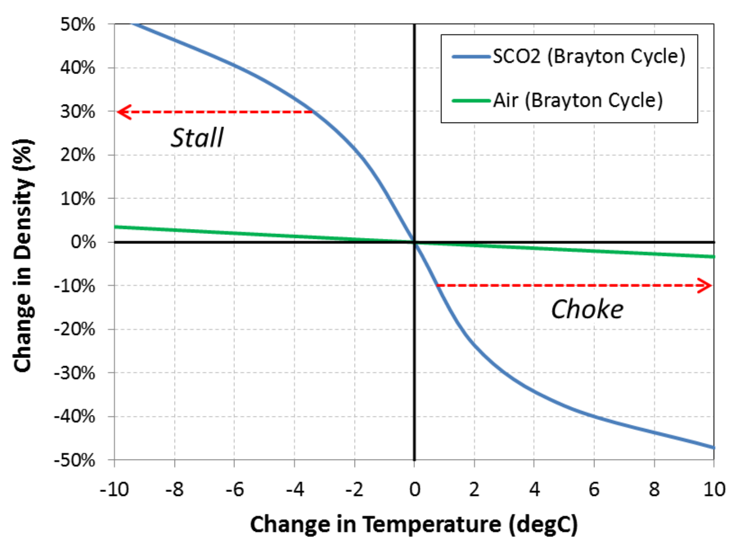

Figure 15. Compressor Inlet Density Variation for sCO_2 and Air Brayton Cycles (Pelton *et al.*, 2017)

Phase Change

Several publications have addressed the potential for and risks associated with operation in the two-phase region, which will occur first at the compressor inlet due to local flow acceleration and the accompanying reduction in static pressure and temperature. Monge *et al.* (2014a, 2014b) proposed a non-dimensional criterion named Acceleration Margin to Condensation (AMC) to quantify the margin between the expected fluid properties in the inducer and the saturation line, illustrated in Figure 16. The AMC is defined as the throat Mach number at which the static properties of the fluid lie on the saturation line.

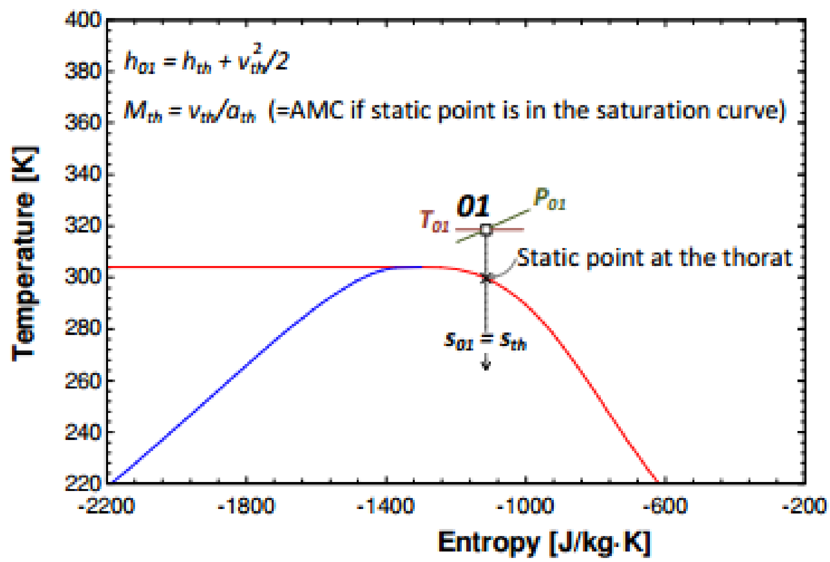

Figure 16. Definition of Acceleration Margin to Condensation (Monge *et al.*, 2014a)

Further publications by Lettieri *et al.* (2014) and other analyze the nucleation time required for condensate to form in a sCO_2 fluid and show that it is typically larger than the residence time of the fluid in the compressor inlet at saturation conditions. In the cited work, an alternate condensation limit was defined as the ratio of the residence time of the fluid in the condensing region to the nucleation time. As long as this ratio is less than one, the residence time is less than the nucleation time and condensation would not be expected. Their test results did show increasing uncertainty in nucleation time as the temperature approach the critical temperature, so caution is still recommended for operation near the critical point.

Noall and Pasch (2014) published data describing stable operating experience of a motor-driven kW-scale sCO_2 compressor in the two-phase region, indicating that stable operation below the saturation line may be possible. Their test data showed steady operation at a variety of points across the entire saturation region with no apparent harmful effects. These results suggest that even if two-phase operation occurs, the densities for liquid and vapor phases at high pressure are similar enough to avoid harmful operation.

TURBINE DESIGN CONSIDERATIONS

High-Temperature Materials

High-Temperature components in sCO_2 cycles operate at temperatures similar to ultra-supercritical steam turbines, and much of the work on alloys for steam applications is also applicable to sCO_2 cycles. For high temperature sCO_2 turbines operating at high temperatures approaching or exceeding 700 °C, designs are creep limited and nickel-based alloys are required to achieve sufficient creep strength. Creep strength and other material properties are significantly decreased in this temperature range and are strong functions of temperature as indicated in Figure 17 for several materials that are frequently considered for sCO_2 applications.

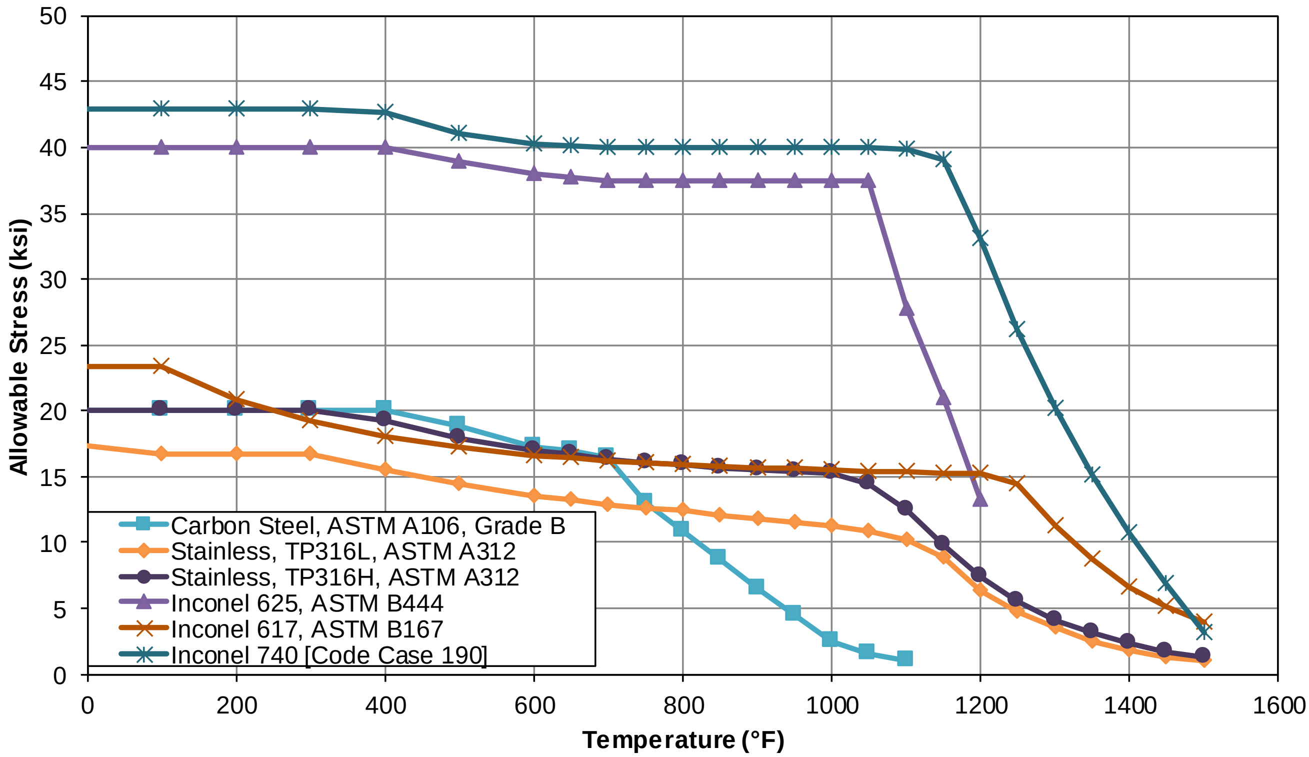

Figure 17. ASME Piping Code Allowable Stress vs. Temperature [Adapted from (Brun *et al.*, 2017)]

In addition to creep properties, another critical consideration for turbine materials is corrosion performance in an sCO_2 environment. Corrosion test data above 650 C is relatively scarce, although multiple test programs described by Brun *et al.* (2017) are currently expanding the available data set. Even as autoclave test data are becoming available at higher temperatures, there is a continued need for materials testing to determine sensitivity to CO_2 purity, corrosion performance for various CO_2 mixtures, and testing in a high flow velocity environment to confirm real-world corrosion behavior.

Overspeed Risk

Similar to gas and steam turbines, a turbine trip valve and protection system is required for sCO_2 turbines to prevent over-speed. Due to compact nature and low inertia of the turbine rotor, sCO_2 turbines are especially vulnerable to sudden loss of electrical load. Fast acting, close coupled turbine trip valves for high-temperature sCO_2 turbines are not available and are still a technology gap at this time but are required for the reliable operation of the turbine. API 616 guidelines for gas turbines require an over-speed of 120% to give the over-speed protection time to shut off the supply flow. Transient simulation of this event is typically used to demonstrate adequate performance of the over-speed system.

Thermal Management

Due to the need for dry gas seals (DGS) with sCO_2 turbines, a thermal management system is required to maintain adequate temperatures for the dry gas seals and results in a temperature gradient required to reduce the temperature of the turbine from either inlet or exhaust temperature down to a temperature that the dry gas seals can tolerate. This region requires careful design to minimize the thermal stresses in the components. Figure 18 shows a typical temperature field in the rotor. A similar gradient will occur in the casing. Axial temperature gradients generate thermal stresses, but radial temperature gradients can cause high thermal stresses and should be minimized. The seal gas flow provides the heat sink required to maintain acceptable dry gas seal temperatures. The thermal seal is located between the DGS and the process and is designed generate an even temperature distribution with no radial temperature gradients. There is usually minimal pressure drop across this seal. Advanced analysis tools such as conjugate heat transfer computational fluid dynamics models are employed to design this critical region of the turbine. Transient thermal analysis using finite element analysis is performed to evaluate cold start-up and hot shut-down scenarios quantifying low cycle fatigue life in these parts.

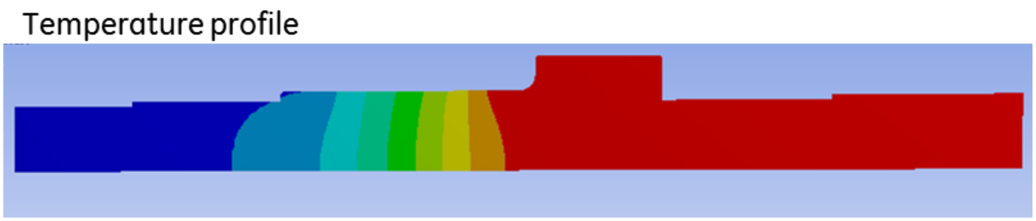

Figure 18. Temperature Profile in an sCO_2 Expander Shaft with High Gradient in Transition Region (Kalra *et al.*, 2014)

Turbine Rotor/Blade Mechanical Design

The sCO_2 turbine may be of the radial in-flow or axial design. Their blade/impeller attachment are similar to other turbomachinery applications with several key differences. First, sCO_2 turbine has higher power density than most other types (the exception being rocket engines turbopumps), so the pressure (static) loading on turbine blades cannot be ignored as with low-density applications. Aggravating this issue is the relatively small size of the wheels and shaft requiring special considerations for shaft attachment. For example, mechanical interference of a wheel on the shaft and relying on friction to keep the wheel from slipping will usually be insufficient to drive the torque to or from the wheel. Features such as splines or keys for pressed on wheels or axial splines (Curvic or Hirth style) for stacked up designs will likely be required.

For axial blades, dove-tail or fir trees may not be capable of handling the blade bending moments on the joint. Furthermore, due to the relatively small size of the wheels for a given power level, there may not be physical space between the blades for these features. On a positive note, the tip speeds of the wheels for sCO_2 turbines tend to be low to moderate compared to other applications. Therefore, integral shrouds are typically used and improve blade dynamics, damping, and aerodynamic performance.

CONCLUSIONS

This tutorial paper has described various sCO_2 turbomachinery design details that address the combined high-pressure, high-temperature, and high-density operating environment with high property gradients near the critical point. These conditions result in high power density turbomachinery designs that can be designed for high operating efficiencies with reasonable machine sizing and staging. The conditions also introduce multiple design challenges for a variety of components, including high bearing surface speeds and loads, dense gas effects on rotordynamics and blade loading, low-leakage shaft end sealing, high-temperature pressure containment and compact thermal management in the turbine, and wide operating range requirements and potential for condensation in the compressors. These challenges require significant engineering to overcome before sCO_2 turbomachinery can begin to displace steam turbines or gas turbines, which have been developed and refined for over 100 years.

Despite these challenges, a number of sCO_2 turbomachinery designs and prototypes have been successfully developed in the past decade, and high-temperature MW-scale units will be tested in the near future. Simulation and prototype test results indicate that the aforementioned design challenges can be successfully overcome by employing existing high-performance turbomachinery component technologies, including advanced bearings and damper seals, dry gas seals, high-temperature high-strength materials, and new manufacturing processes for compact turbomachinery. Significant engineering research and development effort is still required to validate existing prototype concepts and scale machinery up for utility-scale applications. This process can be accelerated through advanced simulation tools including conjugate heat transfer, transient fluid-thermal-structural simulations, advanced rotordynamic stability predictions, and others. These technologies and tools, in addition to data from prototype testing, are expected to enable development and eventual commercialization of sCO_2 turbomachines for a variety of applications in the coming years.

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