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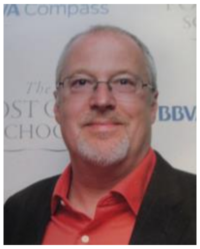
MARINA BAY SANDS

# TUTORIAL: Overview of Turbomachinery for Super-Critical $CO_2$ Applications



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## ABSTRACT

Cycles involving super critical carbon dioxide ($sCO_2$) have the potential to increase system efficiencies well beyond current industry norms. Research on advanced direct and indirect cycles is ongoing in national labs and major companies. $sCO_2$ machinery tends to have small foot print sizes making for excellent applications in marine, or space limited, environments. As $scCO_2$ turbomachinery gains acceptance in various industries the need to understand the applications, potential, and limits is paramount. Discussed in this tutorial are 1) various direct and indirect cycles 2) various applications, and 3) specific impact to turbomachinery design. Specific applications are described in detail including waste heat recovery, power generation, concentrated solar power, and marine applications. Discussed are transient, thermal-mechanical, material, rotordynamic and many other factors affecting the turbomachinery.

## INTRODUCTION

Substantial research and development is being done on cycles and turbomachinery specifically designed for super-critical carbon dioxide. This research is funded by private and publically funded entities with the sole intention of commercializing the technologies developed. Some products are already on the market, others being commissioned in 2017/2018, and more to follow in the coming decade and beyond. The applications range from waste heat recovery, concentrated solar power, to advanced gas turbines for power generation. This paper gives an overview of various aspects of $sCO_2$ cycles and machinery. As a result of reading this overview the reader should have an understanding of why so much focus exists on sCO2, what markets and applications are able to leverage the technology, and what obstacles must be overcome for successful turbomachinery design. The challenges are not only in the turbomachinery, but also in other areas such as controls, combustion, and limits of heat exchangers.

### *Literature Review*

Super-critical $CO_2$ was first proposed as a working fluid in a re-compression Brayton cycle by Combs [1977.] Combs envisioned the working fluid and cycle for shipboard applications where space and efficiency are a premium. In early 2000's the cycle began to regain interest amongst researchers. Ultimately, Wright at Sandia National Laboratories [2010] began more in-depth work to prove out the cycle. Sandia National Lab maintains a $sCO_2$ test-loop to investigate the key technology issues associated with this cycle. In the testing to date the turbo-expander has reached maximum speeds of 45,000 rpm at 315°C with peak flow rates above 4.1 kg/s, and relatively low pressure ratios of just over 1.65. The data from these tests indicate that the basic design and performance predictions for the recompression cycle are achievable as Comb's first predicted. However working with such a dense gas in high-speed turbomachinery showed that the reliability and performance scalability of the turbomachinery still required technology development. More recently work has been done to match the recompression Brayton cycle to the needs of the concentrated solar power plants. The size $sCO_2$ Brayton cycle is being designed to match current modular solar fields and has been identified as being commercially highly competitive, as reported by Ma [2011]. Moore [2007] states that researchers in the oil and gas industry have developed technologies for compressing and pumping $CO_2$ at super-critical pressures for other applications, and hence, compression technology required for the $sCO_2$ Brayton cycle is considered a moderate risk. In contrast, industrial scale turbines for operation on $sCO_2$ do not have a precedent in the industry beyond some small demonstration radial turbomachinery units currently being run in labs.

*Indirect Systems*

Miller et. al. [2017] describes super critical $CO_2$ coal fired Rankine cycles and in particular the effects of cycle architecture, combustion-air preheater temperature, and cooling source type were considered subject to comparable heat source and reference conditions taken from the steam Rankine reference cases. Combinations and variants of sCO2 power cycles — including cascade and recompression and variants with multiple reheat and compression steps — were considered with varying heat-rejection subsystems — air-cooled, direct cooling tower, and indirect-loop cooling tower. Where appropriate, combustion air preheater inlet temperature was also varied.

Bennett, et. al. describes the cycle optimization for an integrally geared compressor expander based on a Brayton Cycle. Where cycle modeling was conducted for both on design and off design. On-design modeling was simulated with all components operating at their design point. This was used to compare the performance of different cycle configurations and design temperatures. Off-design modeling was then performed to investigate the temperature dependence of the cycle efficiency and power output and to develop a control strategy. Strategies considered and discussed include: turbine bypass, compressor recycle, inlet guide vanes, and inventory control. To determine the best operating conditions for each configuration and control strategy, a genetic algorithm was implemented to optimize the cycle performance across the range of operating temperatures being considered. The final selection of cycle configuration, design temperature and control strategy is also presented.

Beyond the turbomachinery the recuperators play an important ole in the cycle optimization. Dyrby, et. al [2014] describe that the recuperators in the Brayton cycle are modeled assuming a constant heat exchanger conductance value, which allows for computationally efficient optimization of the cycle's design parameters while accounting for the rapidly varying fluid properties of carbon dioxide near its critical point. Representing the recuperators using conductance, rather than effectiveness, allows for a more appropriate comparison among design-point conditions because a larger conductance typically corresponds more directly to a physically larger and higher capital cost heat exchanger. The model is used to explore the relationship between recuperator size and heat rejection temperature of the cycle, specifically in regard to maximizing thermal efficiency.

*Indirect System Validation*

MW-scale design and validation testing of indirect $sCO_2$ components and cycles is currently being conducted and developed in high-temperature 1 MWe-scale and 10 MWe-scale test loops at Southwest Research Institute. The 1 MWe-scale test loop is described in detail in [Moore 2015] but relevant details are summarized here for reference. The loop accommodates the full test pressures (80-280 bar) and temperatures (45-715 °C) of a high-efficiency recompression cycle for a Concentrating Solar Power (CSP) application. Since the primary purpose of the test loop is to characterize the mechanical performance of the expander under development, and not to demonstrate a particular system performance, a simple recuperated cycle was chosen with a primary recuperator, an external heater to provide high temperature, and a separate pump to provide high-pressure CO2 flow to drive the expander. The simple cycle loop is less expensive and has less risk to implement. Fabrication of the test loop is complete as shown in a photo of major loop hardware including the heater, expander, and recuperator (Figure ), and commissioning activities are currently in progress. The test loop design-point operating conditions are shown in Table 2.

A 10 MW-scale high-temperature test loop is also being designed for construction and high-efficiency cycle validation under the DOE Supercritical Transformational Electric Power (STEP) program. Although still in the design phase, a preliminary layout of the test facility is shown in Figure 1. The pressure and temperature ranges of the facility will be similar to those shown in Table 2, but the flow will be approximately 10 times higher.

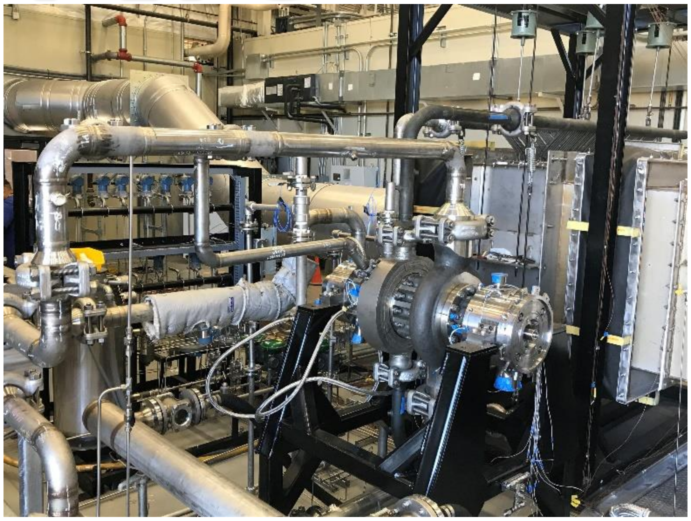

**Figure 1. 1 MWe-Scale SunShot $sCO_2$ Test Loop Photo**

**Table 2. 1-MWe-Scale Test Loop Operating Conditions**

| Component | Exit Temperature, °C[°F] | Exit Pressure, bar [psi] | Flow Rate, kg/s [lbm/s] |
|---|---|---|---|
| Pump | 29.22 [84.6] | 255 [3698] | 9.91 [21.85] |
| Piping (2) | | 254 [3688] | |
| Recuperator HP | 470.0 [878] | 252 [3659] | 8.41 [18.54] |
| Piping (6-8) | - | 252 [3654] | |
| Heater | 715.0 [1319] | 251 [3639] | |
| Piping (9) | - | 251 [3634] | |
| Expander | 685.7 [1266] | 86 [1247] | |
| Piping (10) | - | - | 9.910[21.85] |
| Recuperator LP | 567.3 [1053] | 84 [1218] | |
| Piping (5) | - | - | |
| Cooler | 79.58 [175] | 83 [1204] | |

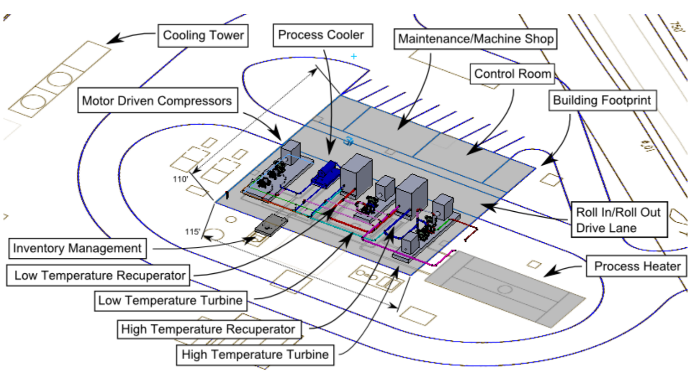


**Figure 1. 10 MWe-Scale STEP sCO2 Test Facility Conceptual Layout**

*Direct Systems*

Direct fired systems dependent on the heat generation coming directly from oxy-fuel combustion of LNG. The Allam Cycle [Allam, 2014] is a high-pressure, oxy-fuel, supercritical $CO_2$ cycle that generates low-cost electricity from fossil fuels while producing near-zero air emissions; all $CO_2$ generated by the system is produced as a high-pressure, pipeline-ready by-product for use in enhanced oil recovery, industrial processes, or sequestration. The Allam cycle is not the only option to take advantage of oxy-fuel combustion. Mcclung, et. al [2015] prepared an excellent paper that describes the various aspects of oxy-combustion cycles. Advanced oxy-combustion coupled with supercritical carbon dioxide (sCO2) power cycles offers a path to achieve efficient power generation with integrated carbon capture for base load power generation. One commonality among high efficiency sCO2 cycles is the extensive use of recuperation within the cycle. This high degree of recuperation results in high temperatures at the thermal input device and a smaller temperature rise to the turbine inlet. When combined with typical high side pressures ranging from 150 to 300 bar, these conditions pose a non-trivial challenge for fossil fired sCO2 cycles with respect to cycle layout and thermal integration.

## GAS PROPERTIES

Compressors in $sCO_2$ applications operate at similar pressures as existing $sCO_2$ 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 3 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.

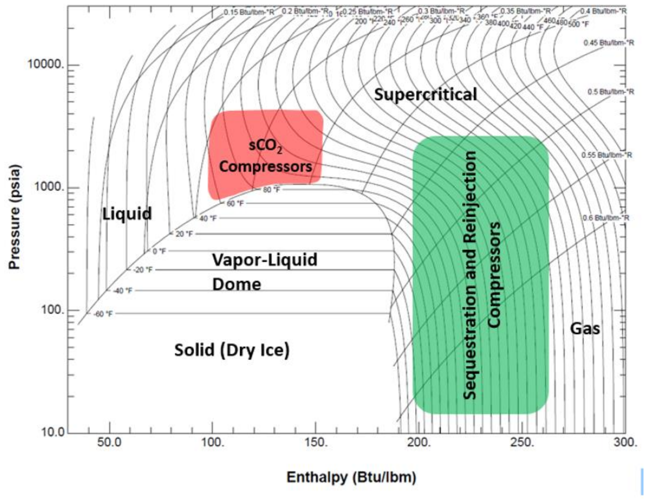


**Figure 3. Compressor operating regions for existing applications and sCO2 cycles [adapted from (Allison et al., 2016)]**

Many researchers have investigated the potential operating risks associated with designing a cycle with the main compressor inlet near the critical point of the working fluid. The primary concern is associated with the rapid change in fluid density that can occur near the critical point with modest variation in temperature or pressure. A significant change in density will result in a change in the cycle mass flow as the compressor adjusts to maintain a stable volume flow. Additionally, if the variations in temperature or pressure are significant that two-phase flow many develop. The region which will develop multi-phase flow initially is 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 fluid properties the saturation line, illustrated in Figure 4. The AMC is defined as the throat Mach number at which the static properties of the fluid lie on the saturation line. Based on this criteria, and a detailed analysis of the flow in the compressor, the design condition can be adjusted based on expected variations in suction conditions to avoid condensation.

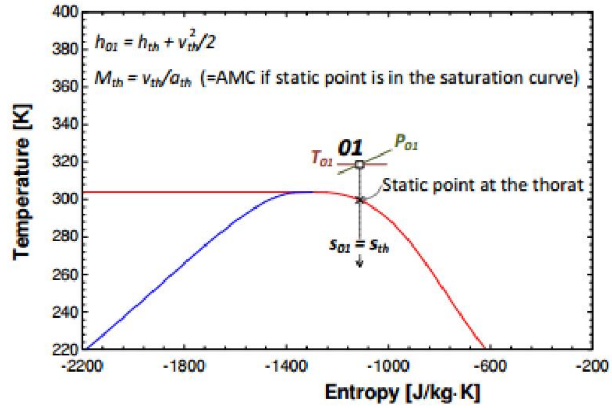


**Figure 4. Definition of acceleration margin to condensation (Monge et al., 2014a.)**

While the AMC is a simple design guideline, it may be overly conservative for more margin in the cycle design than is actually required. Work by Lettieri et al. (2015) and others have analyzed the nucleation time required for $sCO_2$ to condensate and find that it is typically larger than the residence time of the fluid in a typical compressors. Lettieri proposed an alternate condensation limit defined as the ratio of the residence time of the fluid in the condensing region to the nucleation time. When this ratio is less than one, the residence time condensation should not occur.

There is very little practical compressor operating data available to use as a benchmark for future designs. Data on one centrifugal compressor operating near the critical point was published by Noall and Pasch (2014). The reported test results show stable operating of the $sCO_2$ compressor both above and slightly below the saturation line. These results suggest that operation slightly below the dome may be possible without adverse effects of the stability of the compressor.

## COMMON CYCLES FOR $sCO_2$

*Cycle Performance*

The temperature entering the turbine defines what level of efficiency can be achieved for a cycle. Figure 5 shows the maximum theoretical limit of cycle efficiency achievable, the Carnot cycle. Also plotted are several common cycles considered for $sCO_2$ applications. For low temperatures (below 400 C) extracting power is difficult. And little benefit is seen between the recuperated and recompression cycles. The obvious conclusion here is that little benefit is to be gained in the additional cost of including recompression. As temperatures increase above 500 C the benefits of the recompression cycle become very noticeable and a trade can be mad if the additional CAPEX for recompression is worth the efficiency gain. As the temperature increase above 700 C then limitation of the recuperator and turbine blading most be considered. Cycles that are relevant above 700 C become the Transitional cycle, Allam Cycle, and Brun cycle, plus others that are more complicated.

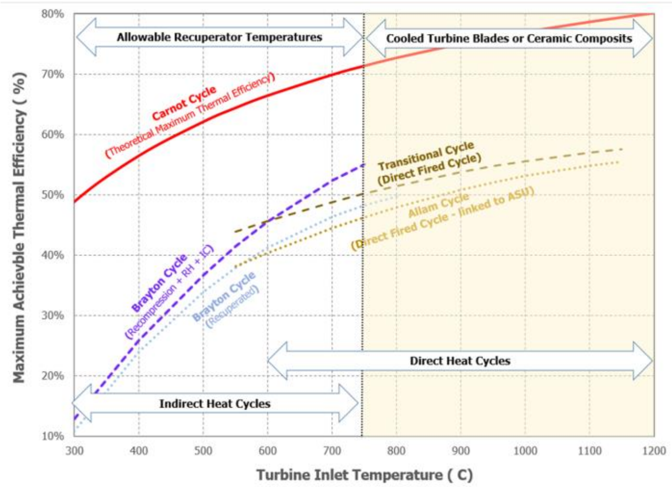


**Figure 5. Several $sCO_2$ related cycles shown as a range of turbine inlet temperature and achievable efficiency.**

*Cycle Benefits and Challenges*

There a number of different cycles and variations on each one of those. The best approach to determining the correct cycle for a potential application extends beyond the best efficiency and must consider factors such as operating costs, installation costs, fuel type (for direct fired systems,) and the grade of heat (indirect systems.) To make the best determination a Levelized Cost of Energy (LCOE) helps define the most cost effective approach for each application. The LCOE assessment is based on the total operating cost for the power plant over its lifetime divided by the total power delivered over the plants life time. The basis advantages and challenges of several common cycles is listed in Table 1. For example a simple Brayton cycle has the fewest components, but has relatively low performance.

**Table 1. Comparison of most common $sCO_2$ cycles.**

| Cycle | Benefits | Challenges |
|---|---|---|
| Brayton (Simple) | • Fewest components<br>• simple package | • Low performance |
| Brayton (Recuperated) | • Improved efficiency from | • Additional cost and technical challenges associated with recuperator |
| Brayton (Recompression) | • Improved efficiency at most conditions | • Additional turbomachinery to incorporate |
| Allam | • Reduced recuperator inlet temperature compared to supercritical cycles<br>• Water can be removed at lower pressure<br>• High power density | • Not very efficient at moderate temperatures.<br>• Complex cycle with additional components and integration with ASU is necessary |
| Brun | • Reduced recuperator inlet temperature compared to supercritical cycles | |
| Trans-critical | • Minimal components<br>• Reduced compression work | • Must deal with phase change |

*Common Applications*

Geothermal, waste heat recovery, nuclear, and concentrated solar power, and power generation are all relevant for the $sCO_2$ cycles and machinery. The typical heat sources on Figure 6 give an indication of what level of efficiency can be achieved for each application. $sCO_2$ machinery is compact and has a small foot print. But the cost can be deceptively high as the need to seal and contain high pressures is present.

Geothermal has the lowest temperature range and therefore has the lowest achievable efficiency. Waste heat recovery covers a wide range of potential heat sources (200 ºC to 500 ºC.) For geothermal and low temperature WHR the cost-to-benefit is typically not present, in this author's opinion. However, as heat sources with temperatures greater than 450 ºC can be identified and applied a favorable cost-to-benefit can be achieved. For even higher heat source temperatures such as nuclear and concentrated solar power the achievable efficiencies coupled with the inherent benefits of the $sCO_2$ gas make for attractive applications.

With temperature above 700 ºC, such as direct fired oxy-fuel systems, substantial efficiencies are achievable with a relatively small foot print, near elimination of Nox, and the ability to reduce or sequester $CO_2$ makes for exciting power generation opportunities.

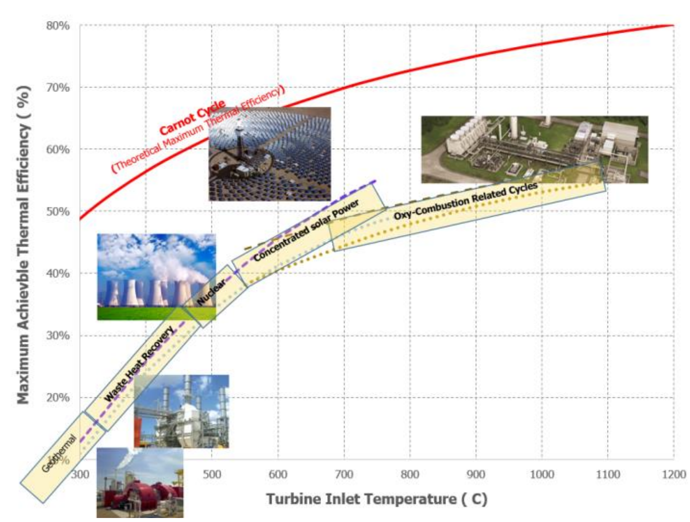


**Figure 6. Various applications as related to different levels of heat availability.**

Figure 7 shows the relationship between sCO2 mass flow rate, Expander TIT, Power Output, and the maximum achievable cycle efficiency for an integrally geared sCO2 Brayton cycle machine. In order to achieve significant efficiencies temperatures above 500 C are necessary. For expander 300 C it is difficult to achieve efficiency's above 15%, but the small machinery foot print and even this level of efficiency may be relevant for some applications.

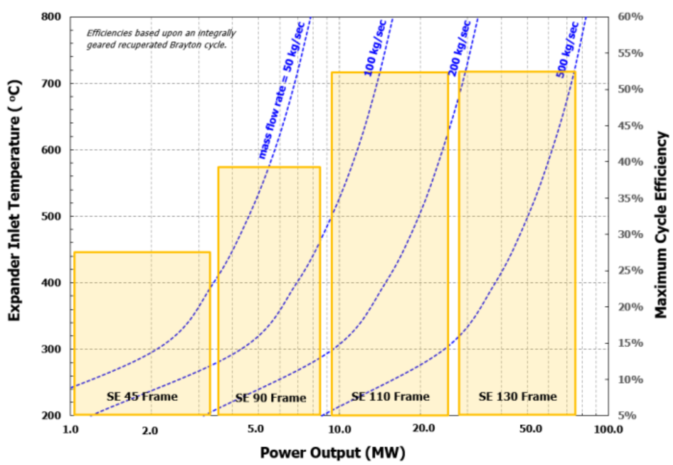


**Figure 7. Interaction of temperature, power, mass flow rate, and efficiency.**

An example of the application of recompression Brayton cycle is for concentrated solar power applications. The US DOE has been supporting the development of advanced sCO2 power blocks for concentrated solar power systems. Figure 8 shows the schematic of the US DOE's SunShot program, where concentrated solar energy is reflected from the field to a receiver where that in turn heats a heat transfer fluid (such as a slat column) that then transmits the thermal energy to the sCO2 loop.

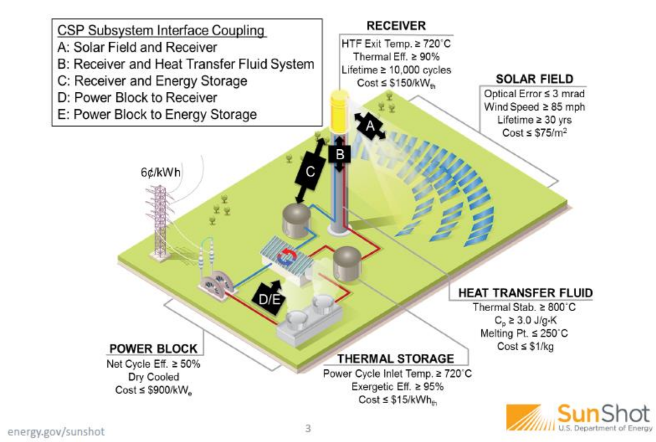


**Figure 8. Concentrated power systems are well suited to sCO2 cycles.**

Such systems have the ability to generate electricity. It is also possible to use combustion in oxy-fuel gas turbine systems to generate electricity.

*Direct Power Generation*

An oxy-fuel gas turbine as part of a super-critical CO2 cycle offers the advantages of high efficiency and reduced NOx emissions. The Oxy-Fuel gas turbine is also a very small foot print as compared to a traditional gas turbine for the same power rating. An Oxy-fuel gas turbine must deal with high-pressures, small machinery sizes, and withstand the operating environment. Figure 9 lists the various aspects of an Oxy-Fuel Gas Turbine.

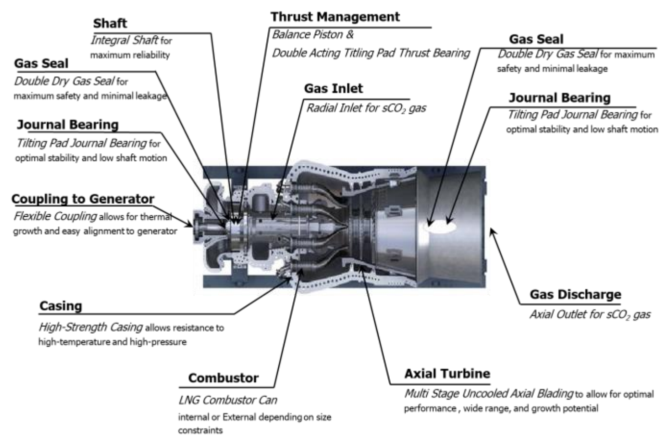


**Figure 9. Advanced gas turbines applying oxy-fuel combustion and sCO2 cycles allow for high efficiency, small foot print, and reduced emissions.**

## TURBOMACHINERY

Design of optimal turbomachinery systems is by definition a cycle of multi-discipline optimizations. The aerodynamic performance is tightly coupled to the structural integrity of the impeller or wheel. There is also a coupling between the aerodynamics (aerodynamic cross-coupling) and the rotordynamics that is also coupled to the structural design of the impeller or wheel. Figure 10 shows the primary factors that are coupled in the process of designing a compressor and/or turbine.

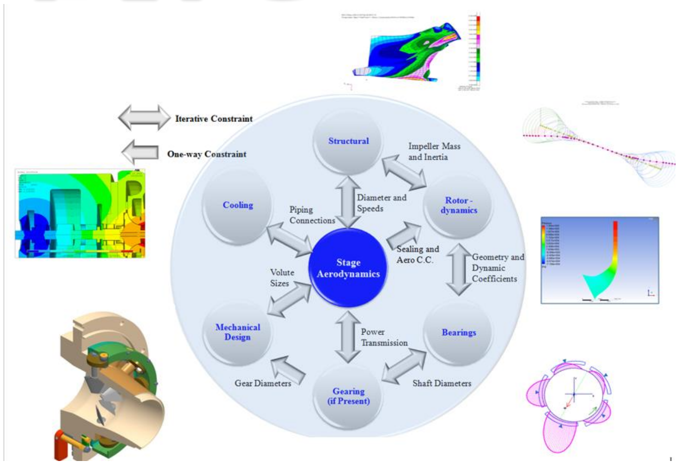


**Figure 10. Turbomachinery design and analysis considerations.**

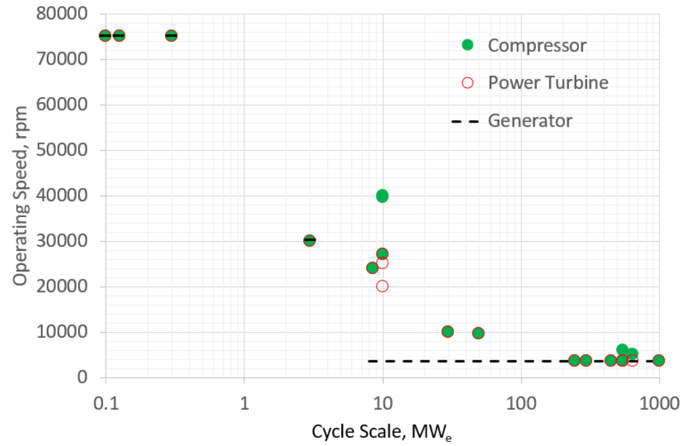


**Figure 11. Summary of Published sCO2 Turbomachinery Speeds and Scales [Adapted from (Brun et al., 2017)]**

## COMPRESSORS, PUMPS, and EXPANDERS

*Inline and Integrally Geared*

Due to the high fluid density, $sCO_2$ turbomachinery designs will typically be smaller and often faster that steam/gas turbine counterparts of equivalent power outputs. Design details 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 in [Brun *et. al.* (2017)], and a summary of published $sCO_2$ turbomachinery speeds and scales from is provided in Figure 11.

Cases shown at 3 MWe or lower typically utilize high-speed turbines, compressors, and generators that operate at high speeds above grid frequency. Higher-power cases between 7-50 MWe all use gearboxes between the power turbine and (synchronous) generator, and cases above 50 MWe have a synchronous power turbine. In most cases, a single turbine drives both compressors and the generator (green and red circles overlap), but other designs incorporate a split-shaft design with a separate higher-speed turbine to drive the compressor.

Several shaft and sealing configurations exist for transmitting shaft power between the turbomachinery and any generators/motors in a $sCO_2$ cycle, and also between turbine(s) and compressor(s). Most MW-scale $sCO_2$ machinery configurations 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. Gearbox losses reduce the overall system efficiency (Beckman and Patel (2000) report gearbox losses near 1.5%), but this penalty may be offset by improved turbine and compressor efficiency at higher speeds.

Dual-shaft layouts may be 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, sealing systems, and control complexity. 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.

Inline turbomachinery have excellent reliability. This is a distinct advantage when developing new products for $sCO_2$ machinery. Part of that advantage also lies in the minimal number of seals that are required. With the high pressures required for $sCO_2$, these are easily achievable with inline machinery. The key challenge for a $sCO_2$ turbomachine is the high gas density makes for small diameter impellers at high rotational speeds. Managing the rotordynamics is perhaps the greatest challenge here. Figure 12 shows an arrangement of an inline $sCO_2$ compressor.

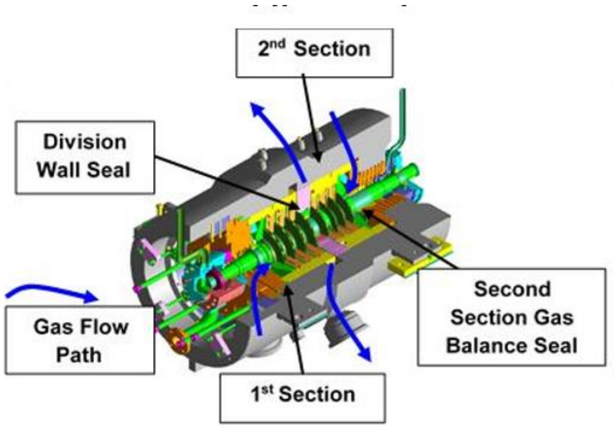


**Figure 12. Example of an inline compressors configurations.**

An integrally geared approach is based upon a bull gear with surrounding pinions. The pinions will have stages over-hung. The advantages of an integrally Geared approach is that each stage may be closely matched to its ideal speed and that intercooling can be applied between compression stages for optimal efficiency. Or that reheat may be applied between stages without significant interruption of the flow. Figure 13 shows a version that integrates the expander and compression stages. The challenge for the integrally geared approach is that seals must be applied for each stage.

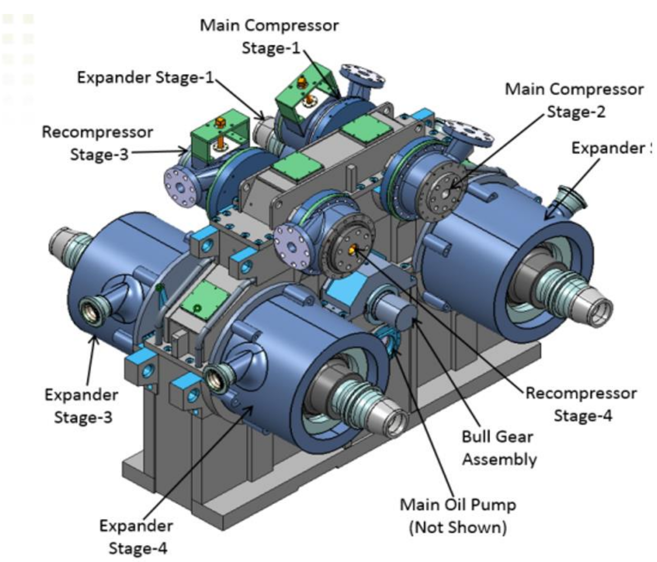


**Figure 13. Example of an integrally geared configuration.**

A 10 MWe-scale axial expander design has been developed by SwRI and GE Global Research Center under the SunShot program for a $sCO_2$ recompression cycle with design inlet/exit pressure and inlet temperature of 251/86 bar and 715°C. The turbine will be tested at a 1-MWe equivalent mass flow of 8.41 kg/s due to existing test loop limitations, so the existing test prototype employs reduced-area nozzle and blade flow passages to maintain design velocities. The turbine, shown in Figure 12 [Kalra 2014], has a design speed of 27,000 rpm and will couple (in the end application) to a generator via a gearbox on one end and to a combined main compressor and recompressor on the other. The turbine design is a four-stage axial expander with shrouded blades.

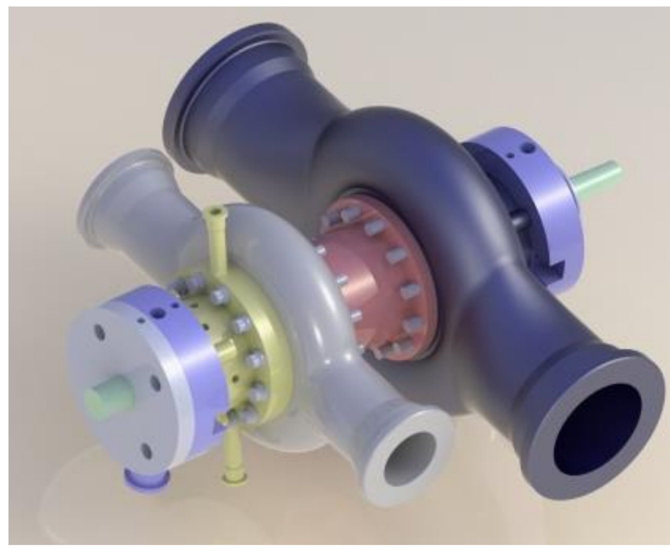

**Figure 12. Example of an Axial sCO2 Expander [Kalra 2014]**

In general, axial stages perform better at lower head and higher volume flow rates than radial machinery (i.e., higher $N_s$ and lower $D_s$). Thus, high-power (high-flow) cycles are more likely to use axial machinery than radial. Maintaining high efficiency over a wide off-design flow range tends to favor radial compressors over axial designs. Sienecki *et al.* (2011) studied turbomachinery types for a $sCO_2$ recompression cycle with scales ranging from 100 kWe to over 300 MWe and concluded that systems below 10 MWe will likely feature only radial turbines and compressors with a single stage or low stage counts. As size increases, the authors note that stage counts for all turbomachinery components are expected to increase. 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. 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 .

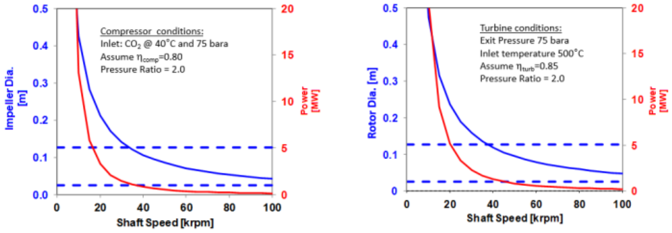


**Figure 15. Example Sizes and Speeds for Radial sCO2 Turbomachinery (Musgrove et al., 2013)**

*Aero-Structural Considerations*

The aer0-structural design optimization is always a tedious and essential component of designing a high performance turbomachinery. The structural considerations must avoid permanent deformation of critical clearances, fatigue, and overload fractures. Manufacturing constraints typically come into considerations for fillet dimensions, minimal castable or machinable thicknesses, and compatibility with the process fluid. The aero design is based initially on mean line design and then CFD to fine tune the performance and maximize range. Aerodynamics performance will typically seek to minimize blade thickness and reduce blockage to a minimum (i.e. minimize fillets at blade leading edges.) These optimizations are directly counter with structural integrity. The challenges for sCO2 applications are related with the need to add pressure loading in addition to centrifugal loads. Material compatibility must also be considered in the selection process. For the expanders thermal gradients must be mapped on to the geometry to determine thermal stresses. Therefore the high-pressure (and high-temperature) of sCO2 can limit the design process for aerodynamic efficiency.

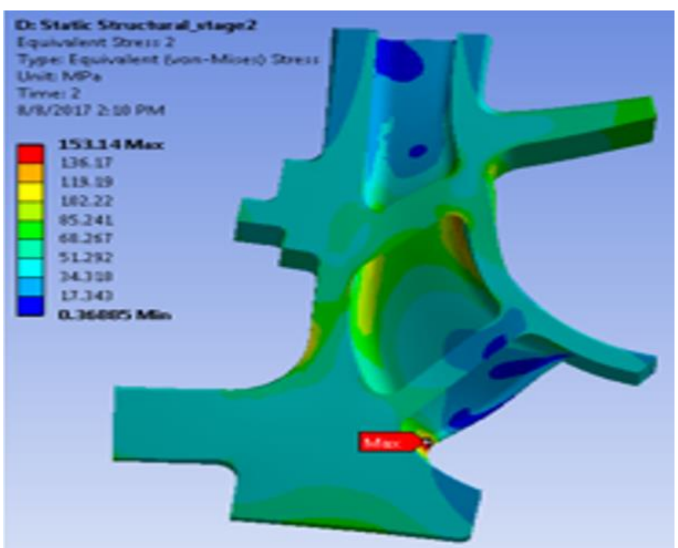


**Figure 16. Aero-dynamic and structural design must be optimized for the high power density machinery.**

## HOT RADIAL and AXIAL EXPANDERS

*Aero-Structural Design*

Selecting the appropriate expander architecture requires balancing the mechanical limitations of the rotordynamics, gears, bearings etc. with a configuration that will maximize efficiency. Initial scoping should be conducted during the earliest phases of design to select a layout and expander configuration that can meet the design objectives. A specific speed vs. specific diameter chart (Balje Plot) is often used as a preliminary screening tool, Figure 17. From this plot the appropriate, size, speed, and type of expander can be easily estimated. Due to the power density of $sCO_2$ machinery, it is critical to review the estimated flowpath size early as they can be small. Small flow paths will be both hard to manufacture and have an adverse effect on performance. Manufacturing approaches and tolerances should also be considered when selecting an expander architecture.

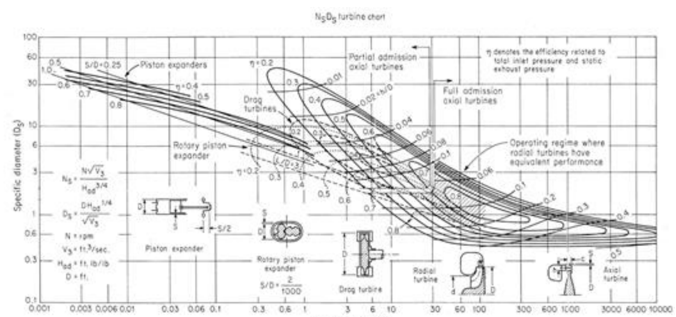


**Figure 17. The type of expander best chosen for an application depends on specific speed and specifics diameter (ref. Balje plot.)**

## MECHANICAL DESIGN

*Thermal-Mechanical Design*

The housing of sCO2 compressor sections must be designed to handle the high pressure loads. The design standard considered most relevant is the ASME Boiler Pressure Vessel Code. Fortunately the high density gas means that the sections are relatively small in diameter, but still managing the pressure containment and bolting is critical to success. An additional challenges is present when designing high-temperature expander sections. The high-temperature limits available material choices that meet ASME BPVC requirements and static sealing and thermal mismatch create potential for leakages that must be resolved. Figure 18 shows a thermal analysis conducted with a commercially available FEA code. The FEA allows the designer to optimize for stress minimization and understand changes to critical clearances. Allowing for hot sections to expand minimizes stresses. Also the section on the need to insulate hot sections to prevent gas loss of energy is critical.

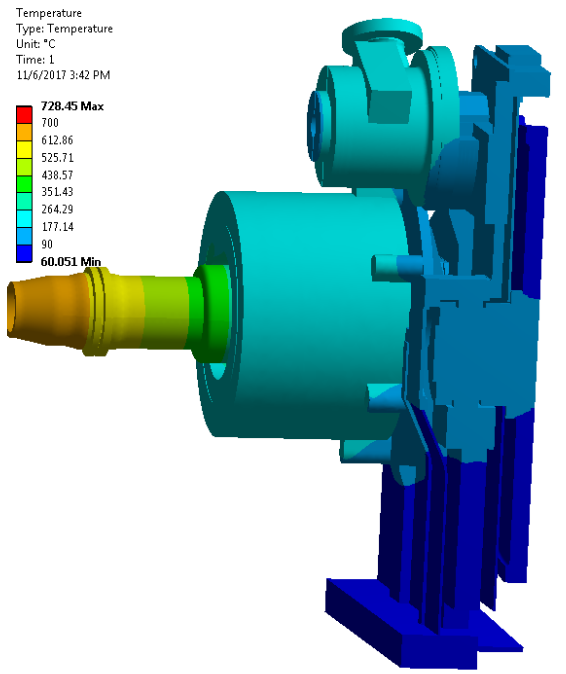


**Figure 18. Hot gas expansion means additional design emphasis based on thermal interactions and creep life.**

*Pressure Containment and Material Compatibilities*

Allison [2017] defines the limits to materials and pressure containment. One of the primary challenges with casing design for sCO2 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 a $sCO_2$ turbine. In general, sCO2 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).

*Rotordynamics and Bearings*

Table 2 lists the various types of bearings, in industrial scale and research related $sCO_2$ 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.

$sCO_2$ 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.

**Table 2. Bearing options for consideration.**

| **Bearing Type** | **Benefits** | **Challenges** |
|---|---|---|
| **Rolling Element** | Low parasitic losses.<br>Minimal oil flow requirements. | Life limited.<br>Requires secondary damping system for systems transitioning through critical speeds. |
| **Fluid Film (Oil)** | Excellent damping properties.<br>Lack of cross-coupling.<br>Long and reliable life | Sealing process gas from bearing oil. |
| **Fluid Film (Process Gas – Foil)** | Reduced sealing challenges.<br>Lack of lubrication system. | Limited load capacity<br>Lack of lubricity and risk for surface wear at start-up.<br>Varying properties depending on state and conditions of $CO_2$. |
| **Magnetic Bearing** | Provides damping to rotor systems.<br>Ability to provide open loop control to minimize shaft motions.<br>Low parasitic losses. | Advanced control systems required.<br>Limited load capacity. |

$sCO_2$ turbines and compressors experience high pressures, temperatures, and fluid densities that can result in rotordynamic challenges. Although each of these challenges has been successfully addressed in rotordynamic design for other application, the unique combination of these factors in $sCO_2$

applications plus the requirement for long operating life present a design challenge for lateral rotordynamics, including risks of including lateral instability, high cross-coupling forces, and (particularly in the turbine) 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 that of water. Within turbomachinery for both of these applications, swirl is imparted to this high-density fluid in cavities around the rotor (e.g. into annular seals and around compressor impellers or turbine rotors), resulting in sub-synchronous cross-coupling forces applied to the rotor that can excite rotor modes for machinery running above critical speeds. If excitation from cross coupling forces (which scale with density) exceed energy dissipation from damping in the rotor-bearing system, a rotordynamic instability will result.

With this high density, a number of strategies have been developed to improve stability of high-density turbomachinery. The most common approaches to improve stability are to (1) incorporate damper bearings or damper seals (e.g. hole-pattern seals) to improve the damping ratio of the shaft's lowest mode, and (2) utilizing swirl brakes or shunt injection to reduce swirl in these cavities and entering the seals is critical to improving stability. Accurate calculation of rotor critical speeds, damping, fluid swirl, and cross-coupling forces in seals and around aero stages is required during the design phase in order to prevent instabilities. This calculation methods should, at a minimum, comply with API 617 methods but higher accuracy methods may also be necessary (e.g., [Moore 2007]).

$sCO_2$ turbines in particular may require long shaft lengths to isolate end seals and bearings from turbine operating temperatures. 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. The shaft length reduces the frequency of rotor critical speeds so that they are more susceptible to swirl excitation. High rotor temperatures reduce the elastic modulus of the shaft material, which also reduces critical speeds (particularly rotor bending modes) and can make it difficult to meet separation margin requirements between running speed and critical speeds. $sCO_2$ compressor systems for the main compressor may require range extension features (e.g. variable IGVs), which may also increase shaft length for compressor systems.

*Seals*

$sCO_2$ applications require careful selection of both shaft end seals (preventing leaking to atmosphere) and internal annular seals within the turbomachine (minimizing internal leakage and providing damping for rotordynamic stability).

Integrally geared $sCO_2$ compressors can be configured to run subcritical and therefore may not be at risk for instabilities due to $sCO_2$ cross-coupling. For inline $sCO_2$ compressors for a recompression cycle, the main compressor and re-compressor may be put on the same shaft in a back-to-back arrangement. This arrangement minimizes net thrust loads but is likely to require operation above the machinery's lowest critical speeds. The suction and discharge pressures for each compressor are nearly identical, resulting in a low pressure differential across the division wall seal and this seal would not have a strong effect on internal losses or rotordynamics. Due to its higher inlet density, the main compressor tends to have fewer stages with smaller impellers than the re-compressor. This results in a net thrust imbalance, which is often remedied by a balance piston. The balance piston seal experiences a high pressure differential and could be equipped with an effective damper seal depending on balance piston location relevant to rotor mode shapes of concern.

Typical damper seals applied to stabilize high-pressure reinjection compressors include hole pattern seals and labyrinth swirls with swirl brakes [Moore (2002)]. Hole pattern seals contain an array of cylindrical or hexagonal holes as shown in Figure 13, and typically operate against a smooth rotor. These seals provide stabilizing damping that can result in increasing stability with increasing discharge pressure if properly applied.

**Figure 13: Example Hole Pattern Damper Seal (Moore et al., 2002)**

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  shows one example of a swirl brake. This passive anti-swirl device can actually create negative swirl in the seal due to counter-rotating vortices at the inlet of the seal lands, which can convert a normally destabilizing seal into a stabilizing one with little detriment to the leakage and the performance of the turbomachine.

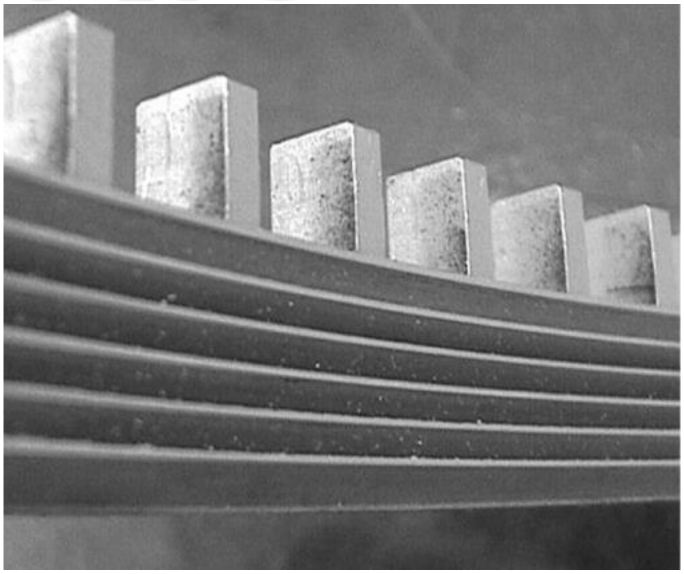

**Figure 20. Swirl Brake Installation at Impeller Eye Seal (Moore et al., 2002)**

For industrial-sized $sCO_2$ turbomachinery, axial face seals are the lowest-leakage option and are therefore the most often considered due to the sensitive of closed-loop cycle performance to atmospheric leakage. Dry gas seal (DGS) technology is currently the state of the art technology for minimizing shaft seal leakage to atmosphere. A variety of DGS configurations exist, including ingle seals, tandem seals, and tandem seals with an intermediate labyrinth seal. Tandem seals with an intermediate labyrinth are generally used in natural gas applications to minimize gas leakage to atmosphere. However, for non-flammable applications like $sCO_2$, a single seal is adequate and minimizes shaft length.

Existing DGS are also applied to shaft sizes that are typically 4 to 6 inches in diameter (Bidkar *et al.*, 2016a) and potentially up to 13.75 inches (John Crane, 2015). Utility-scale $sCO_2$ turbomachinery is predicted to require larger seals and commercially available options 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.*, 2016b). 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.

Dry gas seals operate with a very low clearance (approximately 3-10 μm) in order to minimize leakage to atmosphere. In order to ensure reliable operation of the seal, an appropriate supply system is required to condition the supply gas (most often taken from the compressor/pump discharge) for injection into the seal. During operation, most of the supply gas leaks bak into the process across an internal labyrinth seal. The supply gas system also typically involves conditioning and backup components to ensure a continuous flow of clean, dry gas to the seal during machinery operating and standby conditions. 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 flow is usually heated to 80-100 °C to avoid multi-phase dome and dry-ice regimes.

The supply and vent connections an example single dry gas seal system are shown in Figure . The DGSs are fed with a supply of clean $CO_2$ (green), most of which leaks inboard through a process labyrinth seal into the process in order to prevent liquid or particle contamination. An $sCO_2$ turbine would also likely utilize this buffering flow to protect the DGS from high process temperatures. The small remaining flow leaks radially inward through the running gap between the DGS rotor (light blue) and stator (purple) and then is directed outboard to mix with separation air. The separation air (blue) is clean, dry shop air that is provided to the labyrinth buffer seals in order to prevent oil migration from the bearings into the DGS. Approximately half of the separation air leaks to atmosphere via the bearing vent; the other half leaks inboard into the seal, mixes with the $CO_2$ leakage, and the resulting mixture (red) is directed out the seal vent to atmosphere.

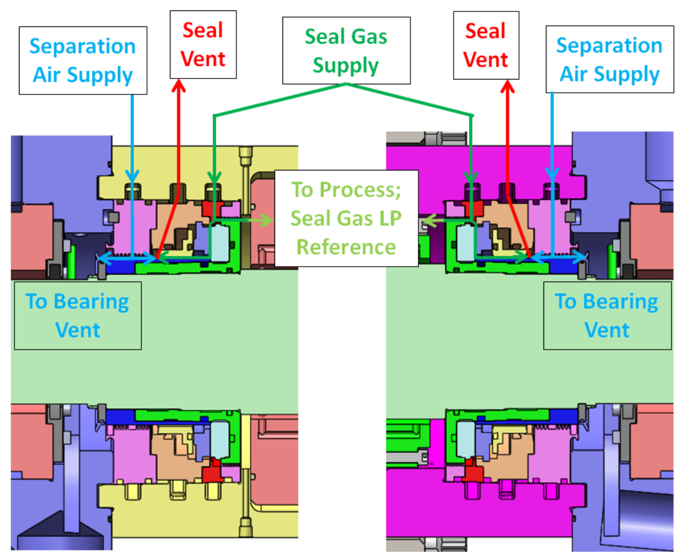


**Figure 21. Example Dry Gas Seal Detail Views and Flow Diagrams**

Dry gas seals also contain temperature-limited elastomeric or polymer seals for a sliding static seal on the stationary ring. Due to temperature limits of this static seal, most dry gas seals have a maximum operating temperature of 350 °F (177 °C). This poses a number of risks for the DGS in the event of a shutdown or failure for hot sealing in typical $sCO_2$ turbine 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. The high leakage in this scenario will also cause high backpressure at the seal vent and may cause hot gas to enter the bearing cavity (with risk of oil heating and fire) if the vent system is improperly designed.

A DGS supply system for an sCO2 turbine should contain the following elements:

- Filtration to prevent particles and liquids from entering the seal through the supply.
- Pressure and flow measurement to ensure adequate buffering and cooling flow.
- A control valve (potentially an individual valve for each seal), operating independently on feedback control to meet the seal's mass flow requirements for thermal management and process buffering.
- A heater to prevent dry ice formation in the seal vent. Due to high specific heat near the critical point, this heater duty may larger than for typical gas compressor seals. The heater operates on feedback control to achieve a supply temperature set point measured at the turbomachine's gas supply connection.
- Differential pressure regulators on the separation air supply to ensure proper oil buffering.

A backup supply system is also required for a DGS that is typically fed by the compressor discharge pressure to provide seal gas when the compressor is shut down during loop fill, pressurized hold, and emergency trip scenarios. For $sCO_2$ turbines, this backup system is particularly critical since the DGS flow is used for cooling in addition to buffering.

## OXY COMBUSTION GAS TURBINES

A direct-fired $sCO_2$ oxy-combustion cycle is highly attractive for power generation applications because of the cycle's inherent carbon capture, potential for higher firing temperatures and cycle efficiencies than for indirect cycles, and small machinery footprint. The absence of nitrogen in the fluid stream also eliminates NOx formation. However, there is a large amount of uncertainty regarding the combustion process of natural gas in carbon dioxide diluent at supercritical pressures, and well-developed combustion mechanisms at these conditions are not currently available or supported by experimental validation. These data are critical for combustor design, which depends heavily on the timescale of chemical reactions. Oxidation of fuel is a finite rate process and a minimum residence time is required for high combustion efficiency and low emissions. As shown in Figure 22, the well-developed mechanisms presented in the literature are validated for combustion in air ($N_2$ diluent) at pressures less than 20 bar. This is the space most relevant to conventional combustion applications and there is plenty of validation data available. Data become more limited outside of the low-pressure, low-$CO_2$ corner and are primarily limited to low-pressure oxy-combustion (flue gas recirculation) applications and high-pressure rocket combustion applications. Although there are almost no data available in the literature at high pressure, high $CO_2$ conditions, although a number of research and experimental programs are underway.

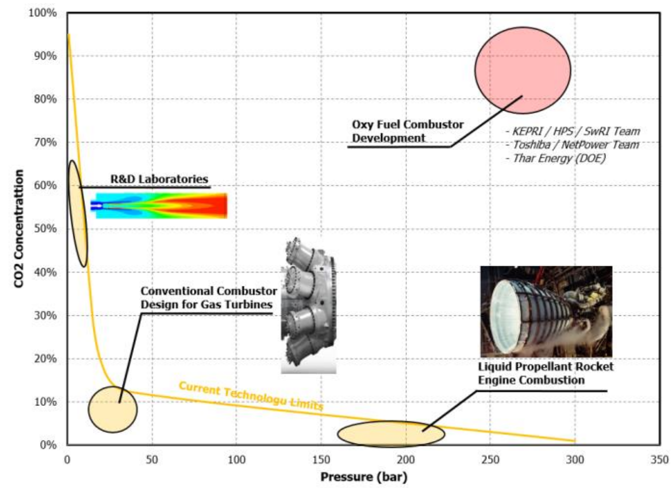


**Figure 22. Combustor design regime for oxy-fuel applications.**

The oxy-combustion process results in combustion byproducts in the waste stream which build up to equilibrium levels based on cleanup and venting from the semi-closed cycle. In addition to uncertainty in the underlying kinetics, there is also significant uncertainty in gas properties (particularly near the critical point), cleanup system performance, and material corrosion at high temperatures in $sCO_2$ mixtures with combustion byproducts.

## CONCLUSIONS

Various super-critical $CO_2$ turbomachinery related applications are currently under development for: waste heat recovery, direct power generation, and combined power cycles. Turbomachinery for the $sCO_2$ cycles have unique challenges related to the high pressures of the $CO_2$ and the operating demands of the various applications. The material presented herein is intended to provide the reader with the fundamental knowledge to understand the basic cycles and the turbomachinery for those cycles. Several areas high-lighted are: trades between various cycles, of turbine inlet temperature, and turbomachinery sub-component limitations and needs.

## NOMENCLATURE

| | |
|---|---|
| Nox | Nitrous Oxide |
| $sCO_2$ | Super-critical Carbon Dioxide |
| WHR | Waste Heat Recovery |

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