Tutorial: Turbomachinery Design and Operation for Supercritical CO2 Applications

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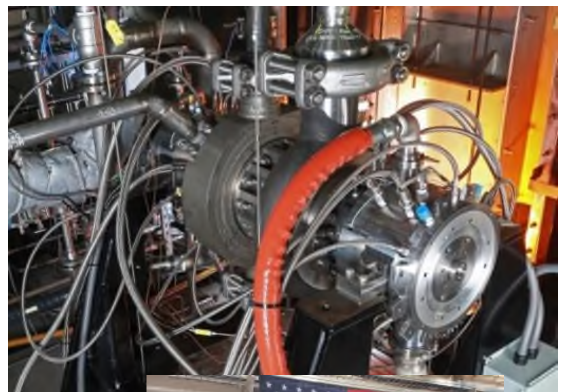
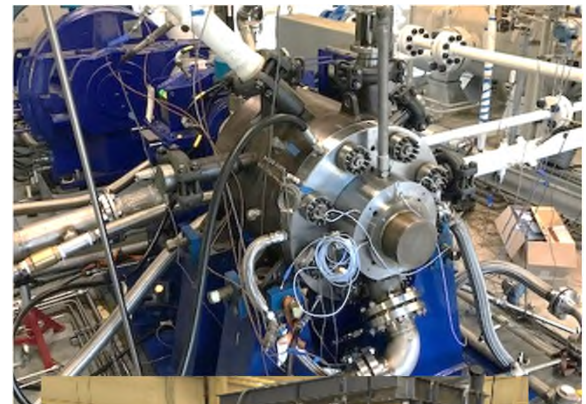
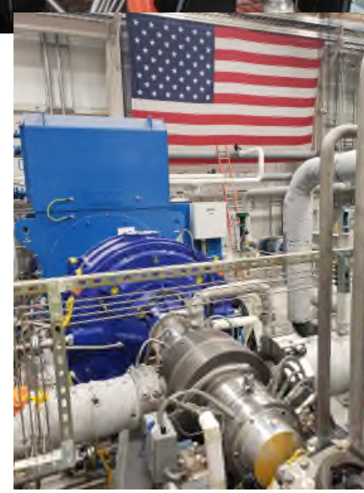
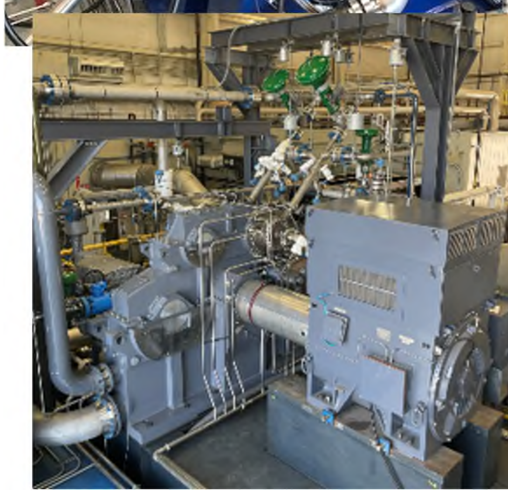








Abstract

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.

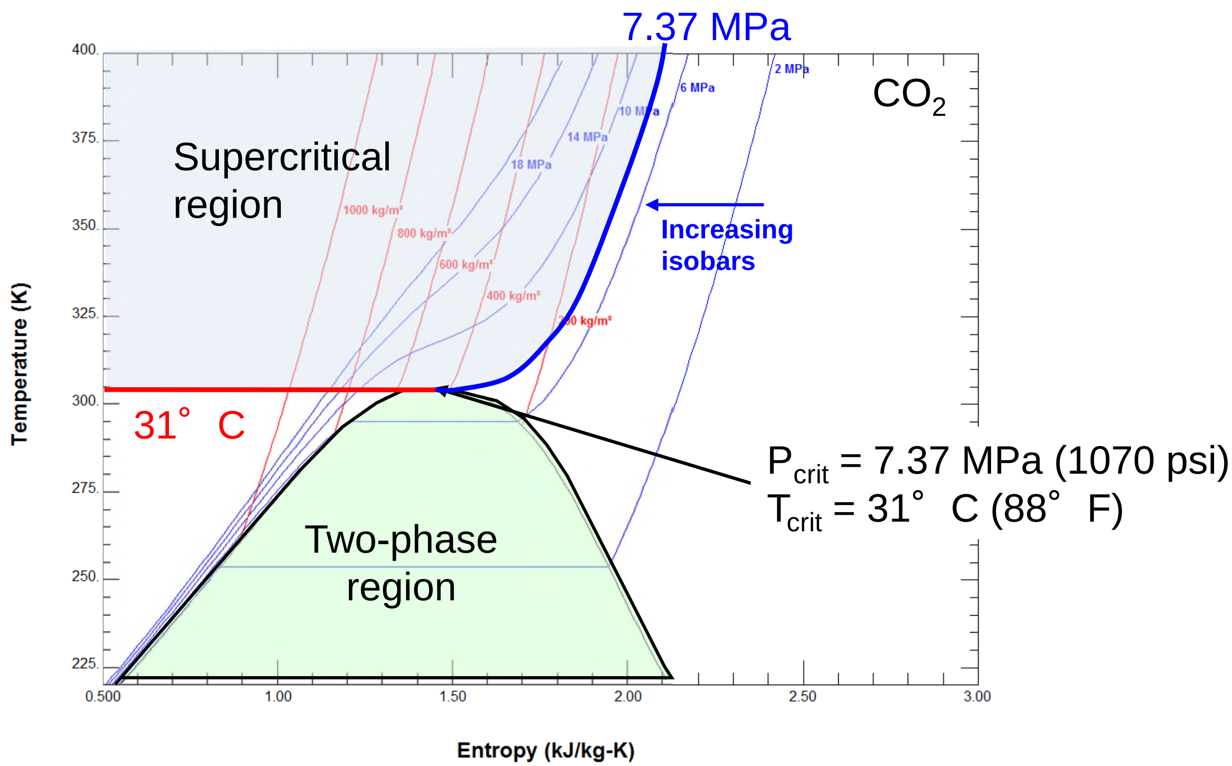
The aim of this course is to provide a basic understanding of the technologies and concepts that affect the design and operation of machinery with supercritical carbon dioxide as a working fluid. The participants will learn about the application requirements, working principles and commercial maturity of sCO_2 turbomachinery, and what technical research is needed to advance the state of the art in this emerging industry.

Outline

- Introduction to Supercritical CO_2
- sCO_2 Cycles
- Machinery Types and Layouts
- Existing Machinery
- Aerodynamic Requirements & Design
- Seals and Bearings
- Rotordynamics and Blade Dynamics
- Thermal Management
- Materials and Pressure Containment
- Test Facilities

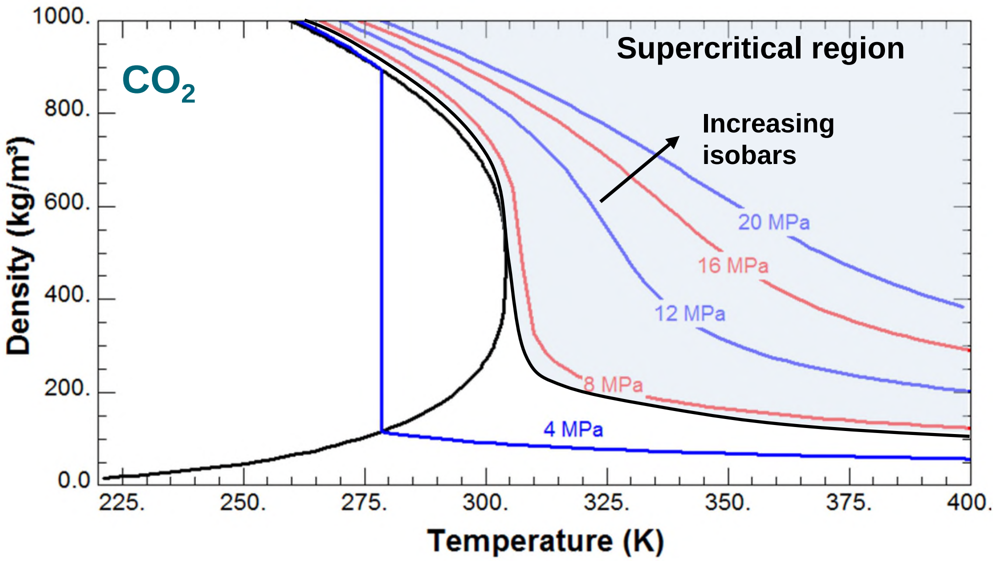
Introduction to sCO_2

A fluid is supercritical if the pressure and temperature are greater than the critical values



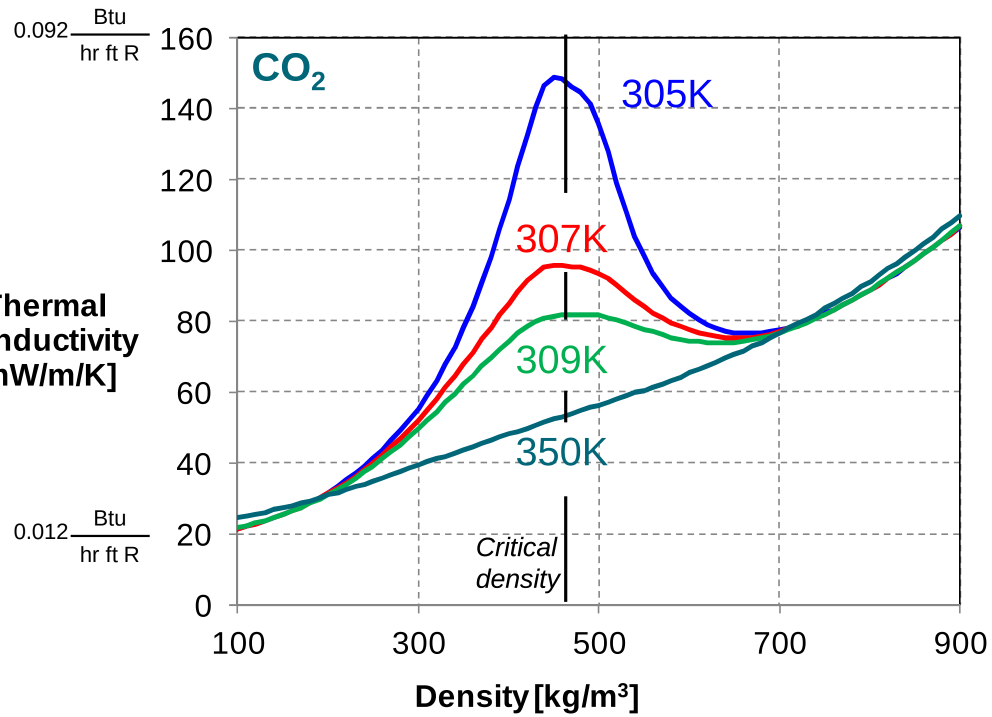
REFPROP (2007), EOS CO_2: Span & Wagner (1996)

Fluid density sharply decreases near the critical point



REFPROP (2007)

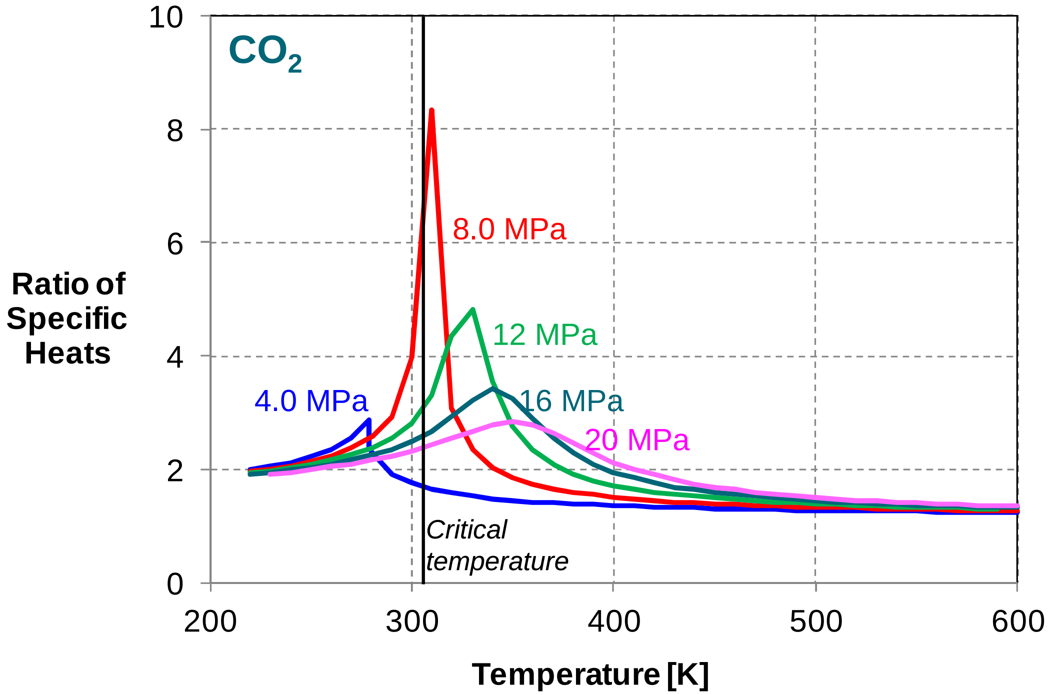
Fluid thermal conductivity is enhanced near the critical region



REFPROP (2007)

Water @ 304K = 620 mW/m/K
Atmospheric Air @ 304K = 26 mW/m/K

The ratio of specific heats peaks near the critical region

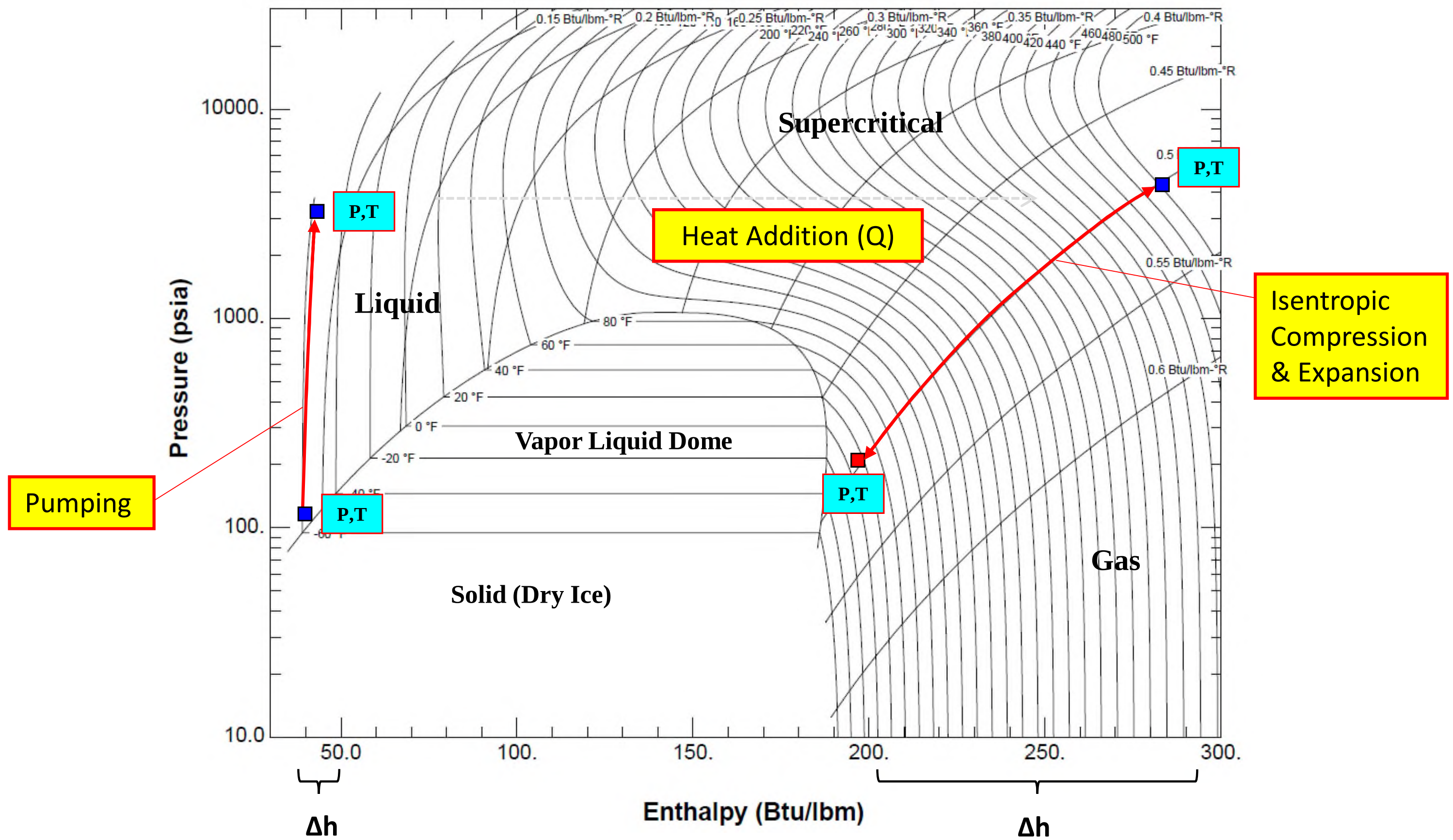


REFPROP (2007)

Air = 1.4

sCO_2 Cycle Overview

Power Cycles: *Compression, Heating, Expansion*



$W = \dot{m}\,\Delta h$ $\quad W_{net} = \dot{m}\,(\Delta h_{out} - \Delta h_{in})$ $\quad \eta = W_{net}/Q$

Applications

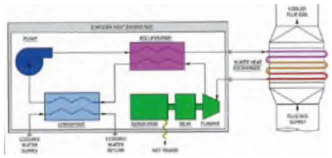
<u>Primary Power</u>
- High grade heat
- Optimized for system efficiency

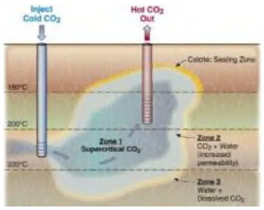

Concentrating Solar Power


Fossil Fuel


Nuclear

<u>Bottoming Cycles</u>
- Low grade heat
- Optimized for net power


Waste Heat Recovery



Geothermal

Primary Power

Primary Power
- High grade heat
- Optimized for system efficiency

Concentrating Solar Power

Heat Source
- Solar collector
- Thermal Energy Storage System

Advantages
- Dry cooling for arid climates
- Compact power block for modular, in tower installation
- Higher efficiency than steam for T>600°C

Fossil Fuel

Heat Source
- Coal or Natural Gas "Boiler" (indirect)
- Oxy-combustion (direct)

Advantages
- Higher efficiency than steam for T>600°C

Nuclear

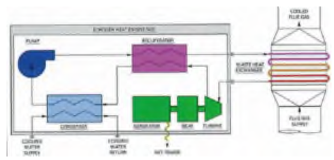
Heat Source
- Nuclear reactor

Advantages
- Higher efficiency than steam for T>600C
- Compact power block for SMR
- Direct cycles using CO_2 are possible

Bottoming Cycles

Bottoming Cycles

- Low grade heat
- Optimized for net power



Waste Heat Recovery

Heat Source

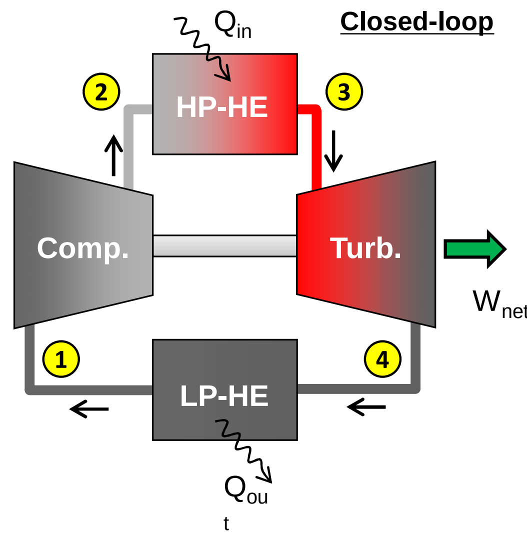
- Gas Turbine Exhaust
- Industrial Flue Gas

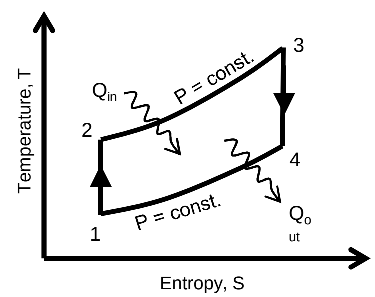
Advantages

- Dry cooling for arid climates
- Compact power block for modular installation
- Non-flammable working fluid (compared to ORC)
- Higher efficiency than 2 Pressure Non-reheat (2PNRH) steam cycles (GT Bottoming)
- Compact turbomachinery provides faster response to load transients than Steam Cycles

Brayton Cycle (Ideal)

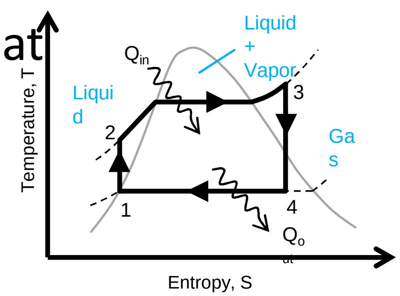
- Processes
  - (1-2) Isentropic compression
  - (2-3) Const. pres. heat addition
  - (3-4) Isentropic expansion
  - (4-1) Const. pres. heat reject.
- Open- or closed-loop

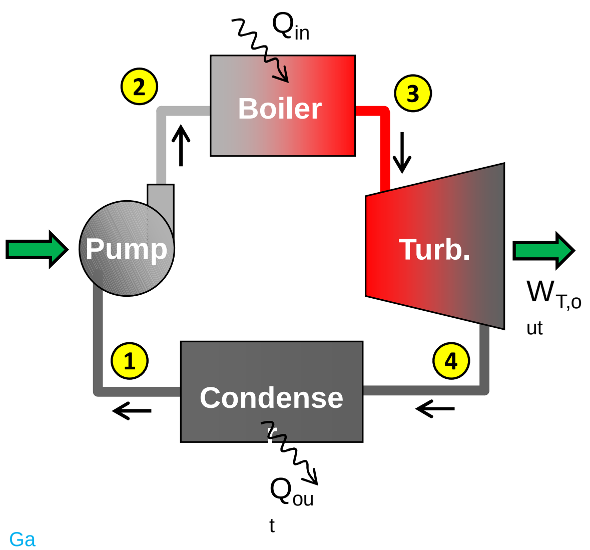




Rankine Cycle (Ideal)

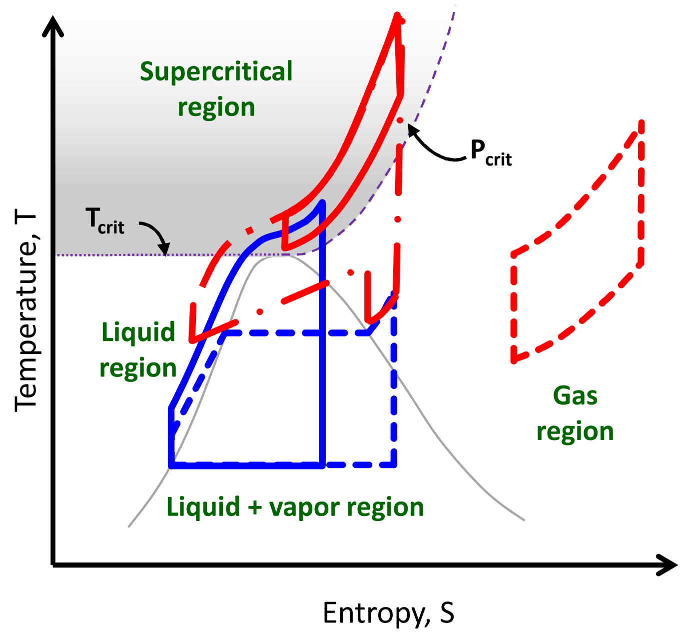
- Processes
  - (1-2) Isentropic compression
  - (2-3) Const. pres. heat addition
  - (3-4) Isentropic expansion
  - (4-1) Const. pres. heat reject.
- Same processes as Brayton; different hardware



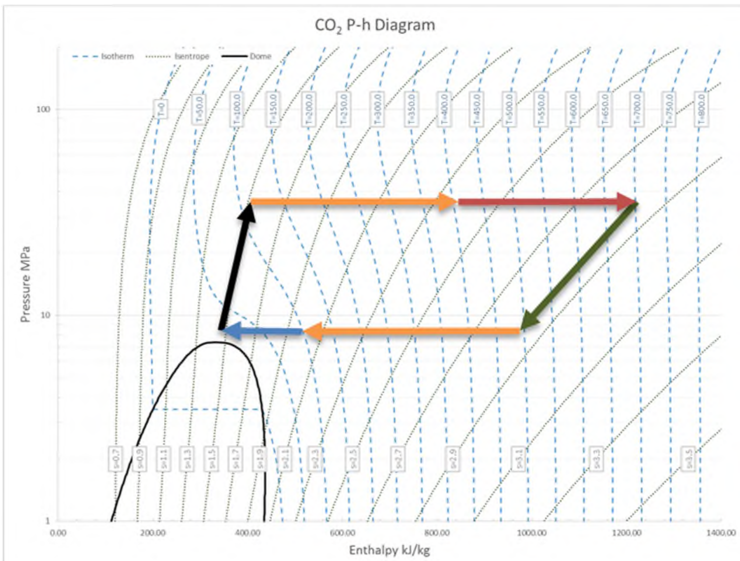


So what do we mean by Supercritical Carbon Dioxide (sCO_2) Power Cycles?

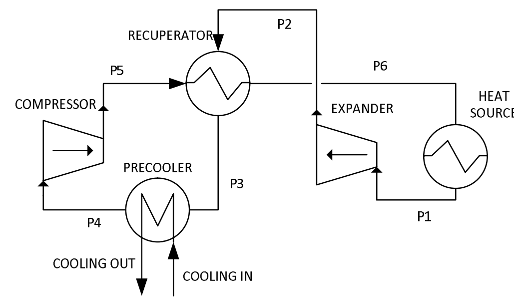
- CO_2 cycles come in multiple flavors
  - Supercritical vs Transcritical
  - Brayton vs. Rankine
  - Simple
  - Recuperated
  - “Compound” cycles



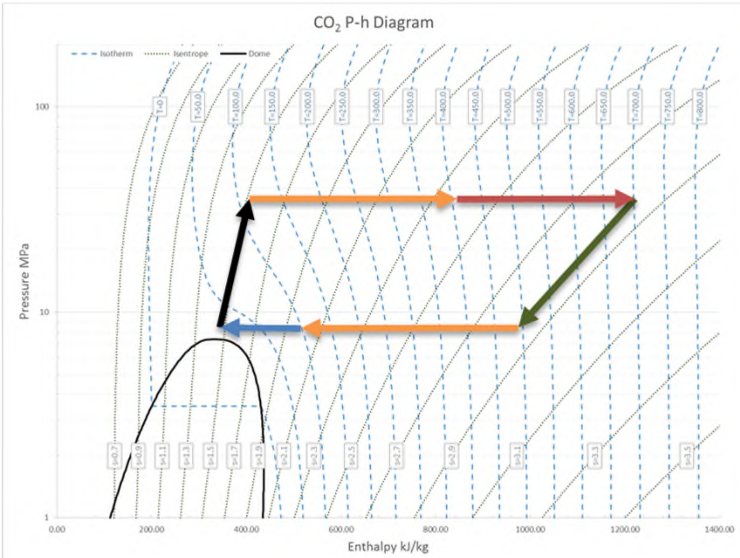
Simple Recuperated Cycle



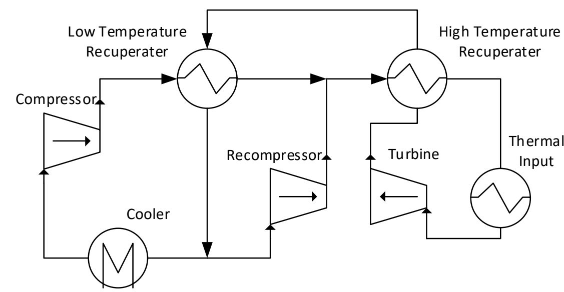
- Recuperated closed Brayton cycle
- Thermal efficiency limited by pinch point in the recuperator



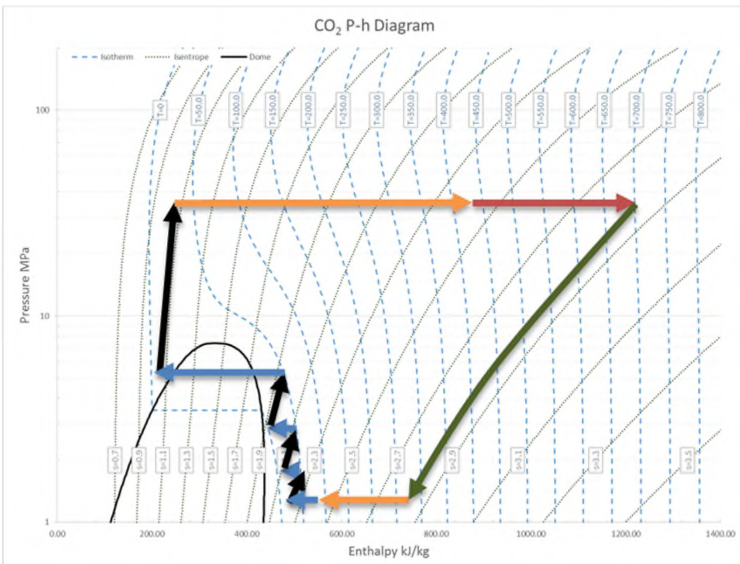
Recompression Cycle



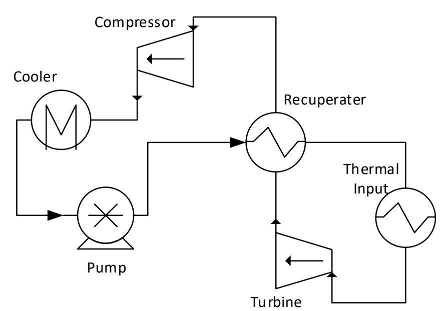
- Flow split used to minimize impact of pinch point in the low temperature recuperator
- Highly recuperated cycle achieves high cycle efficiencies
- Current benchmark cycle for CSP and Nuclear applications



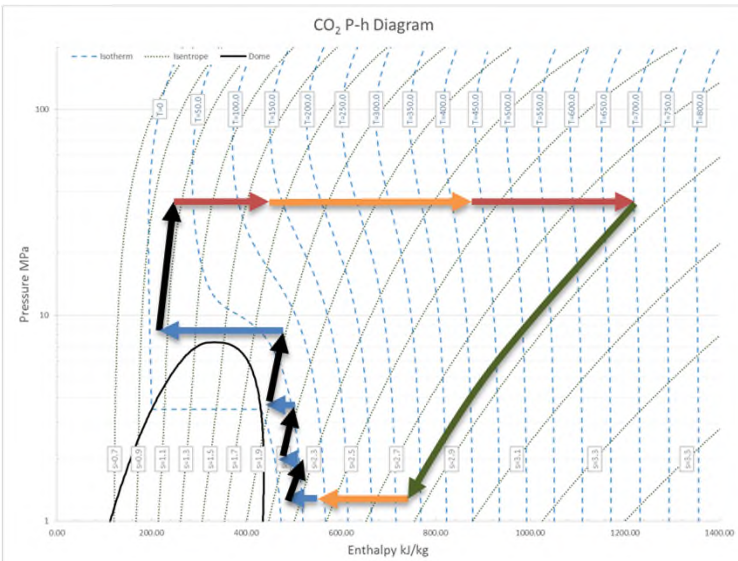
Condensing Cycles



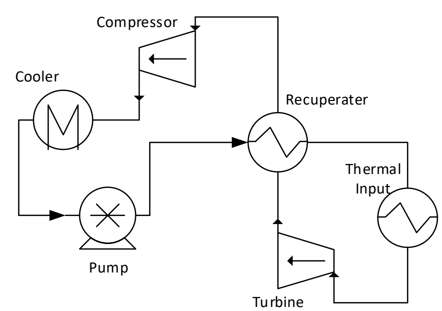
- Transcritical cycle
- Utilizes partial compression, condensation, and pumping to minimize compression work
- Must balance refrigeration / cooling requirements against compression requirements
- Can achieve efficiencies close to recompression cycle



Allam-Fetvedt Cycle (Oxy-Combustion)

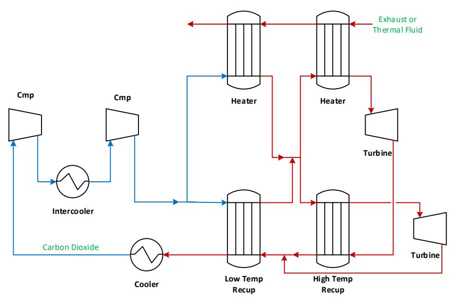


- Transcritical cycle
- Direct fired (oxy-combustion) cycle
- Uses iso-thermal compression to compress around the dome
- Recovers heat from integrated ASU

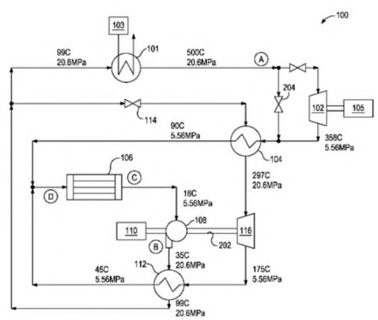


Cascaded Cycles

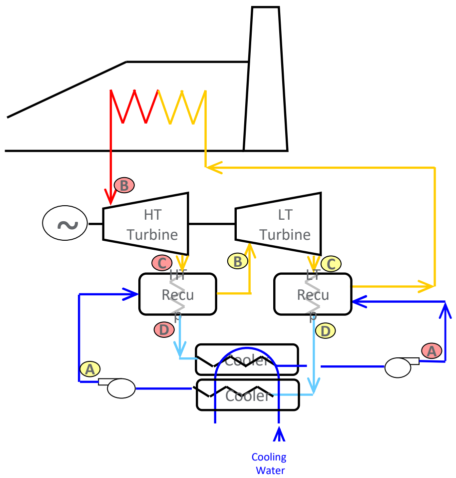
- Compound cycle configuration
- Designed to maximize thermal energy extraction from heat source (Large dT)
- Typically contain multiple turbomachinery trains
- Used for WHR and Fossil applications



"Fundamentals and Applications of Supercritical Carbon Dioxide Based Power Cycles", 2017

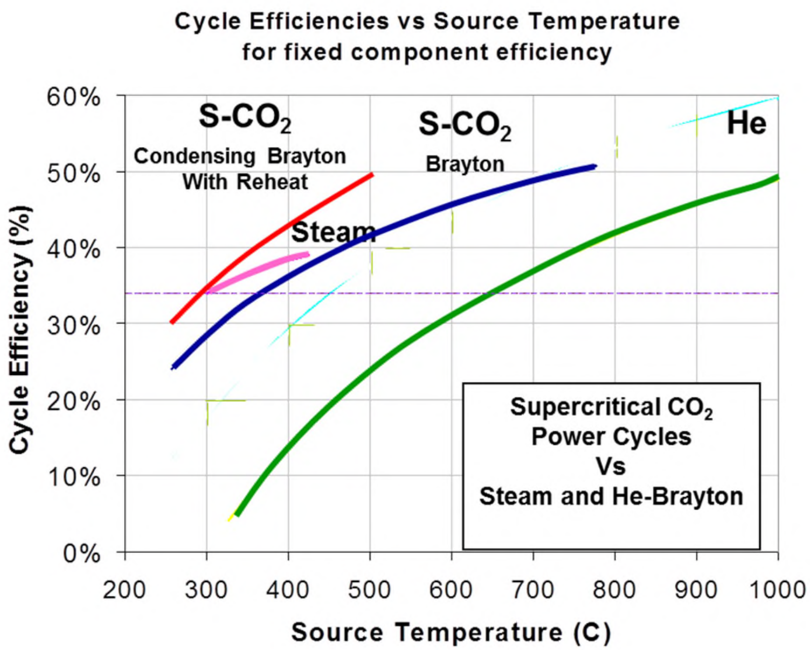


US Patent 20120131918 A1
Echogen Power Systems

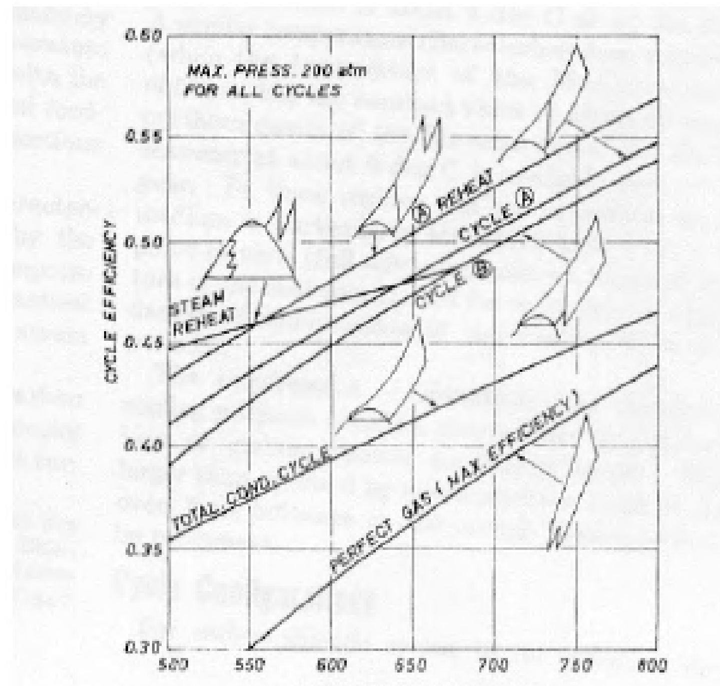


Courtesy GE Global Research

Comparison of sCO_2 Cycles with Alternatives

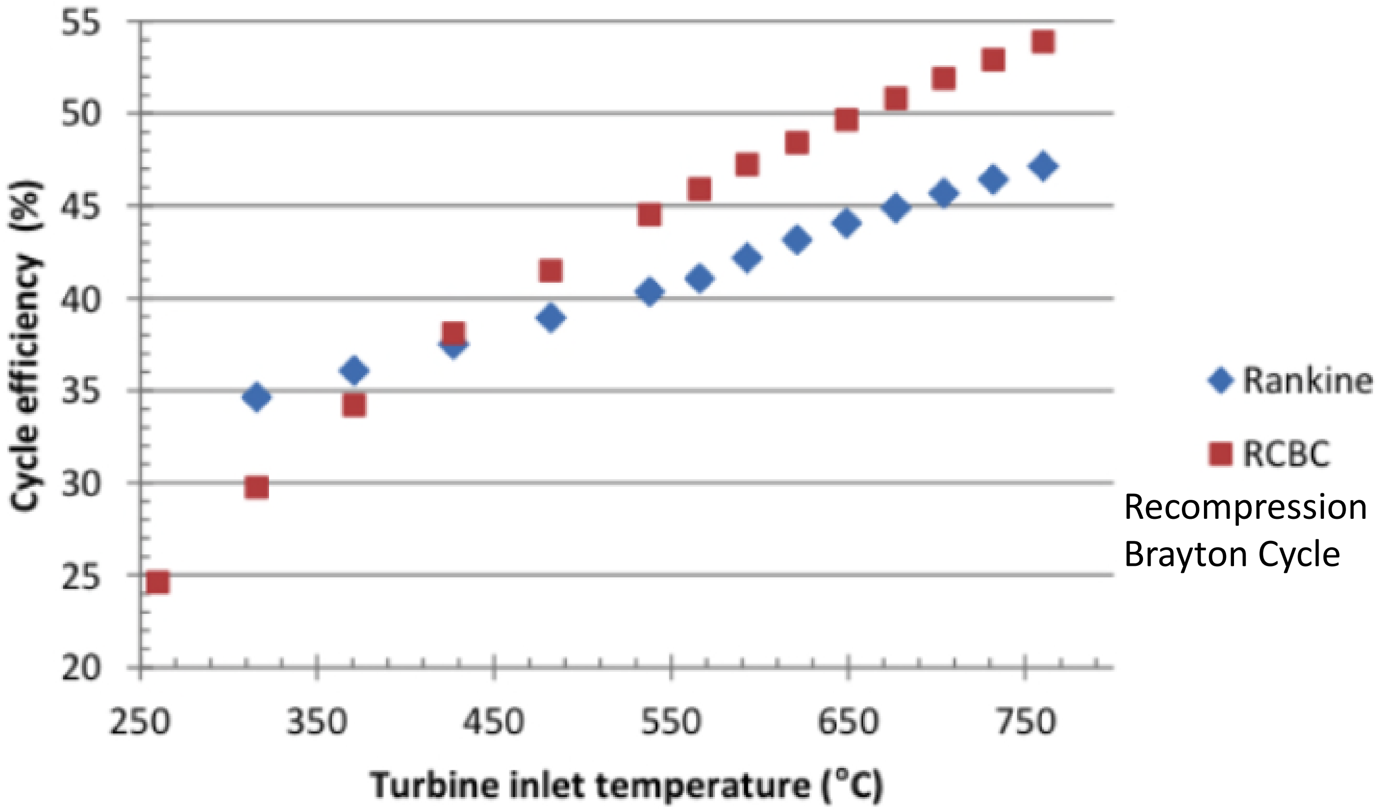


Wright, et. al. 2010



Angelino, 1969

Cycle Efficiency



Recompression
Brayton Cycle

sCO_2 vs. Steam – Why higher efficiency at high turbine temperatures?

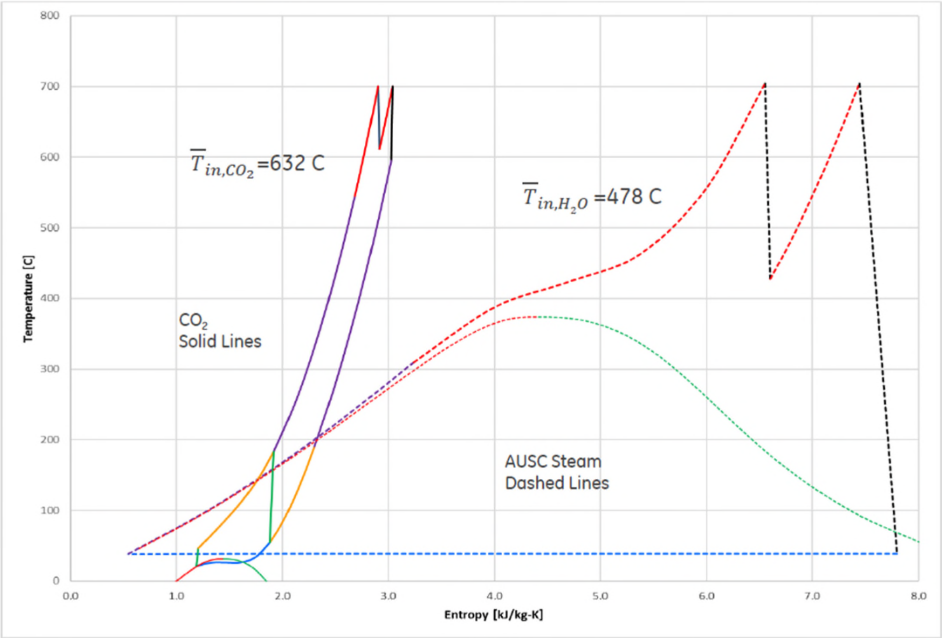
Fossil Cycle

$$\eta = 1 - \frac{\overline{T}_{sink}}{\overline{T}_{srce}}$$

Carnot Cycle Efficiency depends on **AVERAGE** heat addition and rejection temperatures

AVERAGE T_{in} for CO_2 is about 150C higher than for steam
→ ~6.5 points higher Carnot

That's Why

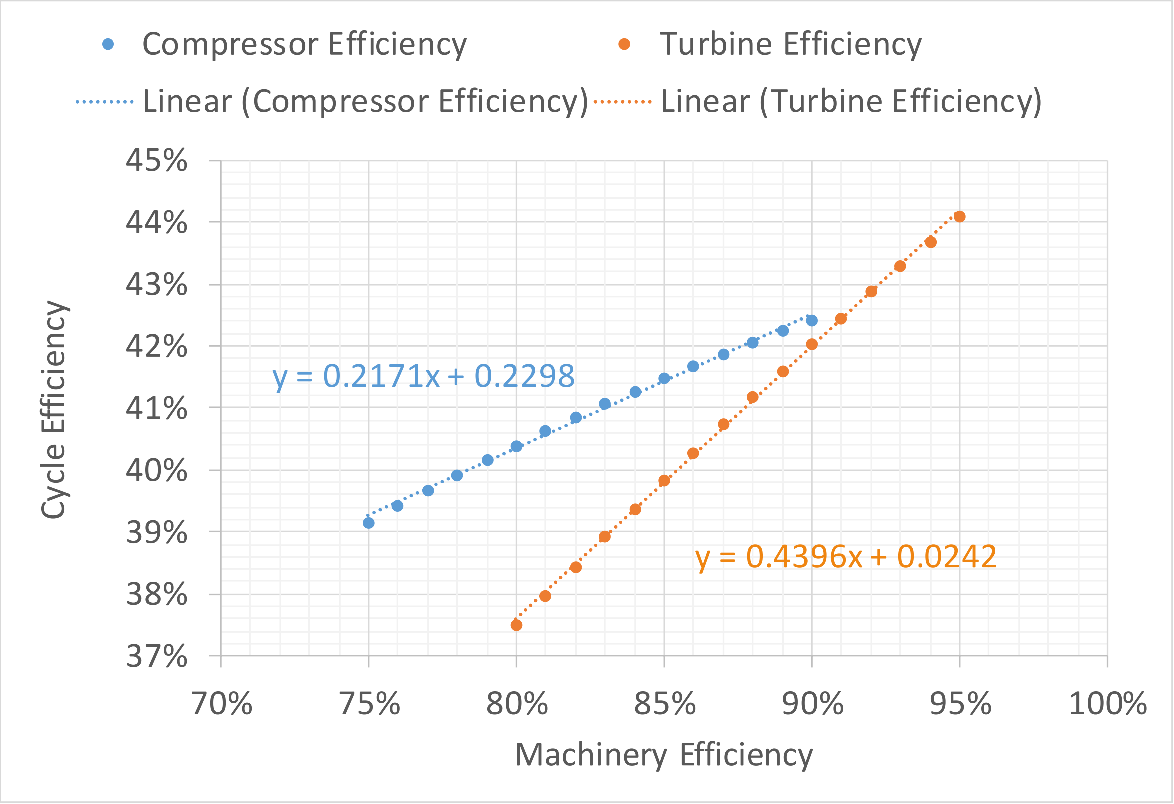


Comparison of a reheat sCO2 cycle to a reheat Advanced UltraSuperCritical (AUSC) steam cycle both with 700C turbine inlet temperatures

Note: Steam cycle is advantaged at T_{cold} due to constant T heat rejection in condenser

The lines cross at lower turbine temperatures because sCO2 heater inlet T floats with turbine exit T and steam boiler inlet T is fixed by feedwater heaters so T_{avg} comes down at ~1/2 the rate

Dependence of Cycle Efficiency on Turbomachinery Efficiency
(Simple Recuperated Brayton Cycle)



Layouts & Machinery Types

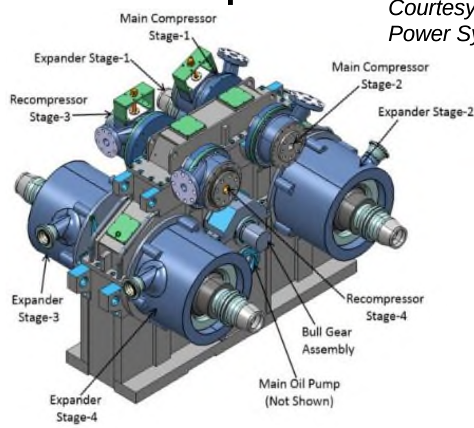
Turbomachines for Supercritical CO_2 Cycles

- Pumps
  - Low Speed Pumps
  - High Speed Pumps (Turbine Driven)
- Main Compressor
  - Much Like a Pump
  - May Operate Over Wide Inlet Density Range During Startup
- Re-Compressor
  - Standard Compressor Real Gas Compressor
- Expander
  - Radial
  - Axial

Integrally Geared vs. Barrel Compressor

Integrally Geared Isothermal Compressor

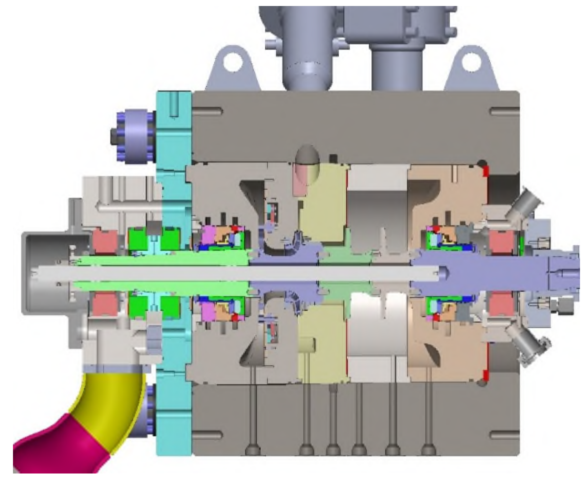
Courtesy of Hanwha Power Systems



Wilkes, J., Allison, T., Schmitt, J., Bennett, J., Wygant, K., Pelton, R., Bosen, W. (2016) "Application of an Integrally Geared Compander to an sCO_2 Recompression Brayton Cycle," in *The 5th International Symposium - Supercritical* CO_2 *Power Cycles*, San Antonio, TX.

- Integrally geared can achieve near isothermal compression
- Can contain up to 12 bearings, 10 gas seals plus gearbox
- Typically driven by electric motor
- Impellers spin at different rates
  - Maintain optimum flow coef.
- Capable of 280 bar

Single-Shaft Multi-stage Centrifugal Compressor



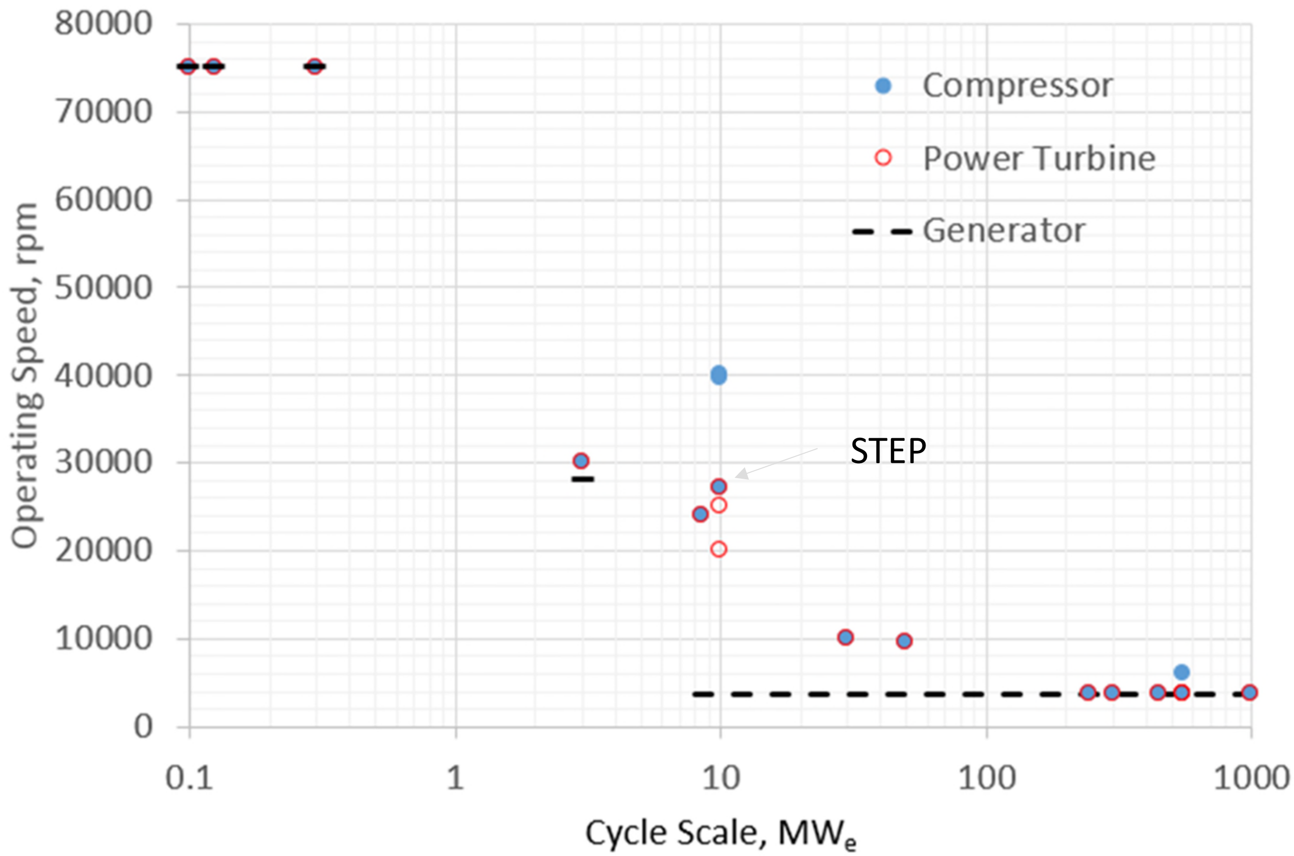
Cich, S., Moore, J., Kulhanek, C., Mortzheim, J., 2020, "Development and Testing of a Supercritical CO2 Compressor Operating Near the Dome, Proceedings of 49th Turbomachinery Symposium, Houston, TX, September 2020, Turbomachinery Laboratory, Texas A & M University

- Multi-stage centrifugal proven reliable and used in many critical service applications currently (oil refining, LNG production, etc.)
- Fewer bearings and seals
- Highest reported pressure of 550 bar in CO_2 and 900 bar in natural gas service

Generator Design

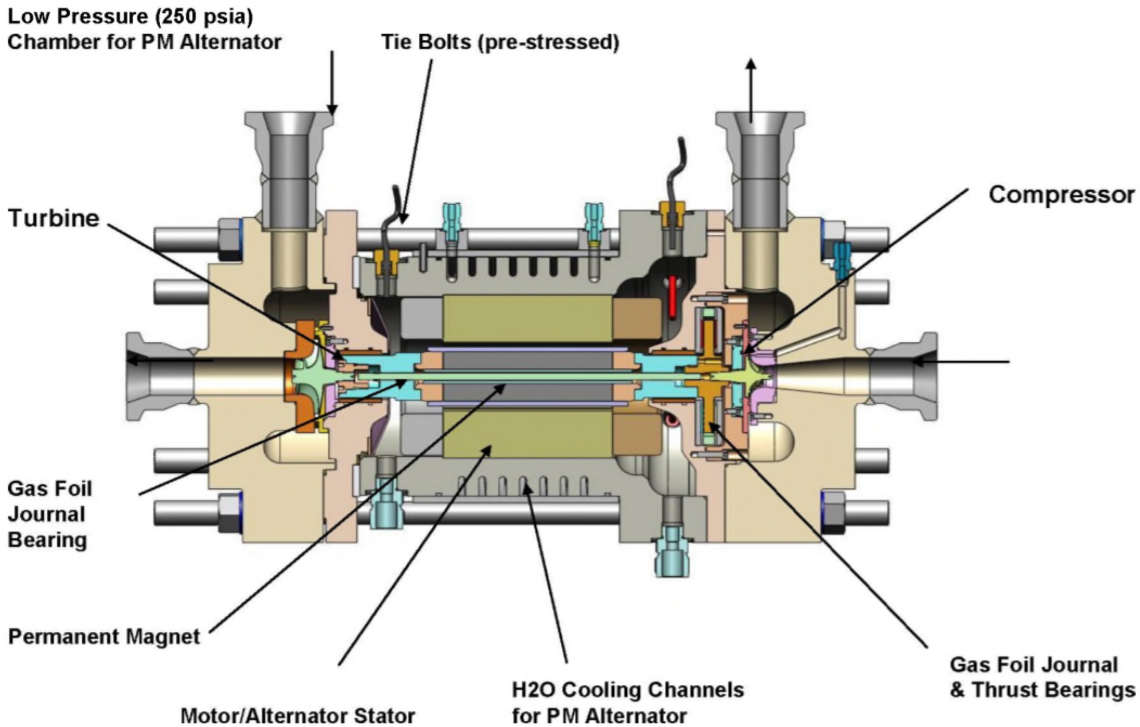
- Low-speed Synchronous
  - Low cost
  - Commercially available
  - Need to be combined with gearbox
- High-speed
  - May drive turbomachinery directly or use small gearbox if needed
  - Limited to 20,000 rpm
  - Lighter than low-speed option
  - More expensive

sCO_2 Turbomachinery Speeds and Scales in Literature



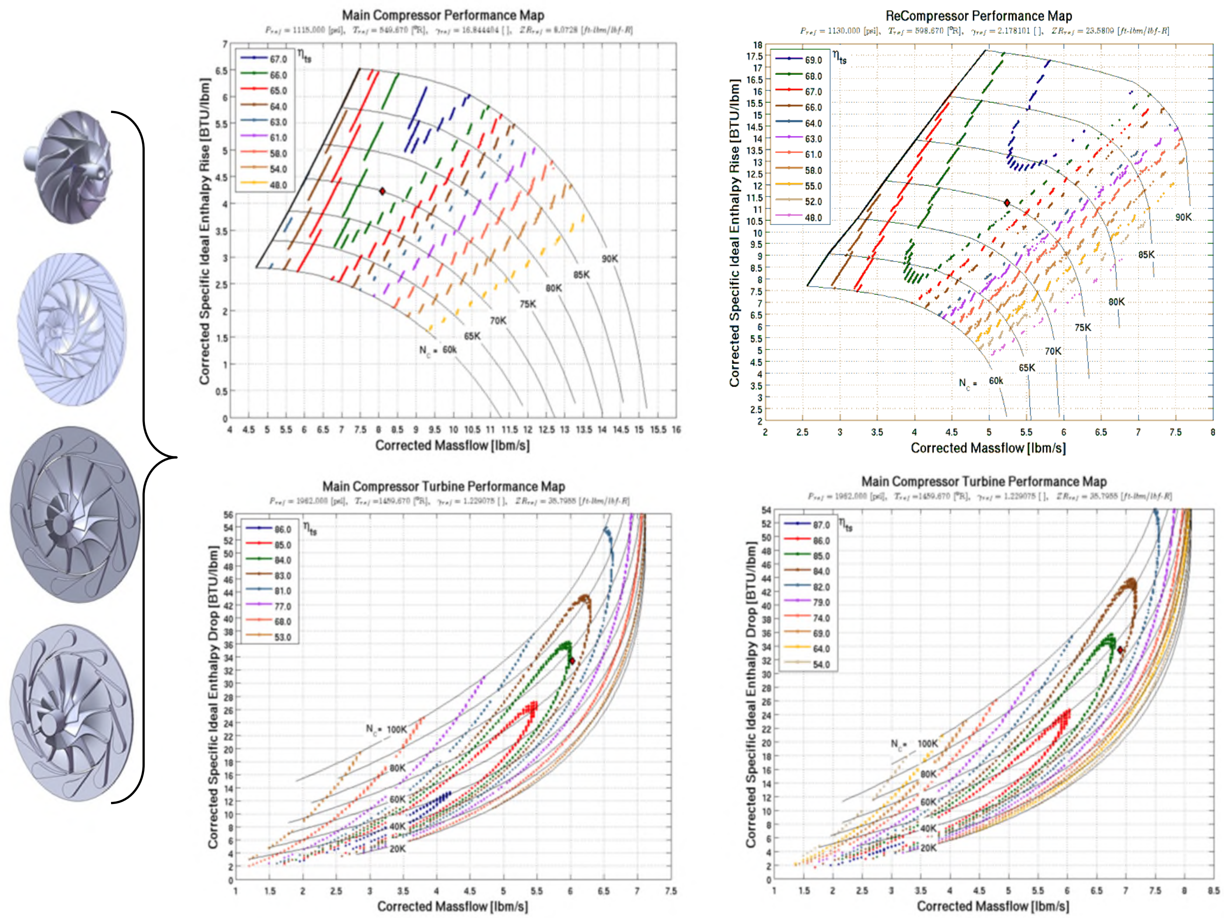
Existing sCO_2 Machinery

Barber-Nichols 100 kW Turbomachinery-Alternator-Compressor Typical of Sandia and IST Test Loops (Wright *et al.*, 2011)



Wright, S.A., Conboy, T.M. and Rochau, G.E. (2011) "Break-Even Power Transients for Two Simple Recuperated s-CO_2 Brayton Cycle Test Configurations", in *Supercritical* CO_2 *Power Cycle Symposium*, Boulder, CO.

Designs for Turbines and Compressors and Performance Maps



Courtesy of Barber-Nichols

TAC Operating Conditions

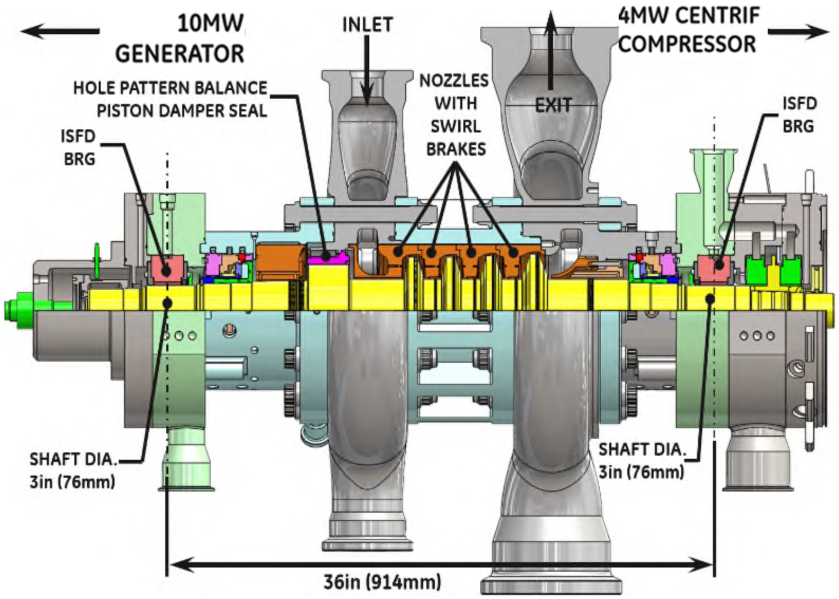
Component	Diameter [mm]	Design Inlet / Exit Pressure [bara]	Design Inlet Temperature [°C]	Design Flow Rate [kg/s]	Design Efficiency (t-s)	Design Isentropic Head [kJ/kg]
Sandia Main Compressor	37.3	76.9 / 141.1	32.2	3.67	66.5%	10.1
Sandia Re-compressor	57.9	77.9 / 140.1	59.4	2.27	70.1%	26.9
Sandia Main Compressor Turbine	68.1	135.8 / 83.4	538	2.47	85%	72.6
Sandia Re-compressor Turbine	68.3	136.8 / 83.3	538	2.88	85%	74.1
IST Main Compressor	38	92.4 / 166.7	36	5.5	61%	10.7
IST Compress-Turbine	53	164.5 / 95.8	299	2.5	80%	52.8
IST Power Turbine	53	164.5 / 95.8	299	2.8	80%	52.8

Experience with TAC Units
(Wright *et al.*, 2011)

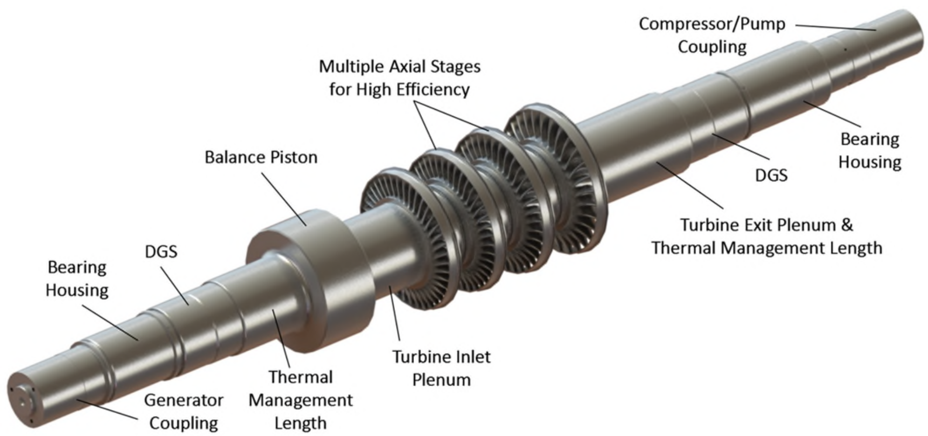
- Generally meet aerodynamic performance predictions
  - Low efficiency due to high relative tip clearances
- Integral design leads to high windage losses and evacuation power
- Modifications (turbine wheel cutouts and compressor pump out vanes) added for thrust balance
- Fouling and erosion (e.g., turbine nozzles)
- Motor controller challenges limit speed
- Gas bearing failures
- Thermal growth induced rubs

GE/SwRI SunShot 10 MWe sCO_2 Turbine (Moore *et al.*, 2015)

- 14 MW SCO2 Turbine Frame Size
- Tested to < 1 MW at 27,000 rpm
- 715C Turbine Inlet Temperature
- 280 bar case rating

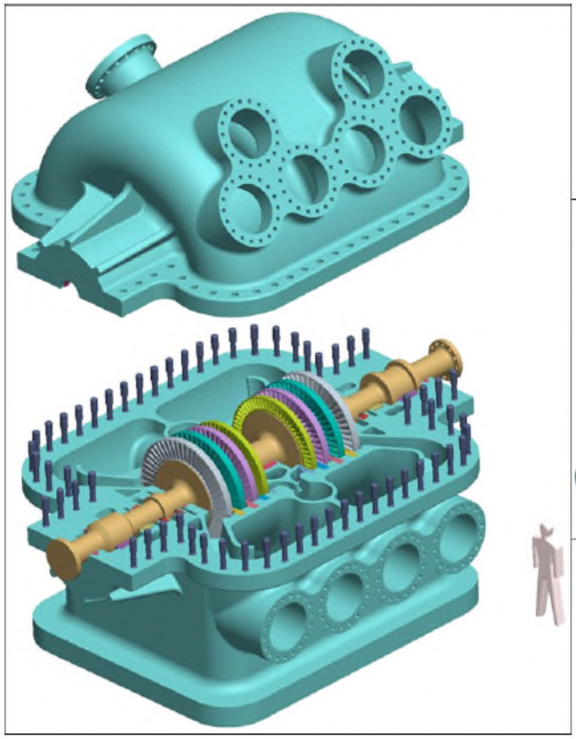


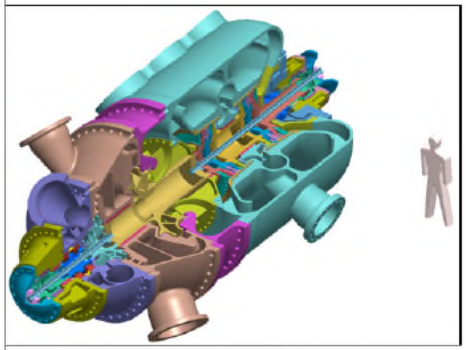
Ertas, B. Delgado, A., Moore, J., 2017, "Dynamic Characterization Of An Integral Squeeze Film Bearing Support Damper For A Supercritical Co2 Expander", Gt2017-63448, ASME Turbo Expo, Charlotte, NC



Moore, J.J., Cich, S., Day-Towler, M., Hofer, D., Mortzheim, J., 2018, "Testing of a 10 MWe Supercritical CO2 Turbine," Proceedings of 47th Turbomachinery Symposium, Houston, TX, September 2018

Gas Technology Institute (GTI) Dual-Shaft Turbomachinery Layout for a 1,000 MWe Recompression Cycle (Johnson *et al.*, 2012)



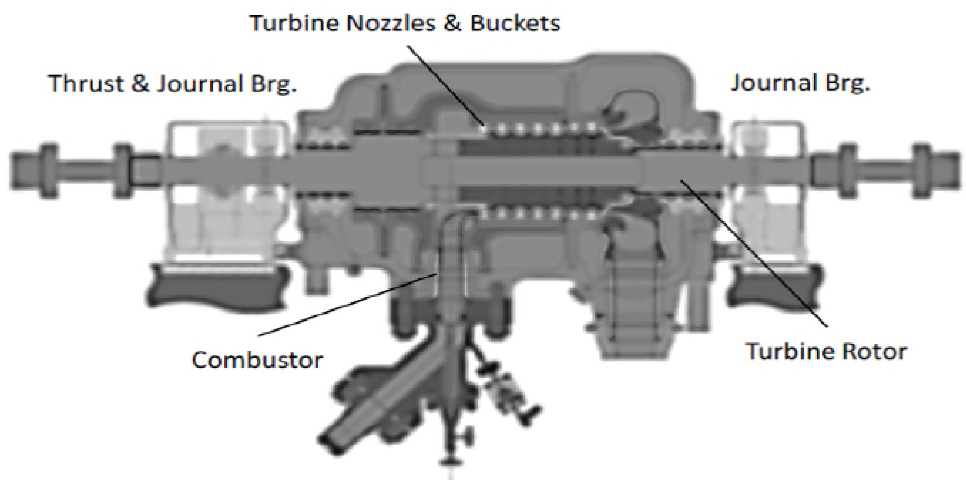


Compressor/Turbine

Johnson, G.A., McDowell, M.W., O'Connor, G.M., Sonwane, C.G., and Subbaraman, G. (2012) "Supercritical CO_2 Cycle Development at Pratt & Whitney Rocketdyne", *Proc. ASME Turbo Expo GT2012-68204*, Copenhagen, Denmark.

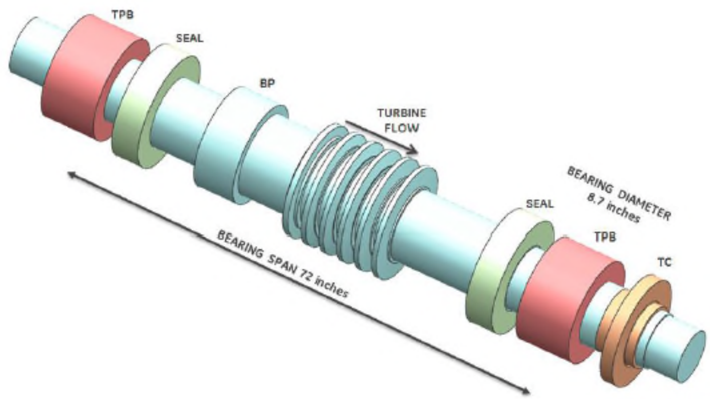
Toshiba Direct-Fired 25 MW sCO_2 Turbine Cross-Section for an Allam Cycle Demonstration (Iwai *et al.*, 2015)

- Turbine Inlet: 300 bara, 1150 C
- Cooled Rotor/Stator

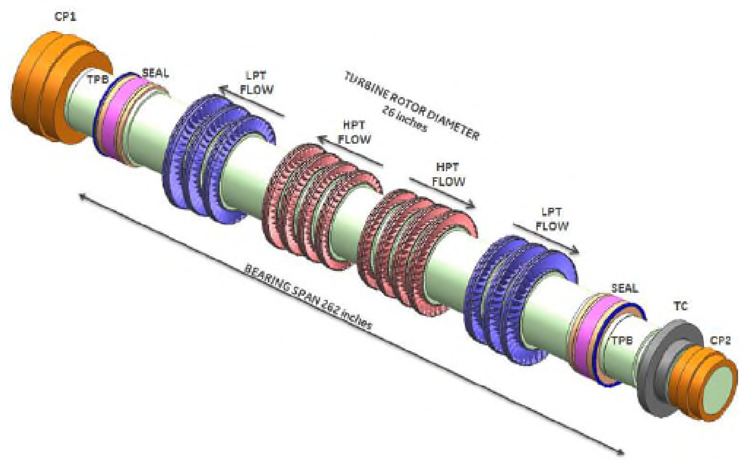


Iwai, Y., Ito, M., Morisawa, Y., Suzuki, S., Cusano, D., and Harris, M. (2015) "Development Approach to the Combustor of Gas Turbine for Oxy-Fuel, Supercritical CO_2 Cycle", *Proc. ASME Turbo Expo GT2015-43160*, Montreal, Canada.

GE / SwRI 50 and 450 MWe Trains



Single-Shaft Turbine Layout for a 50 MWe Recompression Cycle (Bidkar *et al.*, 2016a)

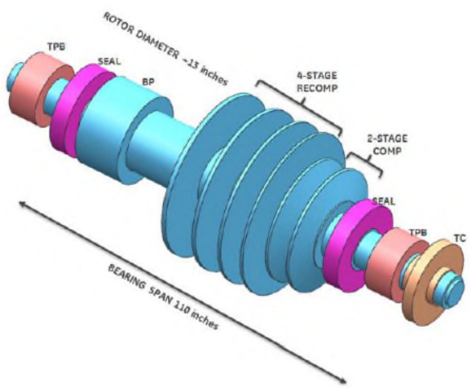


Single-Shaft Turbine Layout for a 450 MWe Recompression Cycle (Bidkar *et al.*, 2016a)

Bidkar, R.A., Musgrove, G., Day, M., Kulhanek, C.D., Allison, T., Peter, A.M., Hofer, D., and Moore, J. (2016) "Conceptual Designs of 50MWe-450MWe Supercritical CO_2 Turbomachinery Trains for Power Generation from Coal. Part 2: Compressors," in *The 5th International Symposium - Supercritical* CO_2 *Power Cycles*, San Antonio, TX, 2016.

450 MWe sCO_2 Power Cycle Compression Application

Single-Shaft Compressor Layout for a 450 MWe Recompression Cycle (Bidkar *et al.*, 2016b)

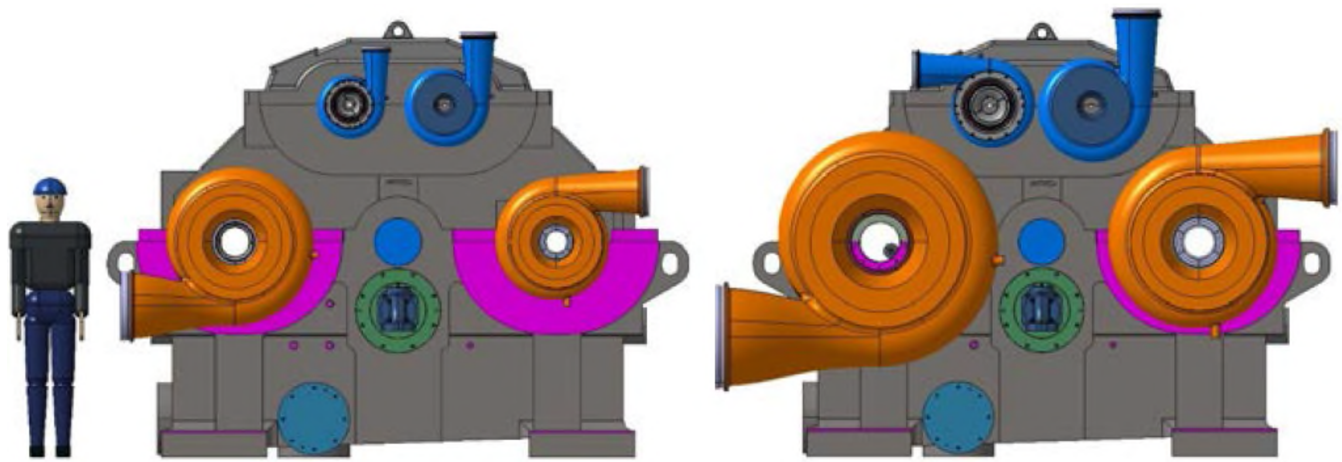


Compressor Design Details for a 450 MWe Recompression Cycle

Component	450 MWe Main Compressor	450 MWe Recompressor
Speed [rpm]	3,600	
Configuration	Centrifugal (back-to-back)	
End Seal Type	Dry Gas Seal	
Balance Piston Seal Type	Hole-Pattern	
Journal Bearing Type	Tilt-Pad Oil Bearing with Squeeze Film Damper	
Mass Flow [kg/s]	2134	1282
Inlet Temperature [°C]	21.5	54.8
Inlet Pressure [bara]	65.9	67.1
Number of Stages	2	4
Cycle Pressure Ratio	3.77	3.80
Isentropic Efficiency [%]	83.0	80.1
First Stage Specific Speed [-]	0.523	0.66
First Stage Specific Diameter [-]	5.28	4.46
Nominal Rotor Diameter [m]	0.33	
Bearing Span [m]	2.79	

Hanwha Techwin / SwRI Integrally-Geared Compander

Integrally-Geared sCO_2 Compander Layout for 5 MWe (left) and 25 MWe (right) Recompression Cycles (Wilkes *et al.*, 2016)

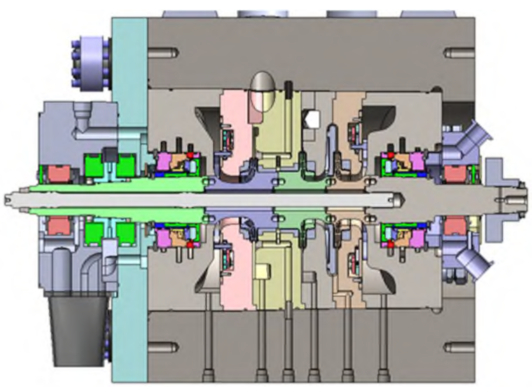


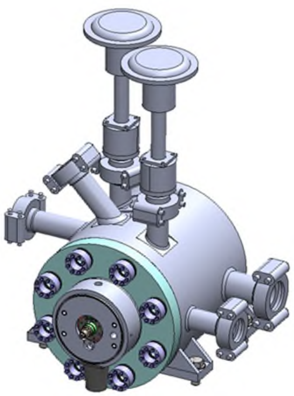
Wilkes, J., Allison, T., Schmitt, J., Bennett, J., Wygant, K., Pelton, R., Bosen, W. (2016) “Application of an Integrally Geared Compander to an sCO_2 Recompression Brayton Cycle,” in *The 5th International Symposium - Supercritical* CO_2 *Power Cycles*, San Antonio, TX.

SwRI/GE-Apollo High-Efficiency sCO_2 Centrifugal Compressor Development

PROJECT OBJECTIVES

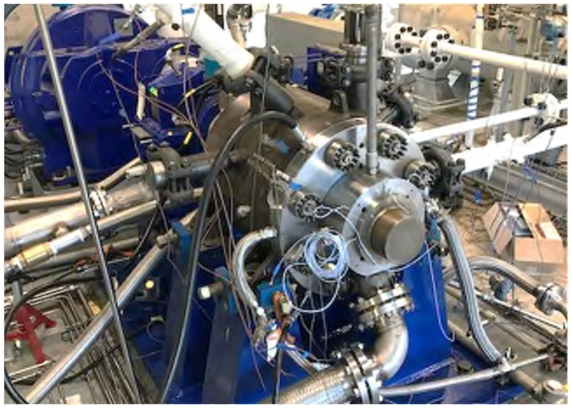
- Develop high-efficiency sCO_2 compression system
  - Main Compressor Efficiency of 80%
  - Preliminary Design completion June 2016
- High efficiency centrifugal impeller
- Variable inlet guide vanes (IGV)
- Advanced aerodynamic design provided by GE implemented into the detail compressor design provided by SwRI.





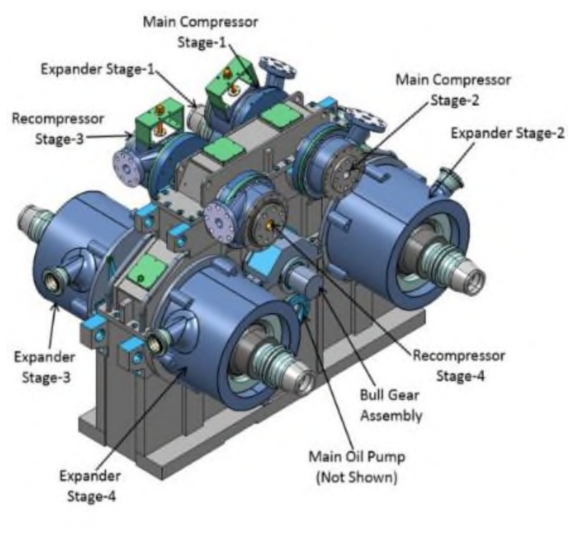
KEY RESULTS AND OUTCOMES

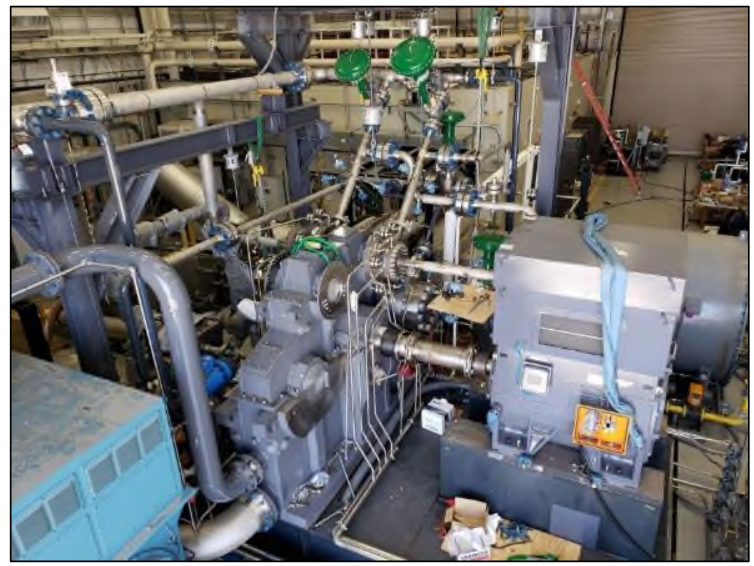
- Full scale testing of a 10 MWe sCO_2 Compressors
- Extended flow range to accommodate swings in ambient temperature
- SwRI sCO_2 Test Facility will verify compressor mechanical and aerodynamic performance over a range of operating conditions
- Testing completed October 2020



Hanwha Techwin / SwRI Integrally-Geared Compander

- Compander tested at SwRI 1 MWe SCO2 Test Loop (Wilkes *et al.*, 2016)
- Achieved 720C at 280 bar



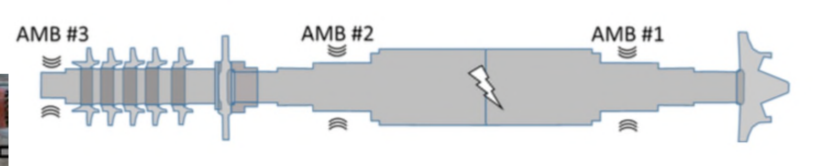


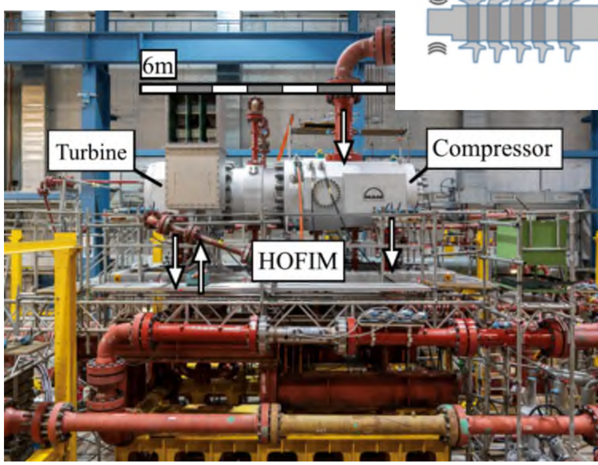
Wilkes, J., Allison, T., Schmitt, J., Bennett, J., Wygant, K., Pelton, R., Bosen, W. (2016) "Application of an Integrally Geared Compander to an sCO_2 Recompression Brayton Cycle," in *The 5th International Symposium - Supercritical* CO_2 *Power Cycles*, San Antonio, TX.

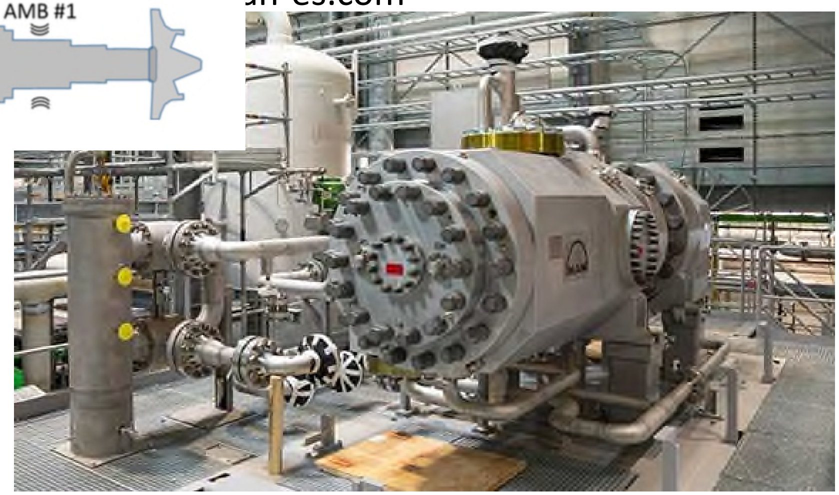
MAN-ES Heat Pump

- Uses SCO2 in a heat pump cycle
- Provides industrial and district heating up to 35 MWt
- Utilizes hermetically sealed motor/compressor and multiphase expander at 10 MW on magnetic bearings (based on HOFIM product)
- COP in 4 range

an-es.com



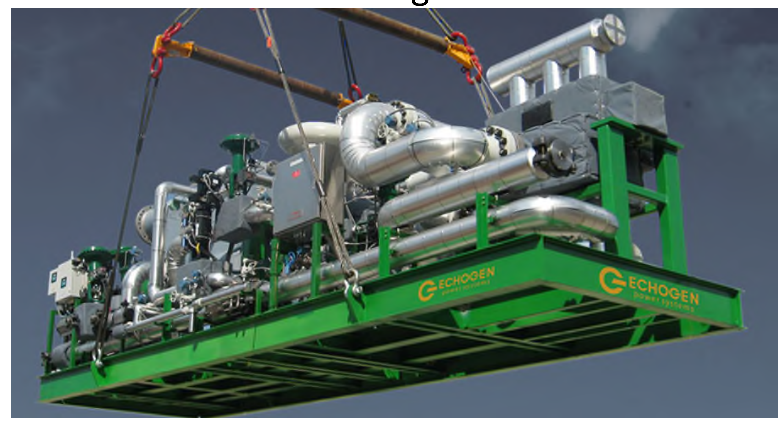




Wolscht, L., Somaini, R., Jacquemoud, E., Jenny, P., 2023, "Full Scale Demonstration and Validation of a 35 MW Transcritical CO2 Heat Pump," GT2023-101356, ASME Turbo Expo, Boston, MA

Echogen Waste Heat Recovery

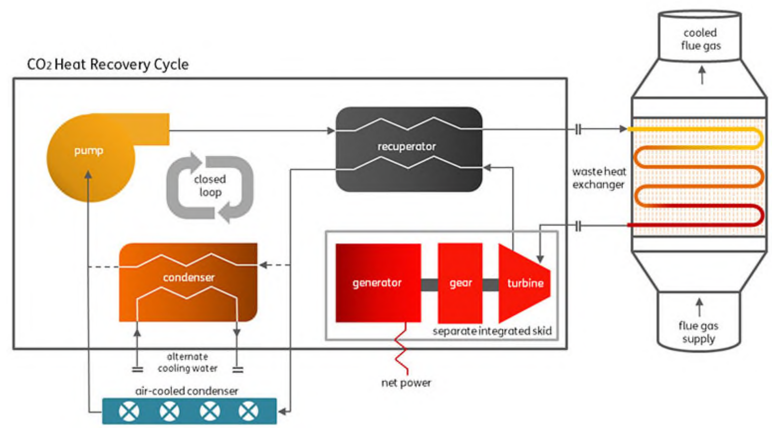
echogen.com



- SCO2 waste heat recovery power cycle
- Utilizes dense phase
- Power turbine electrical output up to 3.1 MWe for 330 hours

(max power at test stand conditions, limited by steam available)

10 MWe SUPERCRITICAL CO_2 POWER TURBINE



https://www.netl.doe.gov/sites/default/files/netl-file/FE0031585-Kickoff.pdf

Supercritical Transformational Electric Power (STEP Demo) Project

Demonstrate an integrated electricity generating power plant using transformational sCO2-based power cycle technology

Demonstrate pathway to efficiency **> 50%**

Demonstrate cycle operability at **>700°C** turbine inlet temperature and 10 MWe net power generation

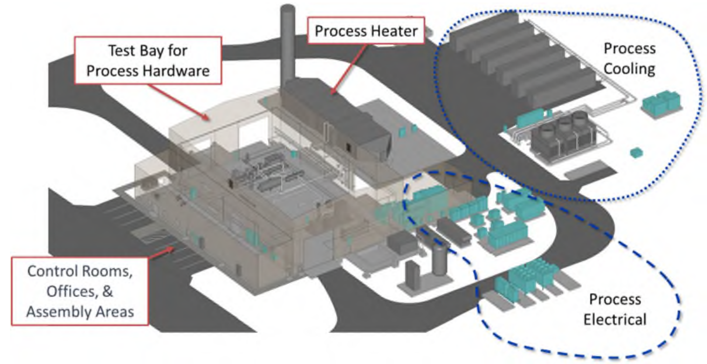
Quantify performance benefits:

- 2-5% point net plant efficiency improvement
- 3-4% reduction in LCOE
- Reduced emissions, fuel, and water usage

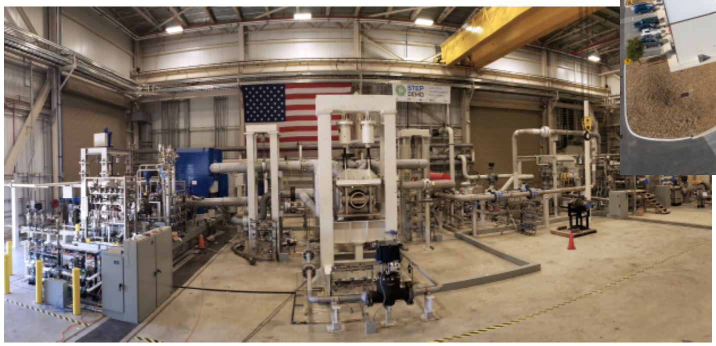
Develop a **reconfigurable and flexible test facility**

- Available for Testing future sCO2 equipment & systems

Achieved **mechanical completion** October 2023

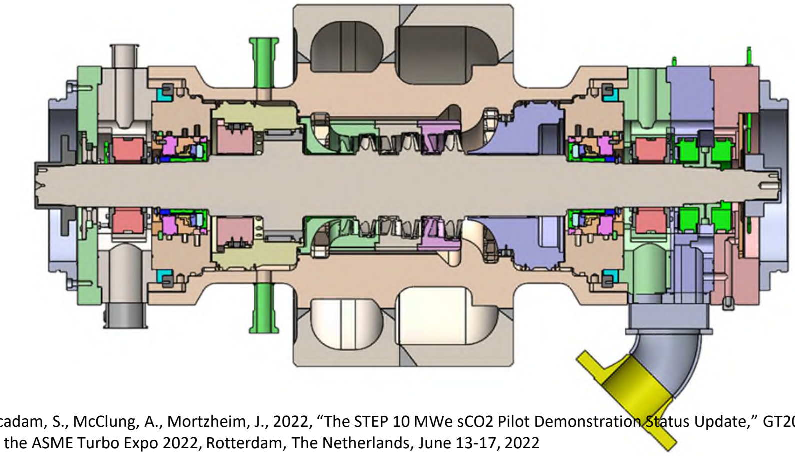


Marion, J., Macadam, S., McClung, A., Mortzheim, J., 2022, "The STEP 10 MWe sCO2 Pilot Demonstration Status Update," GT2022-83588, Proceedings of the ASME Turbo Expo 2022, Rotterdam, The Netherlands, June 13-17, 2022

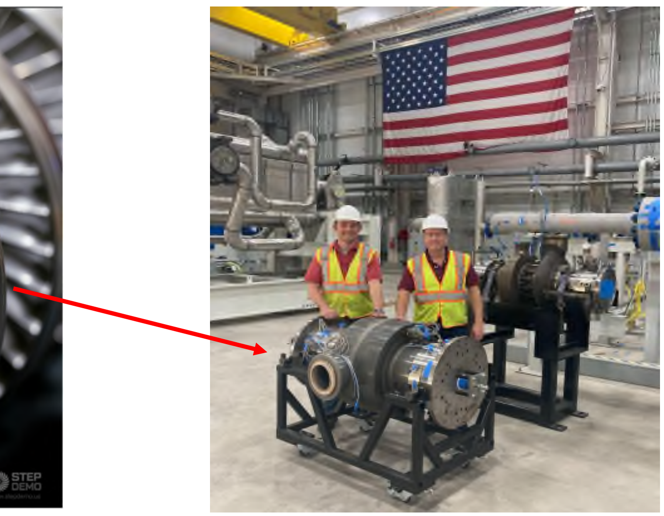


STEP Demo Turbine

- Advance Turbine from TRL 6 (Engineering Prototype) to TRL 7 (Full Scale Prototype)
- Based on Sunshot Design
- 3 stages, monolithic nickel alloy blade/shaft
- Full flow path, 16 MWsh gross power
- 26,650 rpm design speed
- Rotor weight = 85 kg
- Highest power density of any industrial turbine
- Inlet conditions: 250 bar, 715°C
- Fluid film bearings with SFD
- Dry gas seals
- Single Inlet / Single Outlet connections
- Fabricated Inconel 625 barrel style casing
- Assembled and first spin achieved December 2023
- 27,000 rpm at 280C turbine inlet conditions achieved to date
- Completed Simple Cycle Testing at 500°C in October, 2024



Marion, J., Macadam, S., McClung, A., Mortzheim, J., 2022, "The STEP 10 MWe sCO2 Pilot Demonstration Status Update," GT2022-83588, Proceedings of the ASME Turbo Expo 2022, Rotterdam, The Netherlands, June 13-17, 2022



STEP DEMO NATIONAL ENERGY TECHNOLOGY LABORATORY gti SwRI

STEP Simple Cycle Test Program

- Test program of increasing speed, temperature, and power

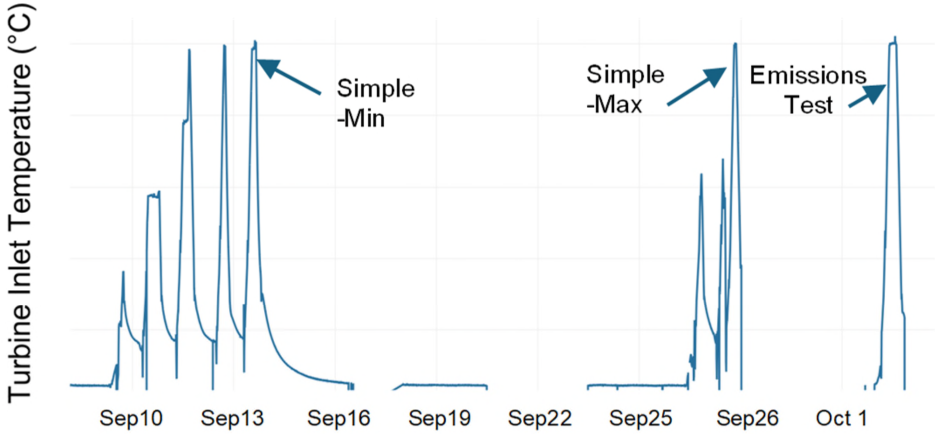
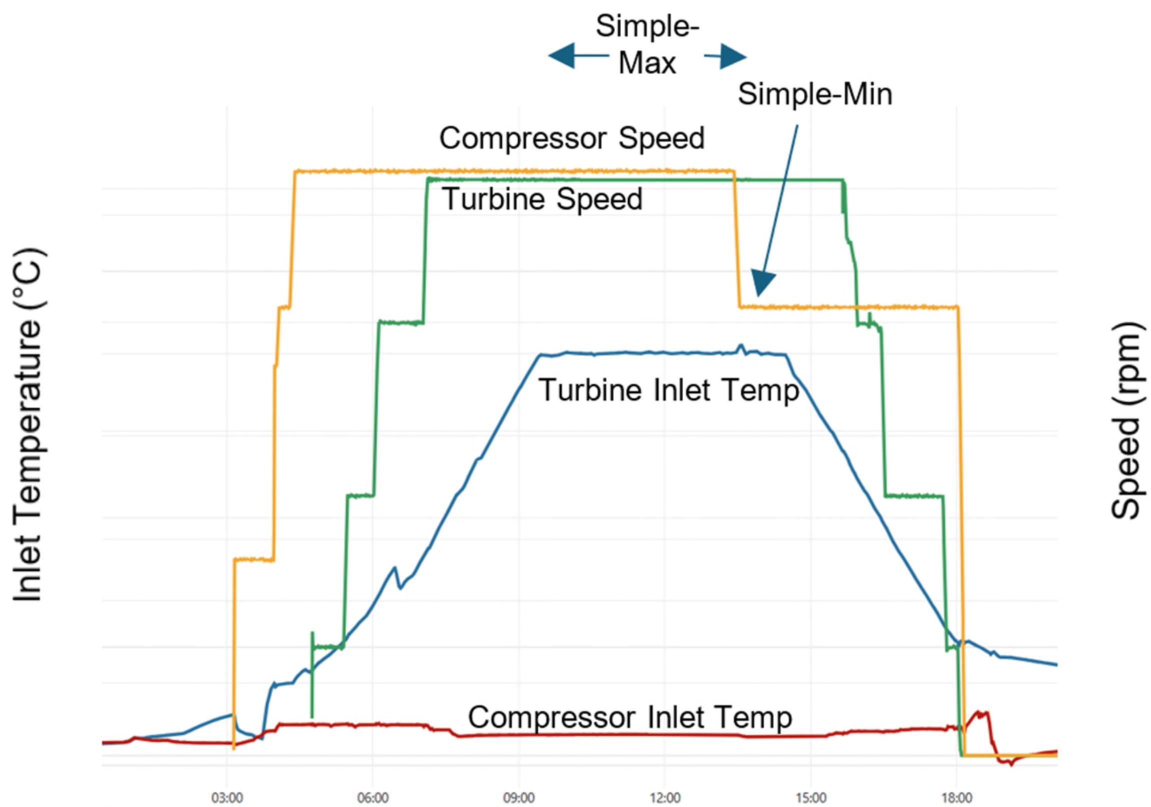


Figure 1. STEP Test Campaign Reaching 500 °C Turbine Inlet Temperature for Simple-Min, Simple-Max, and Emissions Test

Goal	Max Power	Trip
Verify turbine at full speed and high temperature with low load to check performance	100 kW on Load Banks	N/A
500°C TIT with Load Banks	150 kW on Load Banks	N/A
Ammonia injection in the heater for emissions reduction	290 kW on Load Banks	Turbine Gearbox vibrations
Achieve Simple Min	2.6 MW on Load Banks	Overvoltage resulting on a turbine overspeed
Grid synchronize	1.1 MW to Grid	No current measured on breaker bus
Achieve Simple Max	8.3 MW aero 7.4 MW generator 3.9 MW to Grid	N/A
Emission Testing at Simple Max, Repeat Simple Min	3.9 MW gross power to Grid	N/A

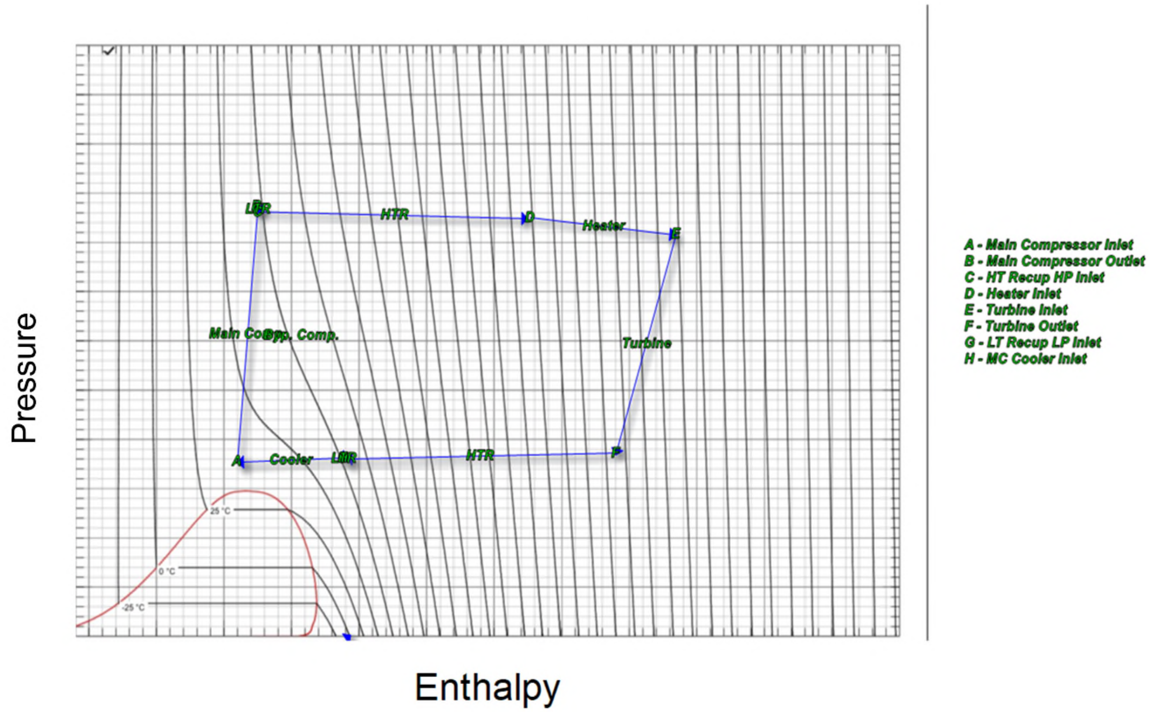
6-hour Emissions Test

- A 6-hour emissions test required to verify natural gas heater emissions test
- Both Simple Cycle maximum and minimum power conditions demonstrated



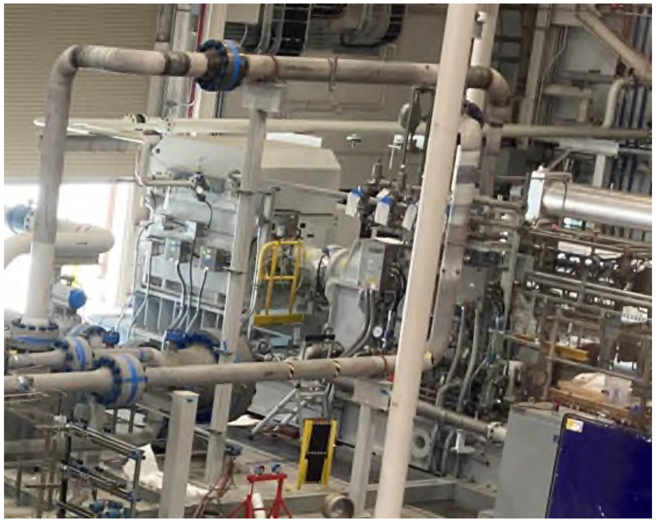
Simple Cycle Power Cycle

- Real-time pressure-enthalpy diagram measured and compared to predictions
- Turbine, compressor, and heat exchangers demonstrated good performance close to prediction



Baker Hughes SCO_2 STEP Compressor

- Supplied main and bypass compressors for STEP facility
- Equipped with variable inlet guide vanes for greater flow turn-down
- Driven by motor with VFD
- Commissioned in 2023
- Have achieved 275 bar maximum discharge pressure
- Discharge density 750 kg/m^3



Conclusions

- sCO_2 systems have progressed through many component and now system development activities to increase technology readiness level
- Low- and medium-temperature systems are commercially available and have active commercial projects via companies like GE Vernova, Baker Hughes, Hanwha, MAN Energy Systems, Echogen Power Systems, and others for waste heat recovery, heat pumps, and energy storage
- STEP 10 MWe demonstration has advanced simple recuperated sCO_2 cycles up to 500°C through full-scale pilot testing
- More hours of operation needed to fully validate the technology.

Aero Requirements & Design

sCO_2 Turbomachinery Aero Design

- Main compressor is primary challenge due to strong real gas properties near dome
  - Equations of state yield errors near dome, particularly for mixtures
  - Computational challenges with CFD
- Inlet flow can vary by factor of **five** near dome due to ambient temperature changes (from 0 to 60°C) for a given mass flow
- Potential for condensation/cavitation at compressor inlet

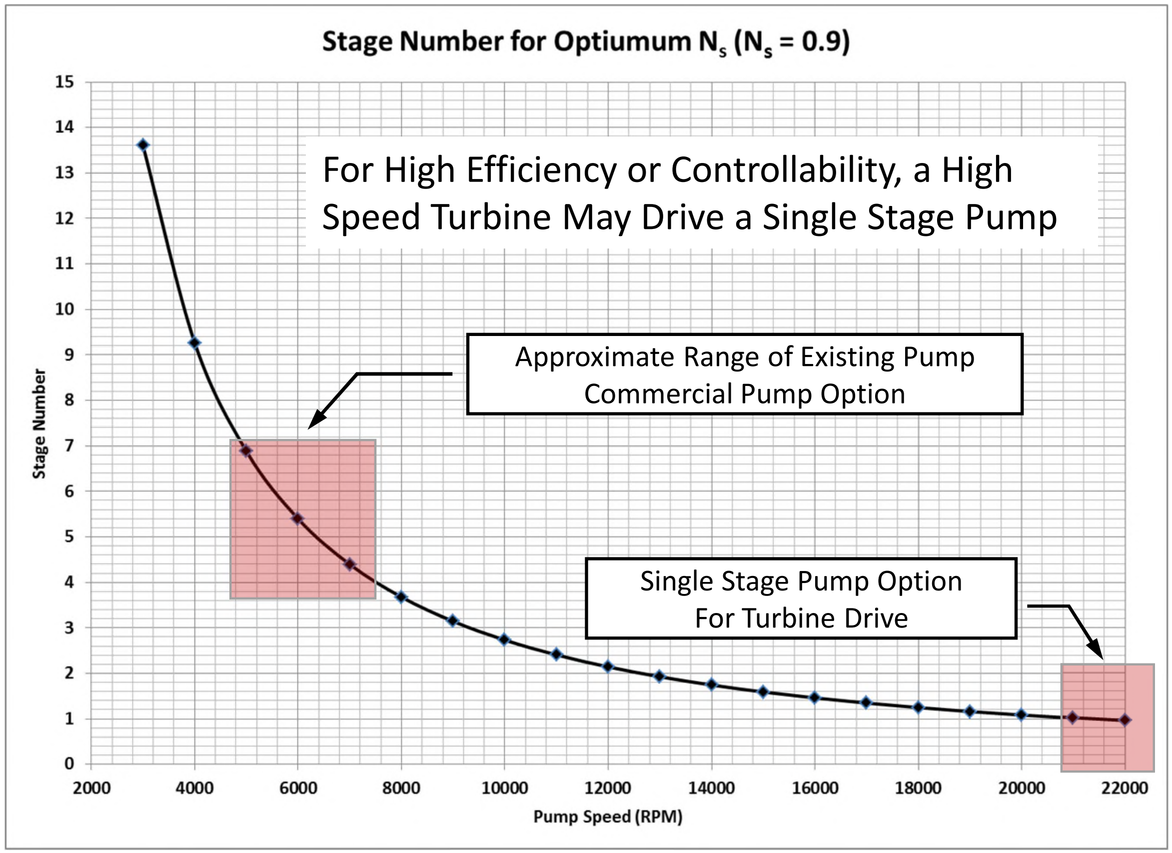
Axial/Radial Designs

- Selection made based on optimal specific speed
- Compressors typically radial, even up to ~1,000 MW scale. Axial compressors possible at higher volume flows. Radial compressor have higher off-design efficiency and turn-down.
- Turbine transition from radial to axial in 10-30 MW range
- Above comments are general, specifics will change based on operating conditions and cycle configuration.

Compressor Design

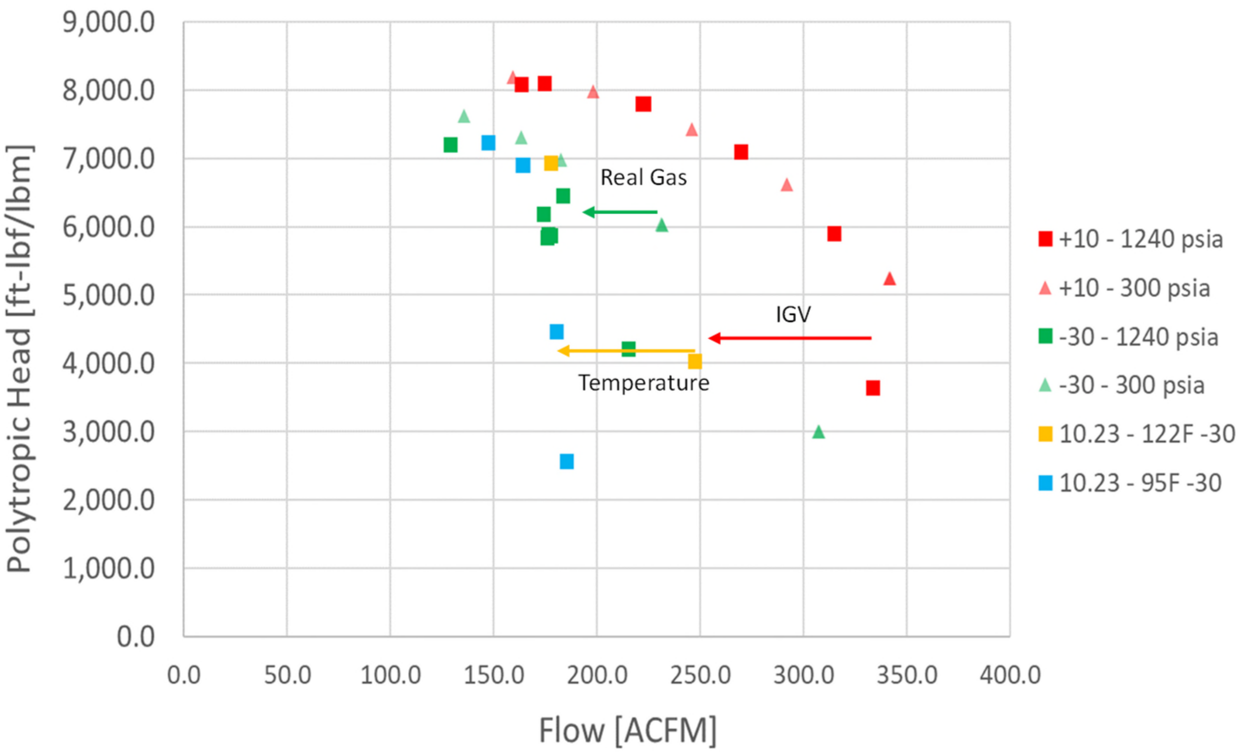
- Centrifugal compressor impellers typically closed
  - Mechanically robust
  - Insensitive to axial motion
  - Low head requirement

Optimum Single Stage Pump Requires N = 22,000 rpm, 2.4MW CO_2 Pump



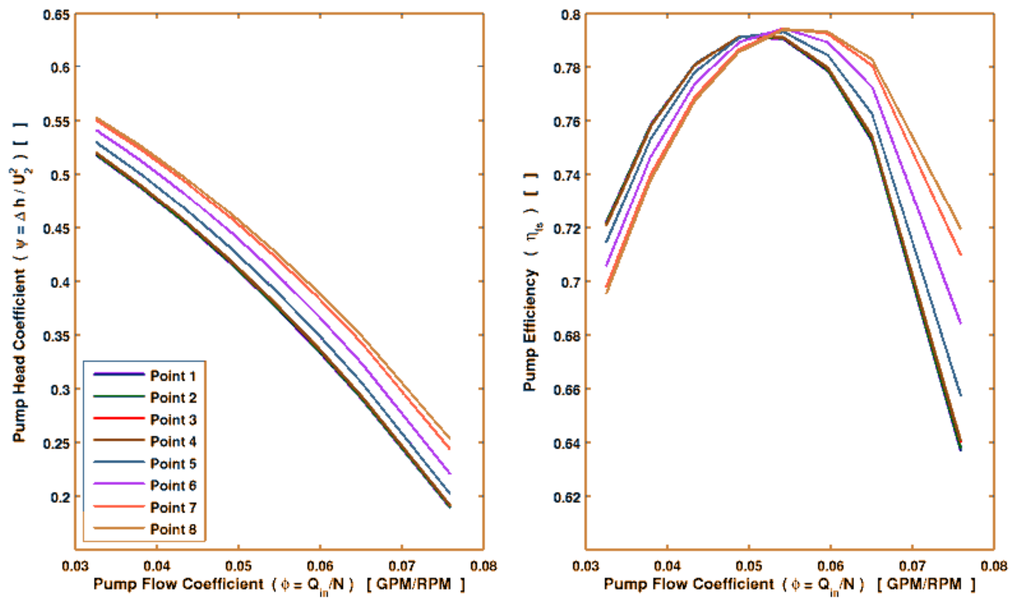
SCO2 Compressor Test Results

- After break-in, testing performed at medium and full pressure
  - 300 and 1240 psi suction pressure
  - +10° and -30° IGV setting angles
  - 95° and 122°F suction temperature
- Both speed of sound and IGV setting has strong effect on the flow capacity of the compressor



Moore, J., Cich, S., Neveu, J., Klaerner, J., Mortzheim, 2022, "**Mechanical and Rotordynamic Test Results of a Supercritical CO2 Compressor Operating Near the Critical Point,"** The 7th International Supercritical CO2 Power Cycles Symposium February 21 – 24, 2022, San Antonio, Texas, Paper # 164

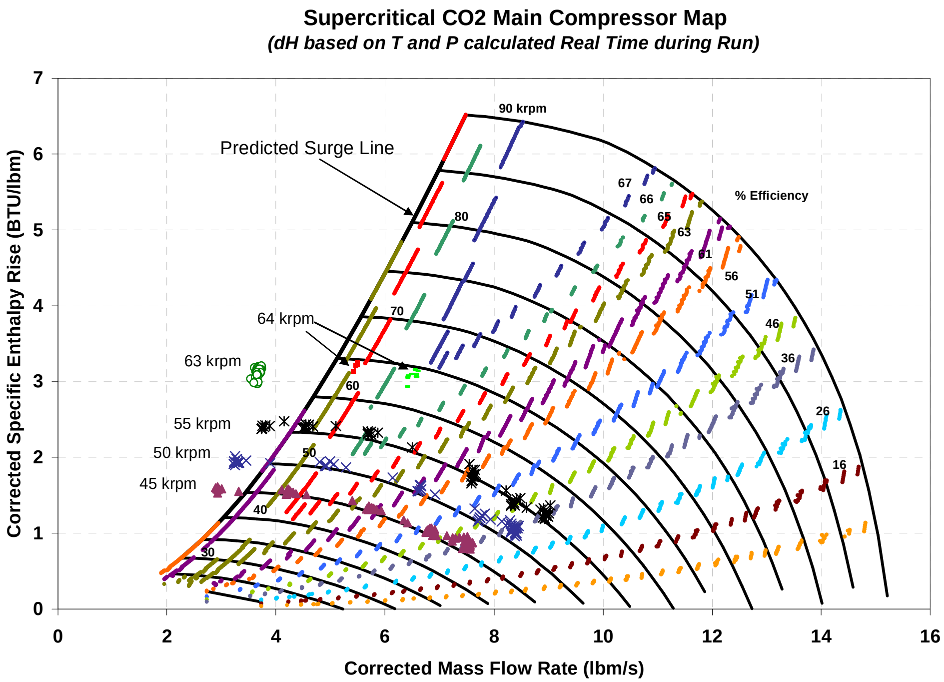
High Speed Pump Head and Flow Coefficients for CO_2
Over Wide Inlet Condition Range
Changes Due to Density Range



Main Compressor Example

Must Work Over a Wide Inlet Density Range (Depending on Control Strategy)

37mm
Wheel
Diameter
30-50 BAR Pressure Rise



Courtesy of Barber-Nichols

Main Compressor Testing Design Point

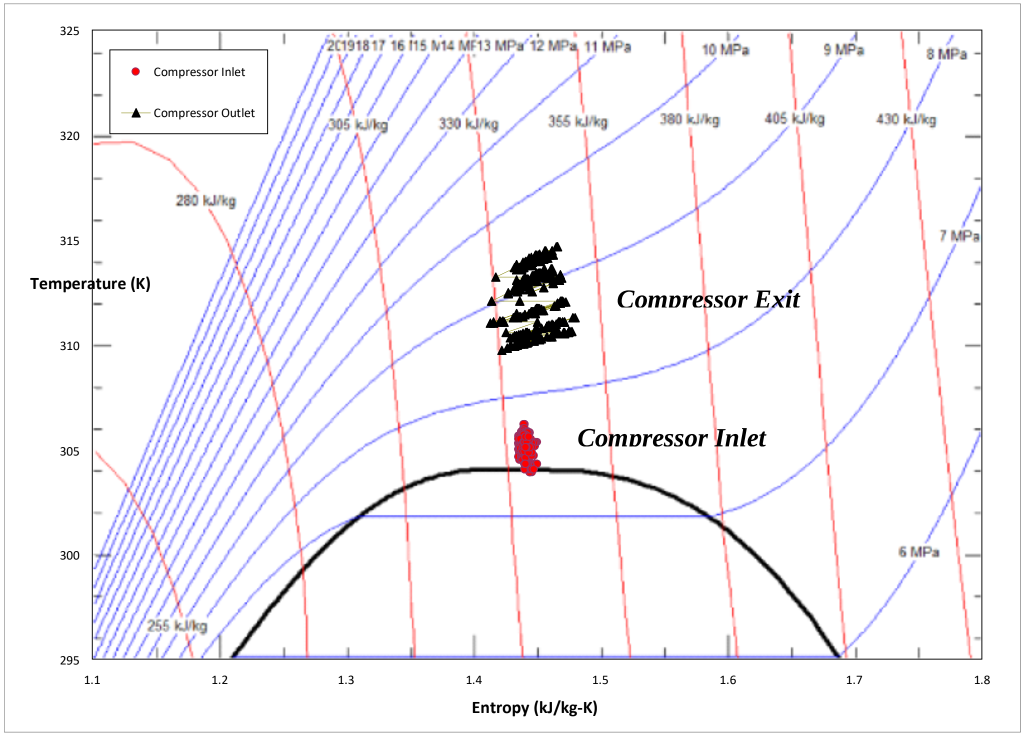
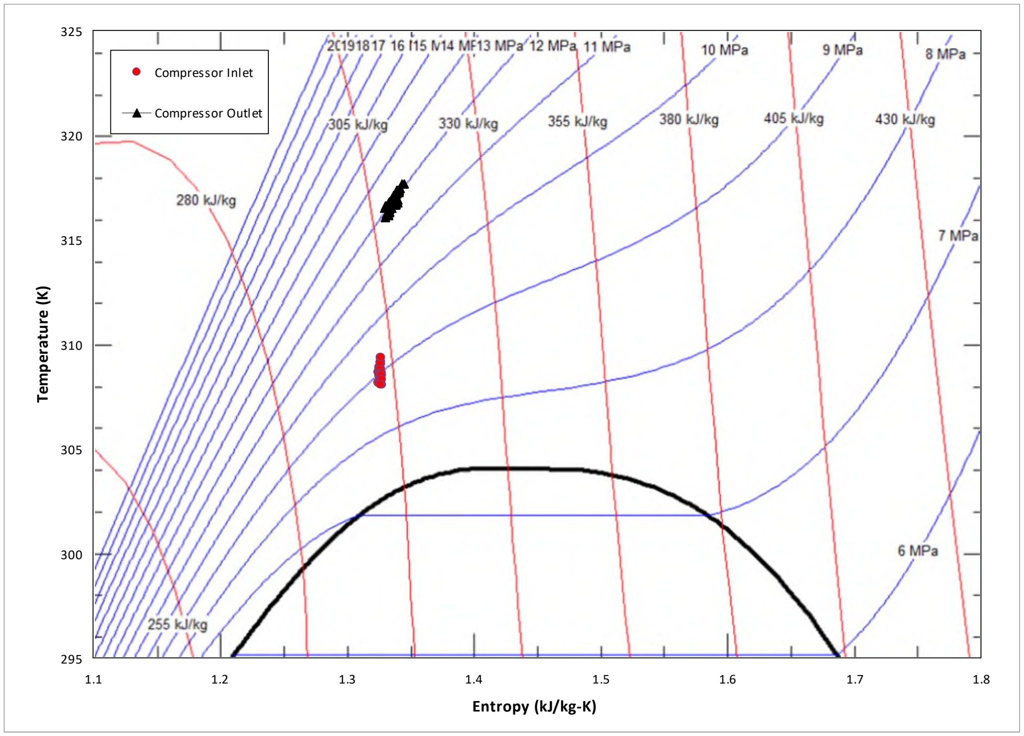


Figure 4a-Design Operating Condition Compressor Inlet and Exit Points

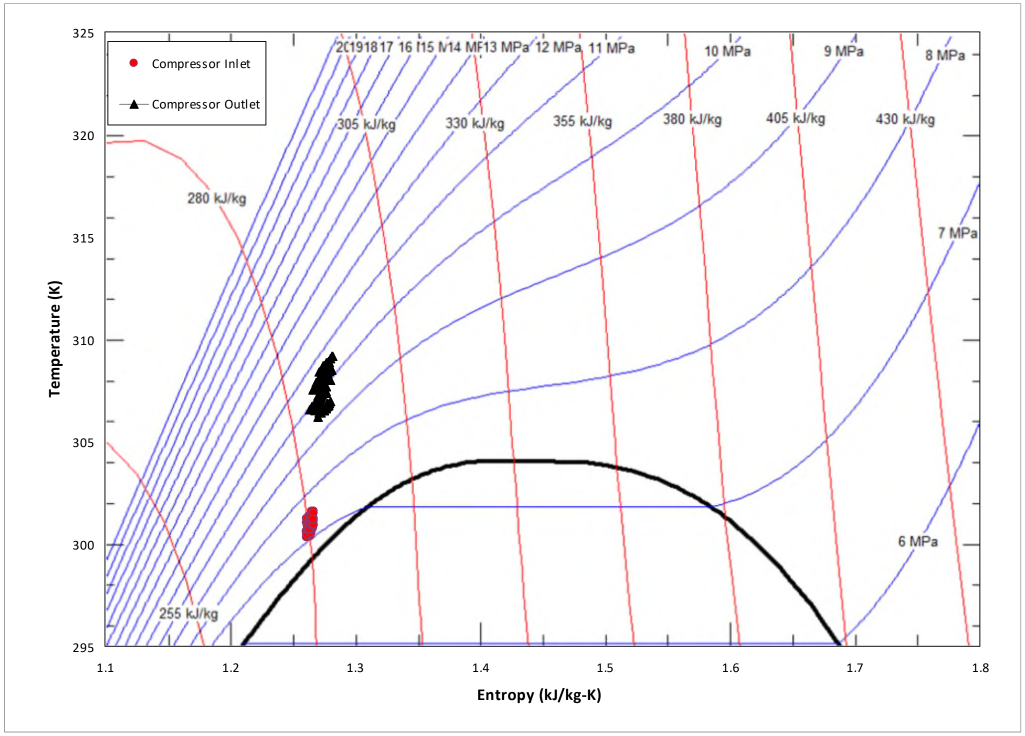
Courtesy of Barber-Nichols

Main Compressor Test
High Pressure, Raised Inlet Temperature



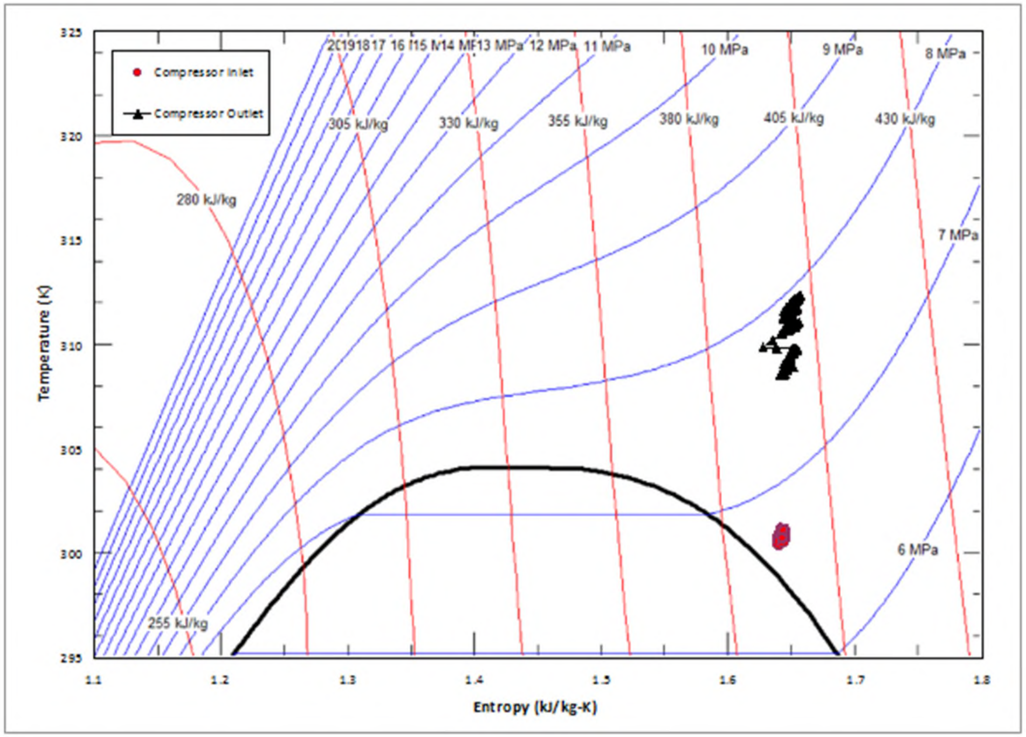
Courtesy of Barber-Nichols

Main Compressor Testing
High Density Inlet



Courtesy of Barber-Nichols

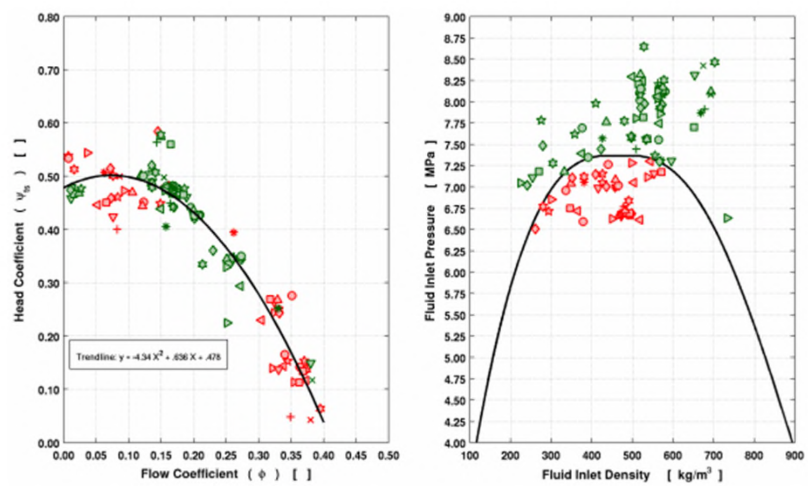
Main Compressor Testing
Low Density Inlet (Startup)



Courtesy of Barber-Nichols

Suction Conditions Near the Dome

- Liquid droplet erosion may occur
- sCO_2 compressor tested at inlet pressures both above and below the saturation line (Noall and Pasch, 2014) without significant notable problems
- Nucleation timing also a factor



Seals: Internal and Shaft End

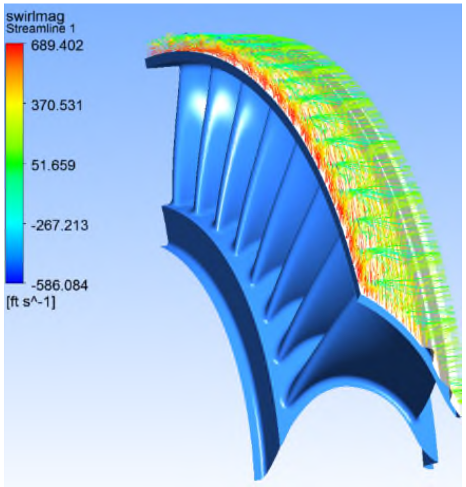
Seals

Labyrinth:

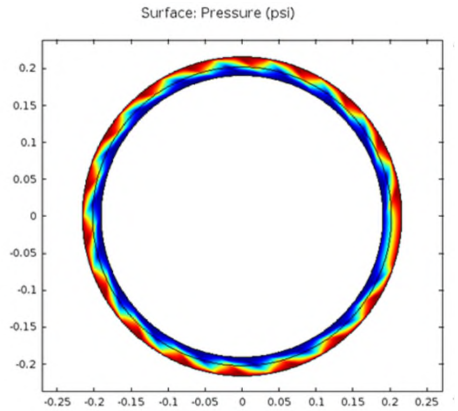
- Labyrinth seals used at blade tip and interstage locations
- Swirl brakes used to minimize swirl entering seal
    - CFD used to optimize and evaluate swirl brake performance

Dry Gas Seals

- Commercially available at the required pressure but limited to low temperature and smaller diameter.
- Requires clean, dry, filtered CO_2 for seal buffer gas
    - Superheat required to prevent liquid and dry ice formation during expansion across face

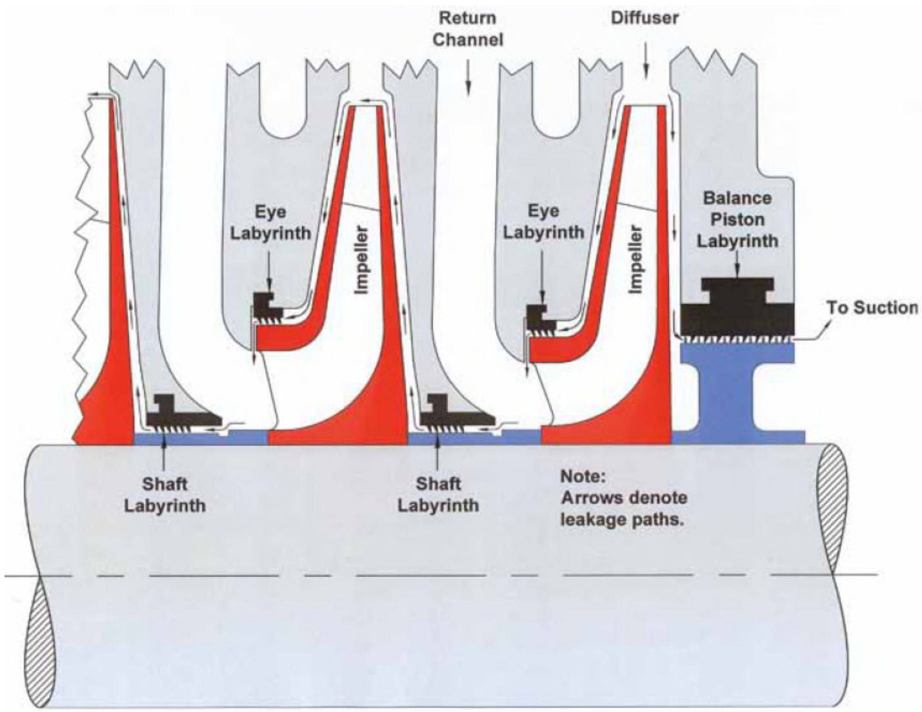


CFD analysis of interstage laby seal flow in CO_2



DGS Face Pressure Distribution from CFD

Annular Gas Seals in Compressor



Swirl Brake Example

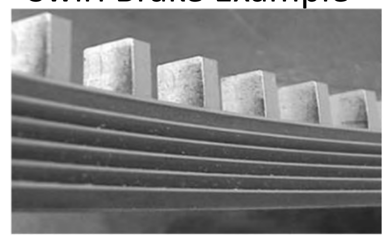
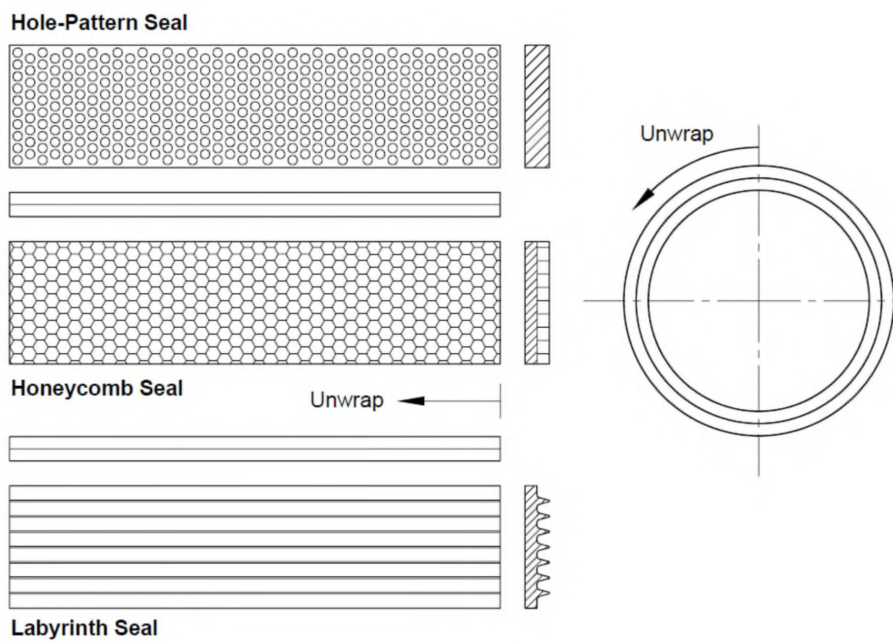


Image source [7-1]

Different Seal Geometries



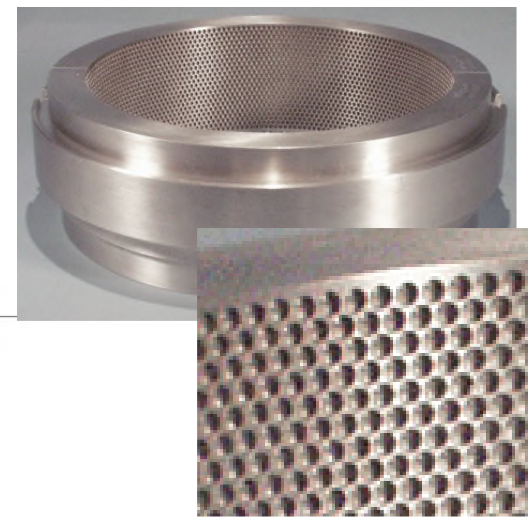
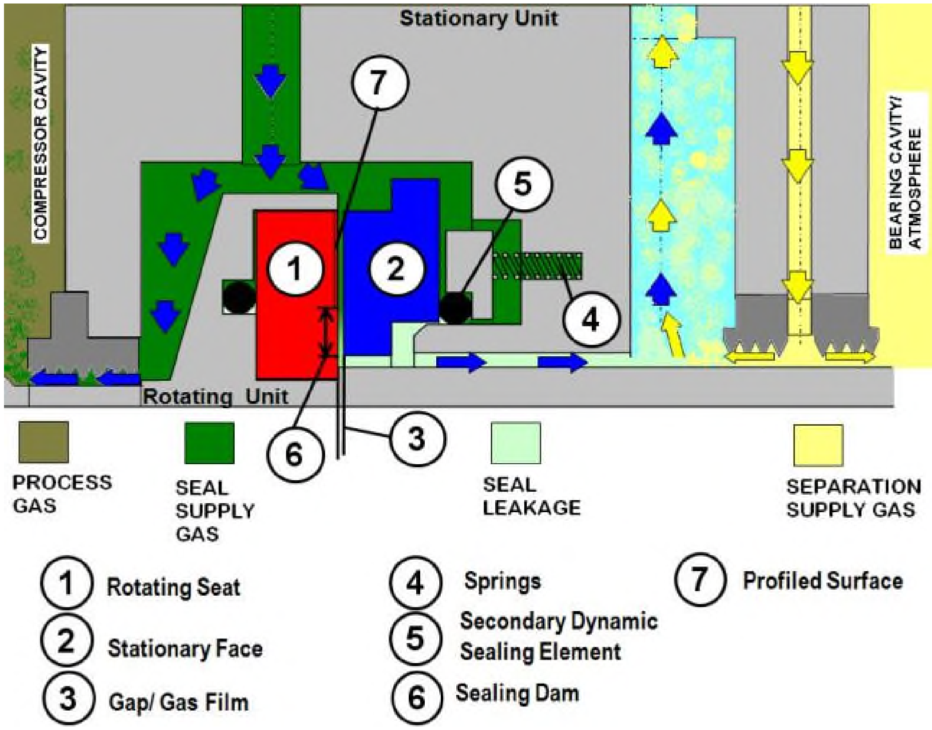


Image source [7-2]

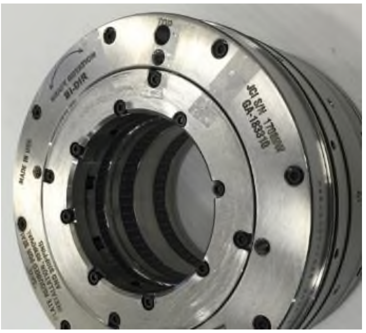
Dry Gas Seals



SCO2 requires clean, dry, warm gas to be supplied to seal
For expanders, this seal gas provides necessary cooling for the seal
Must supply gas during pressurized holds
Many seal failures have been experienced in SCO2

Rotating seal surface…

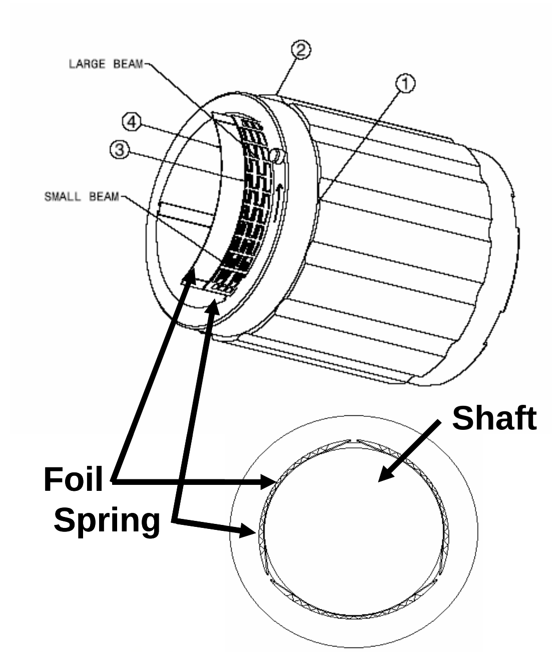
Image source [7-4]

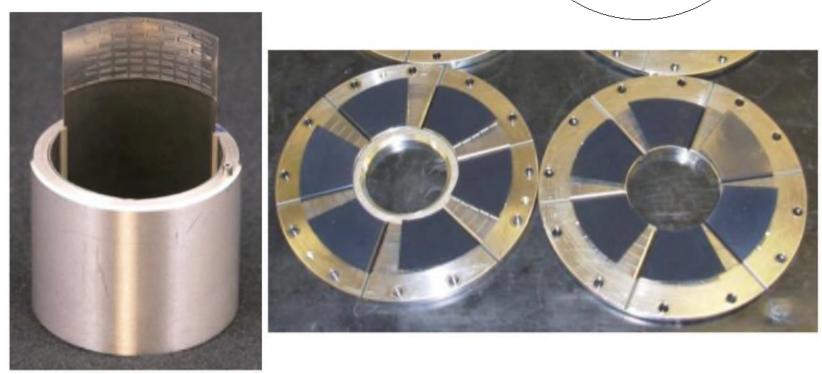


Bearings

Gas Foil Bearings

- Thrust or radial bearing
- Working fluid as lubricant
  - Do not require separate lube system, seals, etc.
- Lower viscosity than typical oil lube
  - Lower load capacity
  - Less damping
- Limited to smaller machinery





Gas Foil Journal (left) and Thrust (right) Bearings (Wright *et al.*, 2010)

Hydrodynamic Oil-Lubricated Bearings

- Thrust or radial bearing
- Oil-lubrication must be separated from dry gas seals
- Good load capacity
  - Used with larger machinery

❑ Types

- Fixed geometry (low performance)
- Tilting pad (high performance)

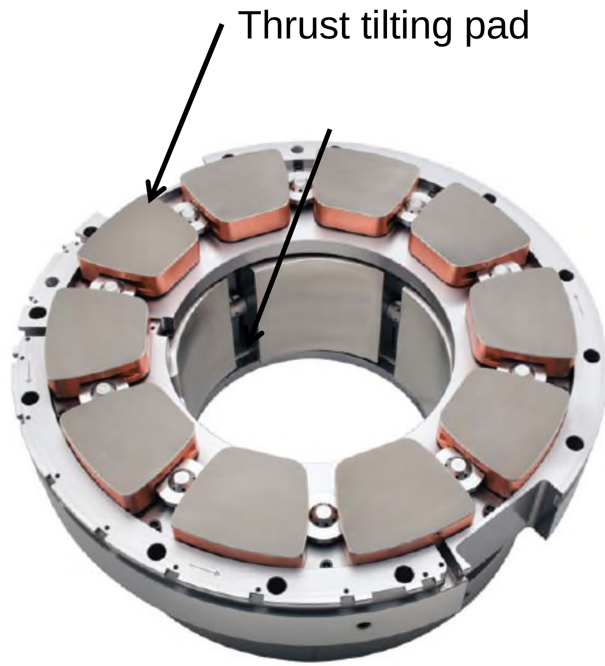
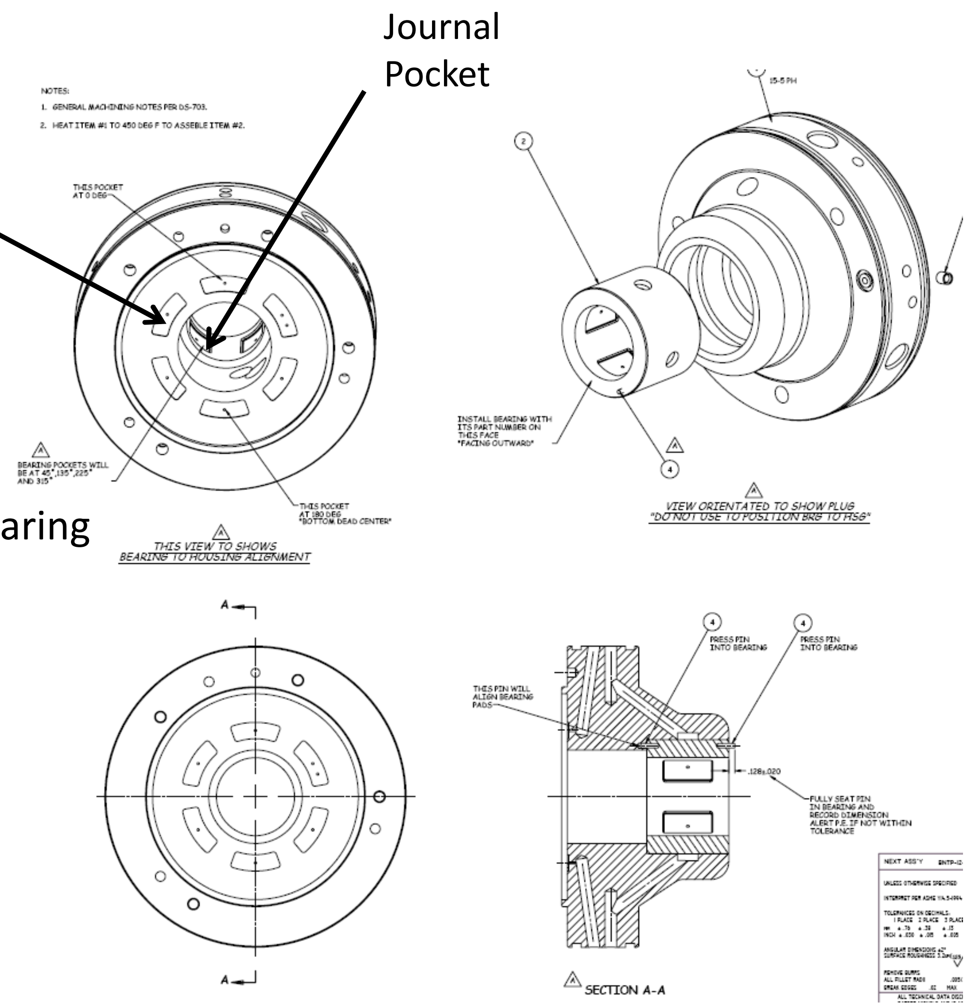


Image source [7-6]

Radial tilting pad

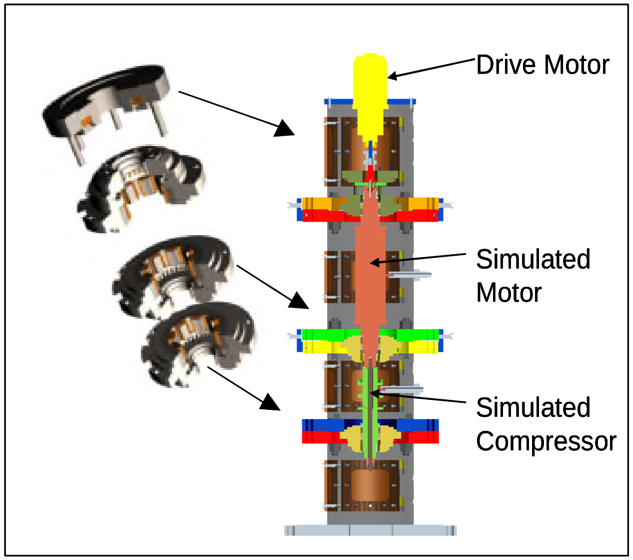
CO_2 Hydrostatic Bearings



- Combined Thrust/Radial Bearing
- High Damping
- Good load capacity
- Hermetic Machine
- Moderate CO_2 Flow Rates
- Self Supplied By Pump

Magnetic Bearings

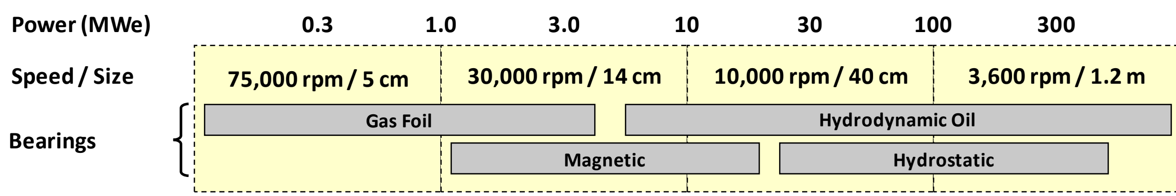
- Provides levitation in radial and axial direction
- Requires active feedback control, position sensors, power amplifiers and electromagnetic actuators
- Can operate in working fluid eliminating shaft end seals
- Thermal management of heat generated from windage and resistive heating required
- Auxiliary Bearings required when de-levitated and as a back-up if levitation is lost
  - Pre-loaded pairs of angular contact bearings
  - Mechanical damper between roller bearings and housing
- Rotor can rotate about mass center minimizing dynamic loads transmitted to casing (auto balancing)
- Expensive but cost can be offset by eliminating lube oil and shaft end seals
- Research underway to develop 500C+ bearings



Ransom, D.L., Masala, A., Moore, J.J., Vannini, G., Camatti, M., 2009, **Development of a Vertical High Speed Motor-Compressor Simulator for Rotor Drop onto Auxiliary Bearings,** Presented at the 38th Turbomachinery Symposium, September 2009, Houston, TX.

Bearing Type Summary

	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 & Excursions	Always	At low speed	Never
Cost	Low	Low	Medium	High
Drag torque	Low	Medium	Low-medium	Very low



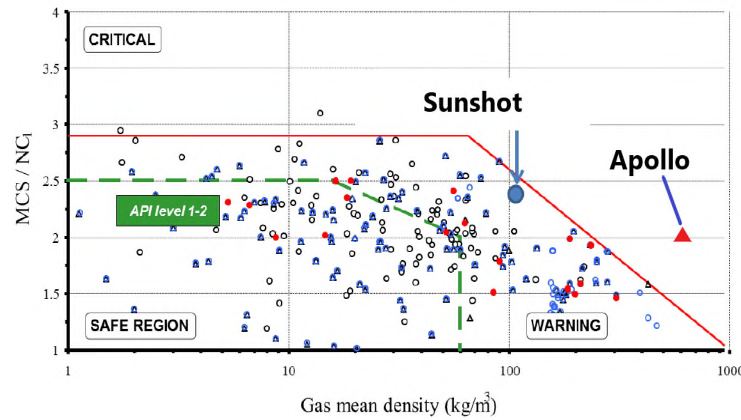
Bearing type as a function of rotor power [adapted from (Seinicki *et al.*, 2011)]

Rotordynamics

Rotordynamics

Challenges

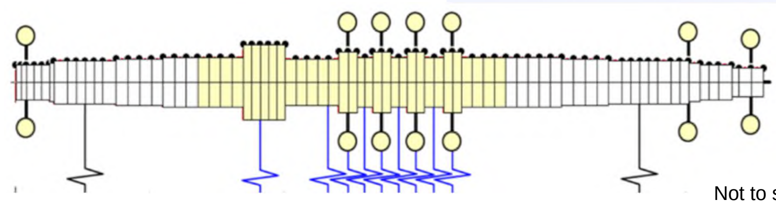
- High gas density
- High operating speed
- Low critical speed (large L/D)
- Similar design methodology as high pressure turbocompressors



Interstage laby seals

- Texas A&M XLTRC code
- Real gas CO_2 properties

$$CSR = \frac{Operating\ Speed}{1st\ Undamped\ Critical\ Speed}$$



Not to scale

Balance piston seal

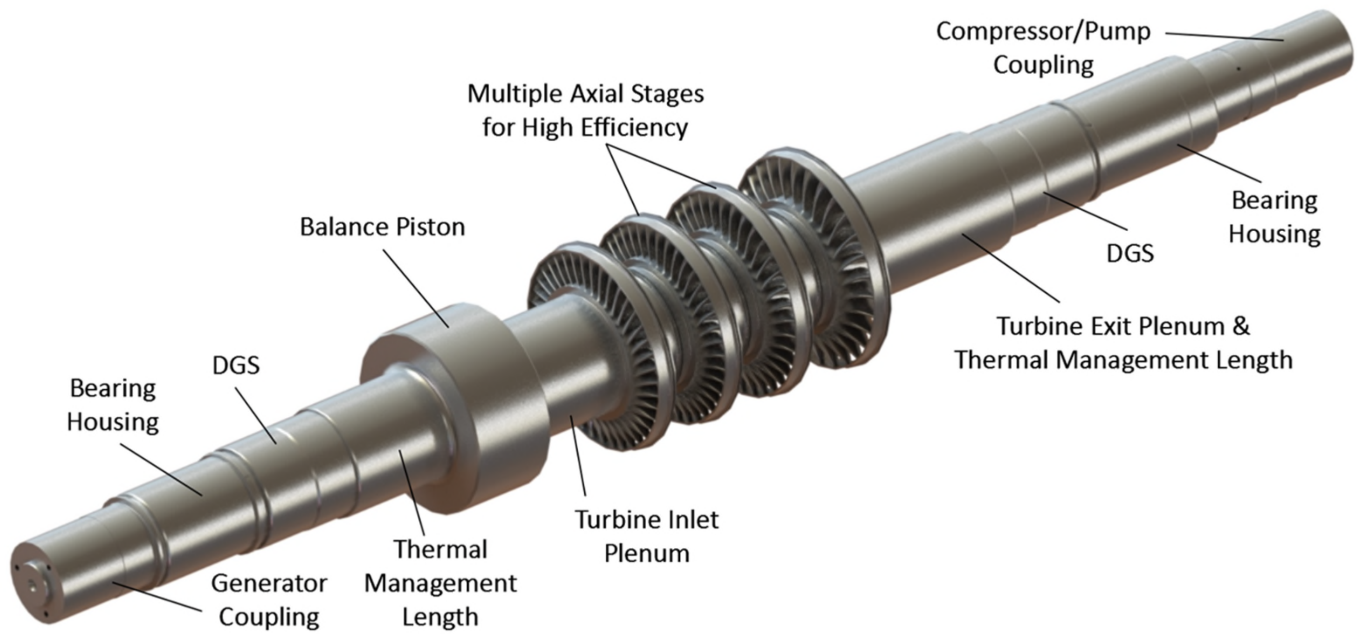
- Texas A&M code
- Perfect gas properties

Result

- Due to uncertainty in seal damping, we used a factor of safety 10x API level II minimum (final logdec > 1.0)

sCO_2 Turbine Rotor Features

- Typical rotor components

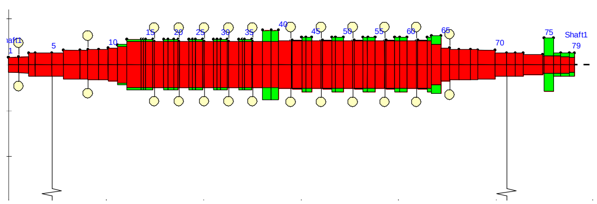


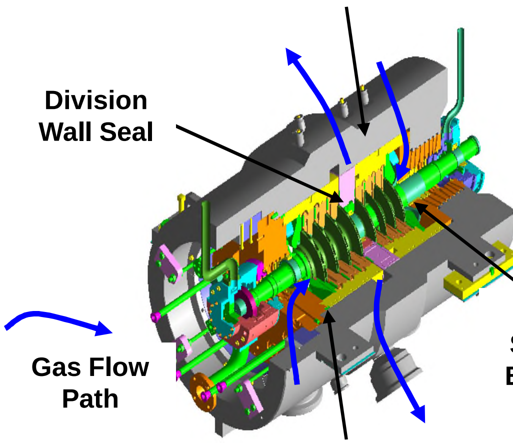
Rotordynamic Modeling

Rotordynamic Modeling

- Similar to other rotors
- Break the series of smaller segments at diameter steps
- Components like impellers, couplings, thrust disks do not add shaft stiffness are modeled as added mass
- Stations added at bearings centerlines

Sample 10-Stage Compressor Model





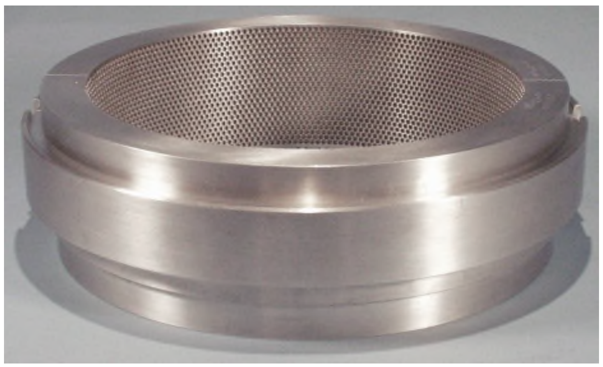
Typical High Pressure Centrifugal Compressor

Reference: Moore, J.J., Soulas, T.S., 2003, "Damper Seal Comparison in a High-Pressure Re-Injection Centrifugal Compressor During Full-Load, Full-Pressure Factory Testing Using Direct Rotordynamic Stability Measurement," Proceedings of the DETC '03 ASME 2003 Design Engineering Technical Conference, Chicago, IL, Sept. 2-6, 2003

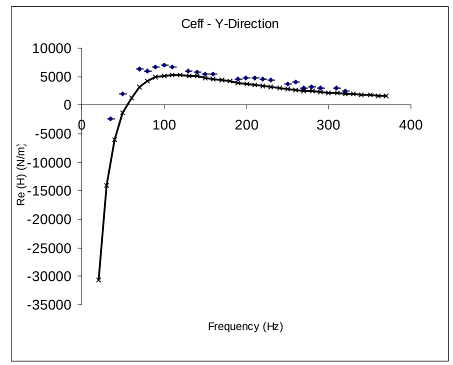
Rotordynamic Modeling

Damper Seal Damping Test Data vs. Predictions

- Damper seals like honeycomb seals provide substantial damping
- Damping increases with increasing pressure differential

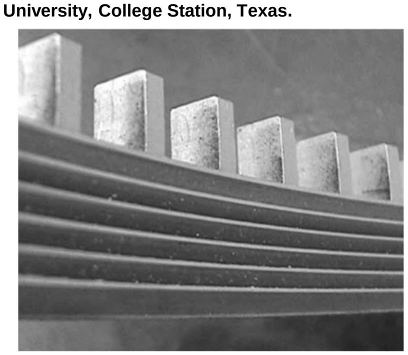


http://www.dresser-rand.com/insight/v9no1/art_6.asp



Reference: Camatti, M., Vannini, G., Fulton, J.W., Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," *Proc. of the Thirty-Second Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas.

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



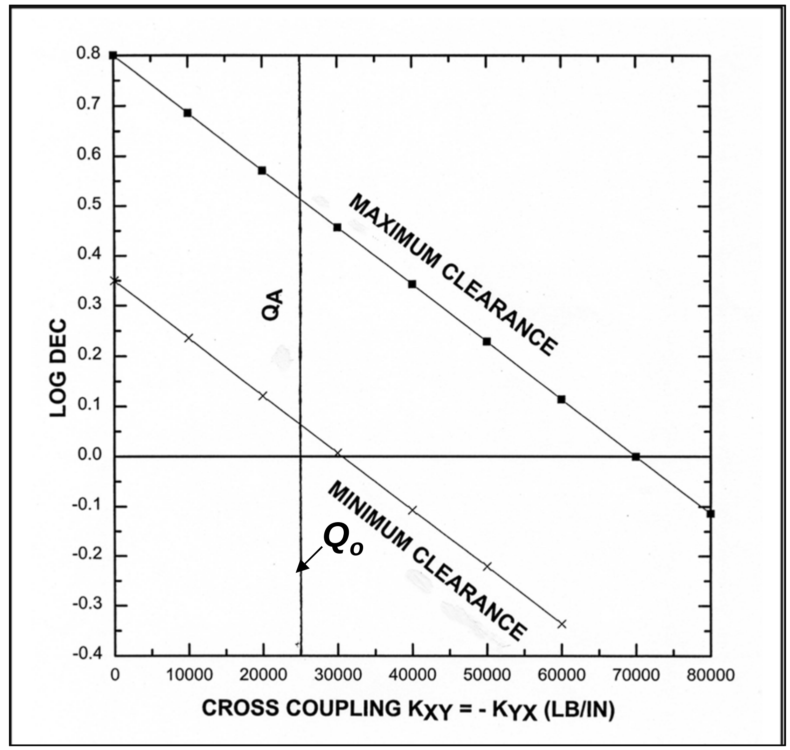
API 617 Requirements

- Stability Plot
  - Plots log dec vs. applied K_{XY}
  - Ratio of zero crossing (Q_o) to Q_A defines stability margin

$$SM = \frac{Q_o}{Q_A}$$

$$Q_A = \frac{189,000 \times HP}{D_c \times H_c \times N} \frac{\rho_d}{\rho_s}$$

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



Physics Based Impeller Kxy Prediction

Alternative to API Q_A prediction

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

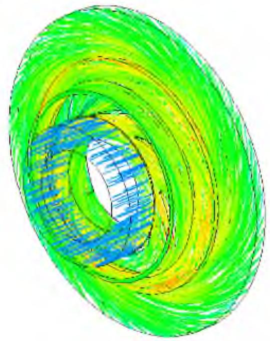
Refined equation based on CFD for Centrifugal Compressors

C_{mr} = CFD-determined coefficient

U = wheel tip speed,

L_{shr} = axial length of the shroud

Q/Q_{design} = is the ratio of flow relative to design flow



▪Moore, J.J. and Ransom, D. "**Centrifugal Compressor Stability Prediction Using a New Physics Based Approach**," Proceedings of ASME Turbo Expo 2009: Power for Land, Sea and Air, June 8-12, 2009 in Orlando, Florida.

API 617 Requirements Applied

- Severity of the Application defined by location on "Fulton" chart
- CSR = Critical speed ratio which is the ratio of running speed and first critical speed
- Horizontal axis is average gas density
  - Average of suction and discharge density
- The greater the CSR and density, the more severe the application
  - Region A – Less severe
  - Region B – More severe

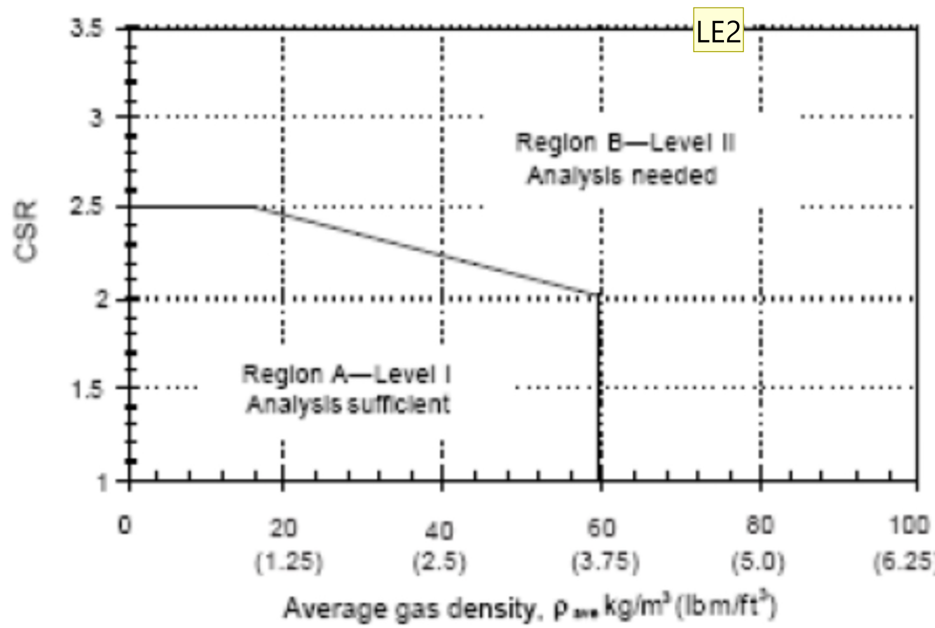


Figure 1.2-5—Level I Screening Criteria

Slide 84

LE2 Do we have a clearer figure we could use for this slide?
Linda Estes, 8/25/2017

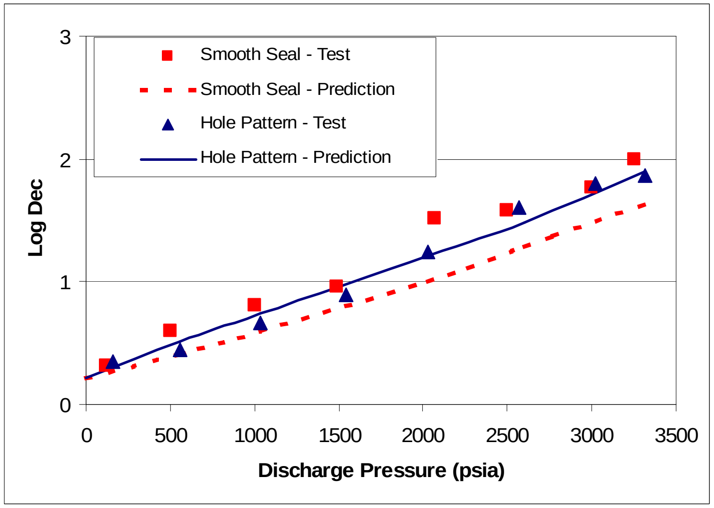
API 617 Requirements

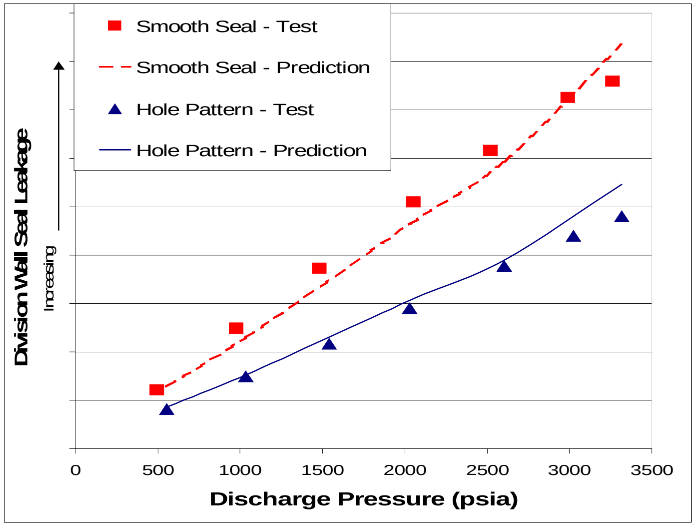
- If any of the following is not met, then a Level 2 analysis is required
  - $SM < 2.0$
  - $\delta_A < 0.1$
  - $2.0 < SM < 10$ if in Region B
- Level 2 Analysis includes the effect of:
  - All labyrinth/damper seals
  - Balance piston seals
  - Impeller/blade row (some believe that only labyrinths are important)
  - Shrink fits
  - Shaft material hysteresis
- Resulting log dec must be greater than 0.1
- Meeting API requirements does not guarantee a stable rotor
- Author's suggested requirements using Level 2 analysis:
  - $\delta_A > 0.3$
  - $SM > 3.0$

Rotordynamic Model Validation

Measured Log Decrement in Centrifugal Compressor

- Similar operating pressure
- Shows damper seal effectiveness
- Log Dec increases as discharge pressure increases
- A smooth seal was tested to simulate a “plugged-up” seal

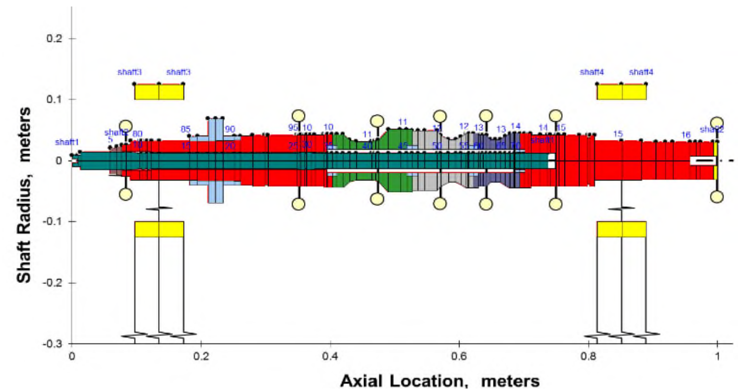


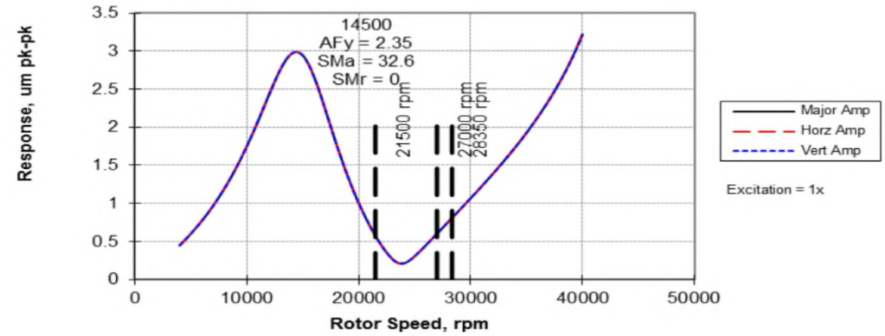


Reference: Moore, J.J., Soulas, T.S., 2003, “Damper Seal Comparison in a High-Pressure Re-Injection Centrifugal Compressor During Full-Load, Full-Pressure Factory Testing Using Direct Rotordynamic Stability Measurement,” Proceedings of the DETC ’03 ASME 2003 Design Engineering Technical Conference, Chicago, IL, Sept. 2-6, 2003.

SCO2 Compressor Rotordynamic Analysis

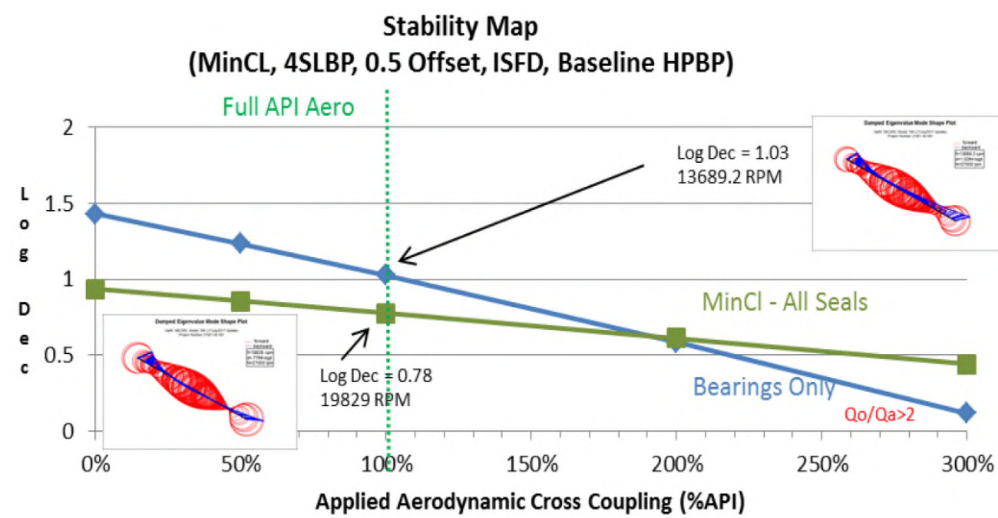
- Final rotor configuration used inboard thrust bearing to provide better control of the bending mode
- Critical speed is well damped with good separation margin from the running speed
- Bending critical well above running speed

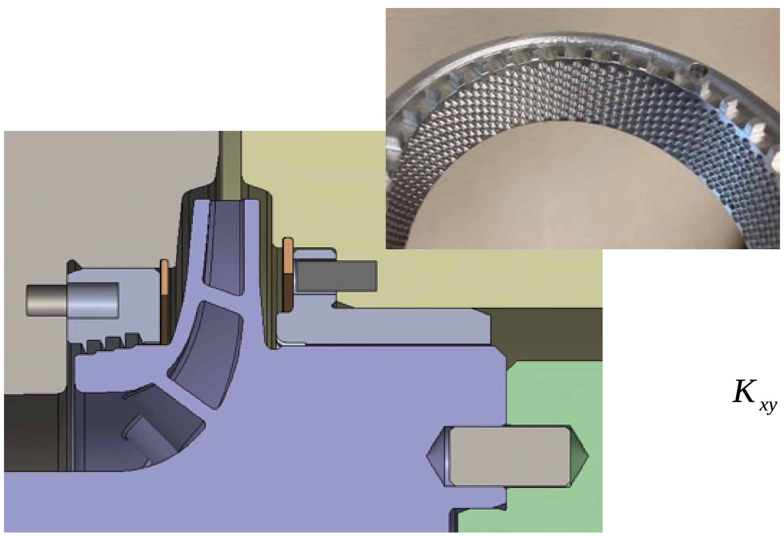




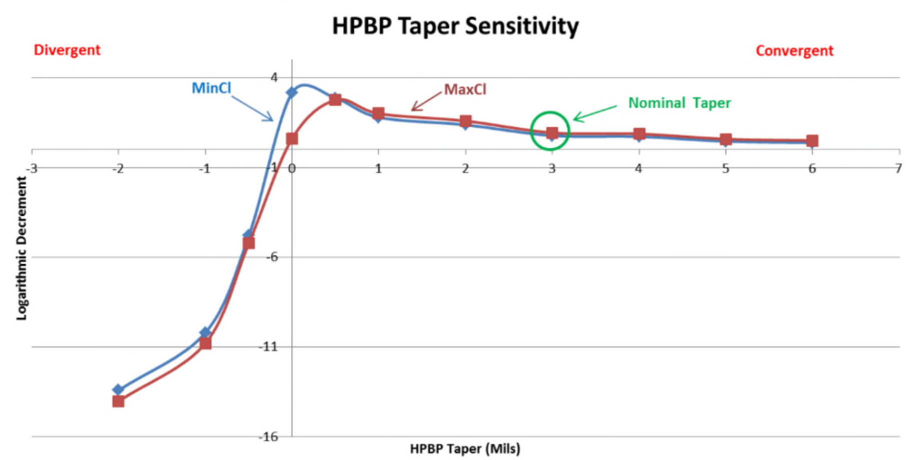
Rotordynamic Analysis

- Rotodynamic forces amplified by high density fluid
- Swirl brakes and damper seals
- Damper seal carrier design to minimize coning of the seal clearance



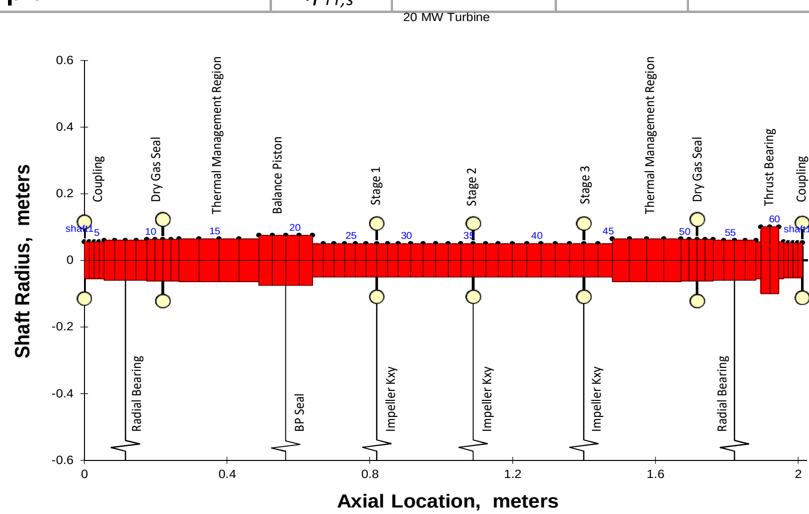


$$K_{xy} = \frac{C_{mr}\ \rho_{dis}\ U^2\ L_{shr}}{Q / Q_{design}}$$



Turbine Design Details for 20 MWe Case Study

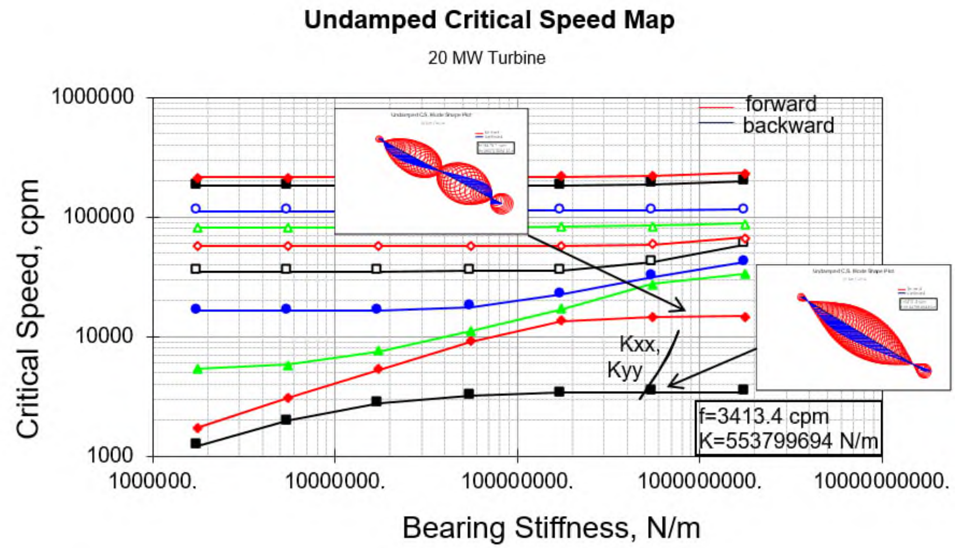
Parameter	Symbol	Units	Stage 1	Stage 2	Stage 3
Inlet Mass Flow Rate	m	kg/s	210	210	210
Inlet Total Pressure	P_{00}	kPa	23720	17370.0	12548.8
Inlet Total Temperature	T_{00}	K	973.15	932.32	890.62
Discharge Total Pressure	P_{08}	kPa	17370	12549	8960
Specific Speed	Ns	-	0.395	0.453	0.526
Pinion Speed	N	rpm	10800	10800	10800
Exit Flow Coeff. (Cm6/U4)	φ	-	0.263	0.286	0.313
U/C	U_4/C_s	-	0.695	0.701	0.709
Stage Pressure Ratio	PR	-	1.366	1.384	1.401
Isentropic Eff.	$\eta_{TT,s}$	-	84.22%	85.85%	86.04%



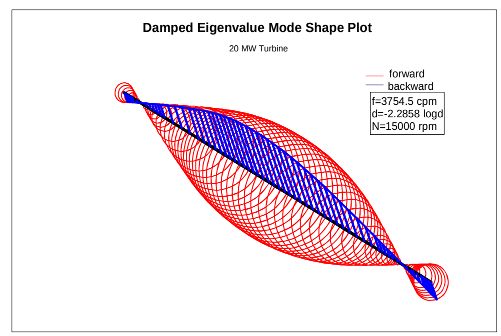
Rotordynamic Results for 20 MWe Case Study

- Rotordynamically challenging due to long slender rotor
- Rotor is unstable without stability enhancing features like damper seals and swirl brakes.

	K_{XY} 1e6 [N/m]
Stage 1	4.02
Stage 2	3.38
Stage 3	2.77

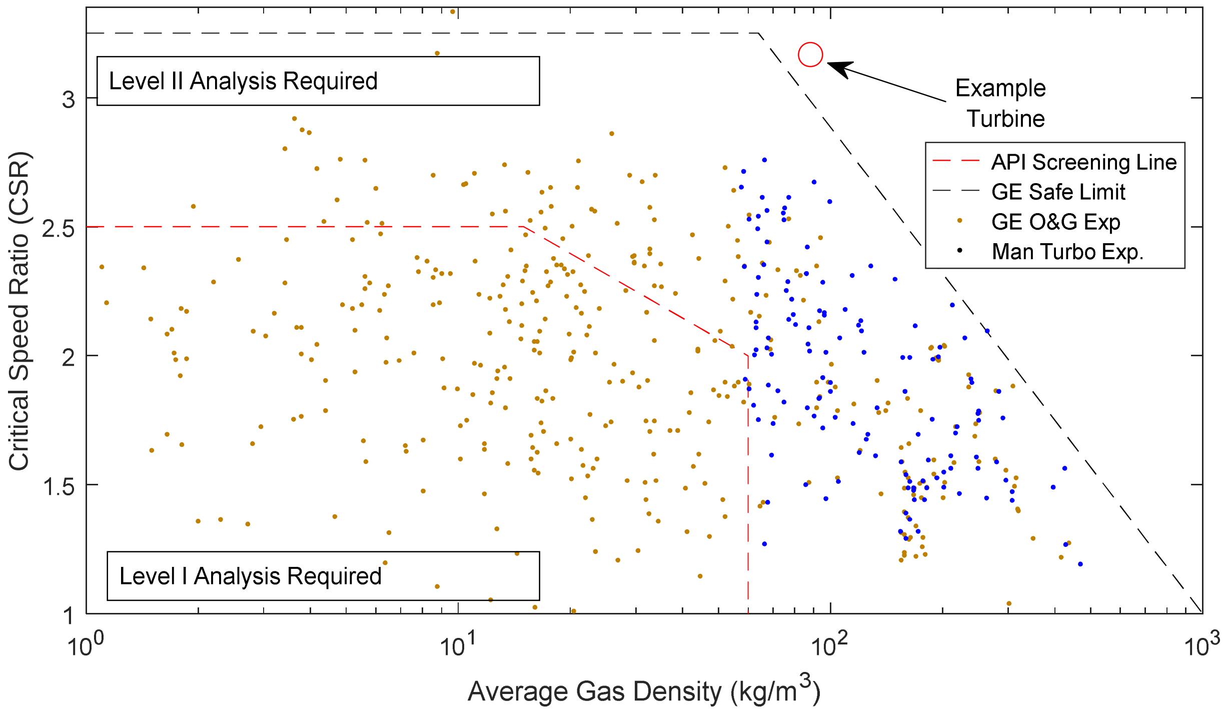


Log Dec = -2.2 without damping devices



Fulton Experience Chart

Example Turbine = 20 MW sample turbine



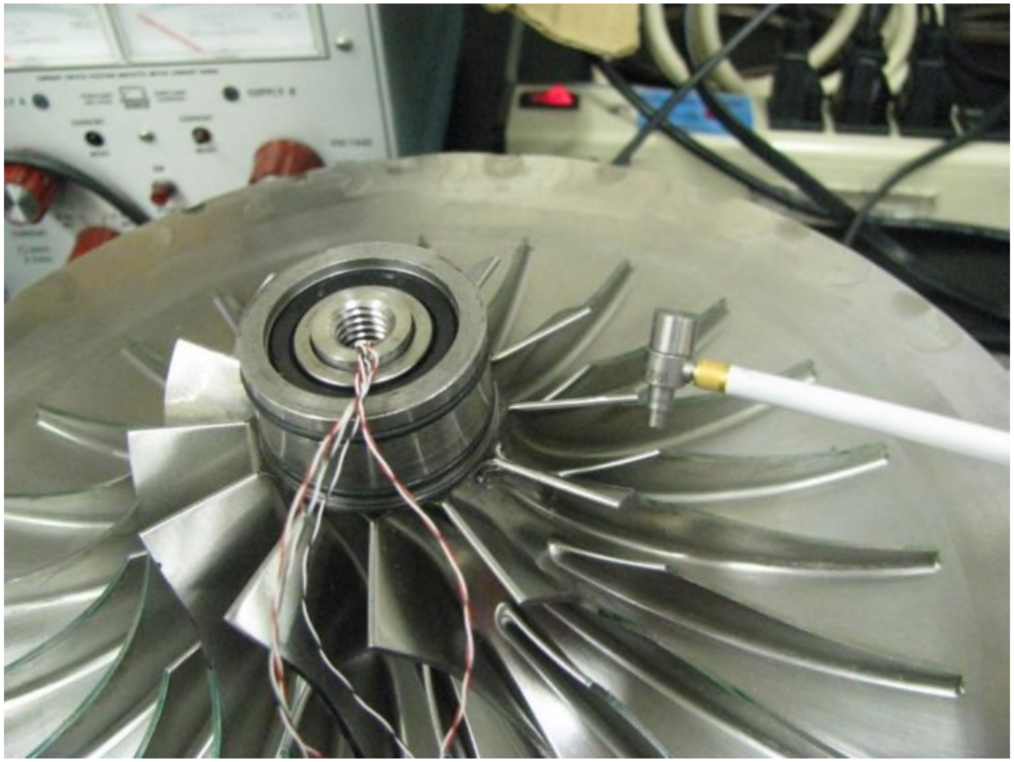
Blade Dynamics

Blade Loading and Dynamics

- High gas density and machine power density results in large blade loading
  - Gas forces need to be considered in addition to centrifugal loads
  - Blade-to-disk attachment requires special consideration
- High gas density also amplifies unsteady wake interaction forces on blades
  - Critical to avoid resonance
  - Non-harmonic excitation from gas separation should be avoided

Modal Test Validation

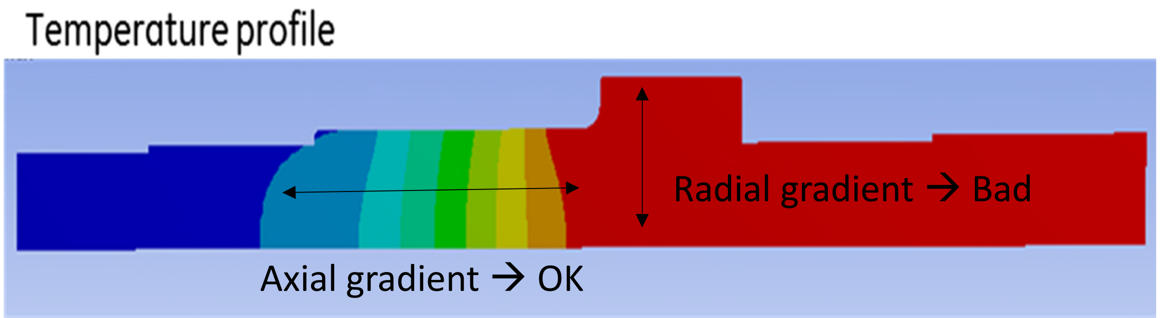
- Modal testing used to validate design
- Effect of gas density and temperature dependent material properties must be considered for accurate natural frequency prediction



Thermal Management

Thermal Management

- Temperature between hot inlet (up to 700°C) and dry gas seal (~100°C) requires smooth temperature gradient to avoid excessive thermal stresses
  - In both casing and shaft
  - Radial temperature gradients should be avoided
  - Heat sink provided by seal buffer gas



Kalra, C., Hofer, D., Sevincer, E., Moore, J., Brun, K., 2014, "Development of High Efficiency Hot Gas Turbo-Expander for Optimized CSP Supercritical CO_2 Power Block Operation," 4th International Symposium – Supercritical CO_2 Power Cycles, Sept. 9-10, 2014, Pittsburgh, PA

- Large thermal gradient coupled to pressure containment including transients is challenging
- May result in life limited designs due to LCF and creep

sCO_2 Turbomachinery Mechanical Design

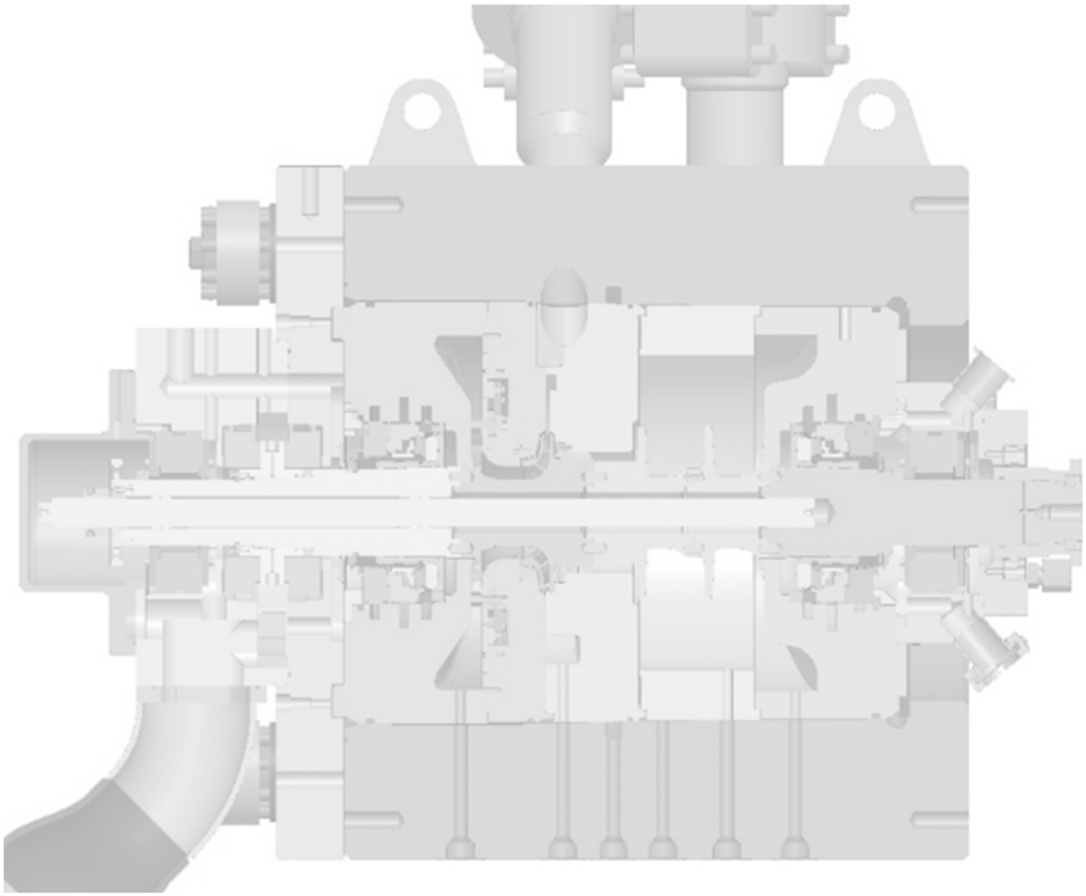
sCO_2 Turbomachinery Mechanical Design Challenges

- High molecular weight, high density results in compact machinery
  - Density near the dome and at high pressure approach that of water
- Amplifies rotordynamic forces on rotor
- Excitation forces acting on blades amplified
- High torque transmission from shaft ends
  - Requires larger seals, bearings, and couplings
- High stage delta-P load impellers and diaphragms
- Explosive decompression in elastomers
- Water in CO_2 is corrosive
- Relatively low peripheral speeds

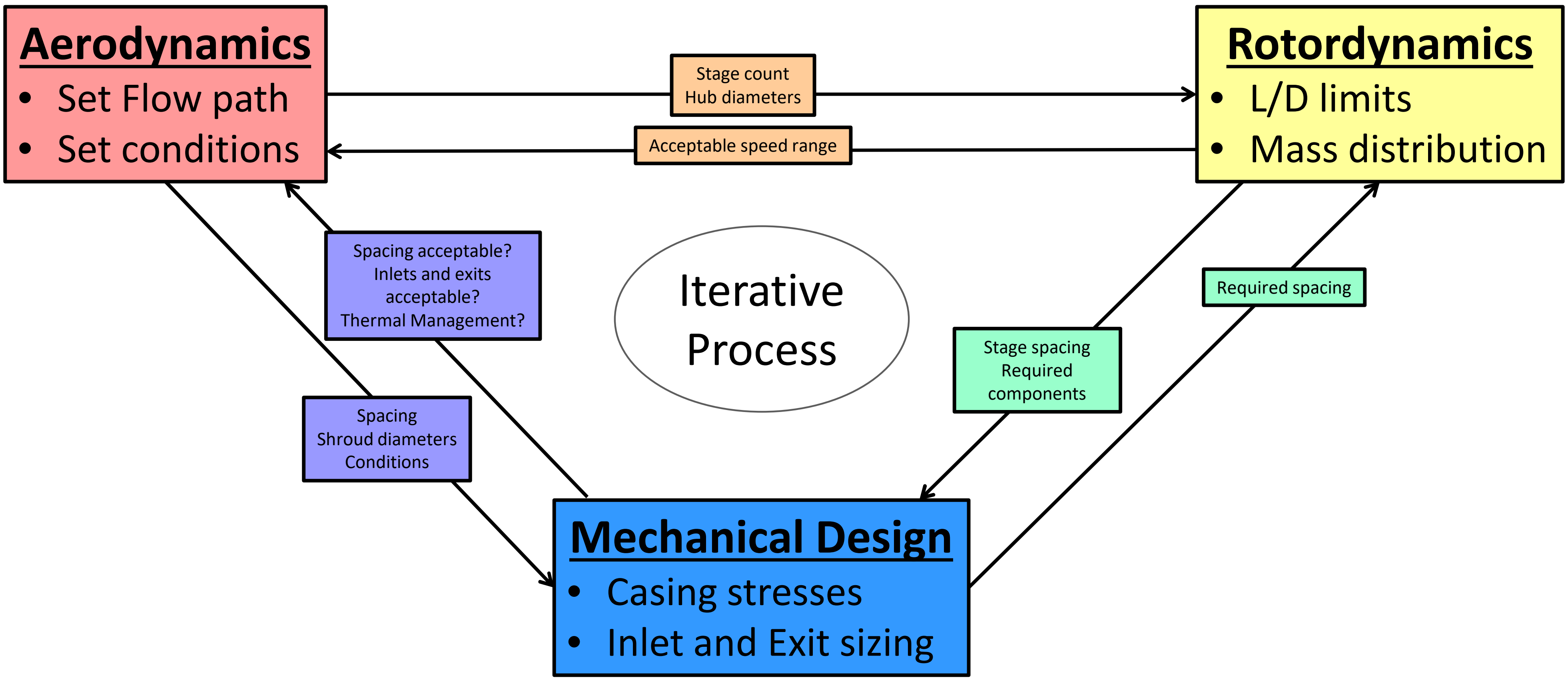
- High blade loading
- High thrust loads
- High shaft speeds
- High critical speed ratios
- Challenging stability
- Flow unsteadiness can cause significant vibration (flow separation, rotating stall, surge, etc.)
- Wide flow range requirement for compressor
- High exhaust pressure requires mechanical face seals
- Pressure containment requires temperature gradients

Example SCO2 Compressor

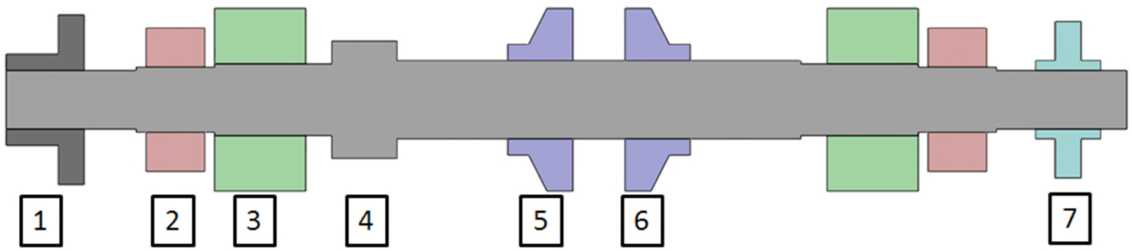
- Design a sCO_2 compressor for use in a closed loop Recompression Brayton Cycle (RCBC) to achieve thermal efficiencies > 50% with both MAIN and BYPASS compressor wheels in the same body
- Design considerations:
  - High-pressure
  - High-density
  - High-speed
  - Large swings in volume flow
  - Packaging



Example SCO2 Compressor



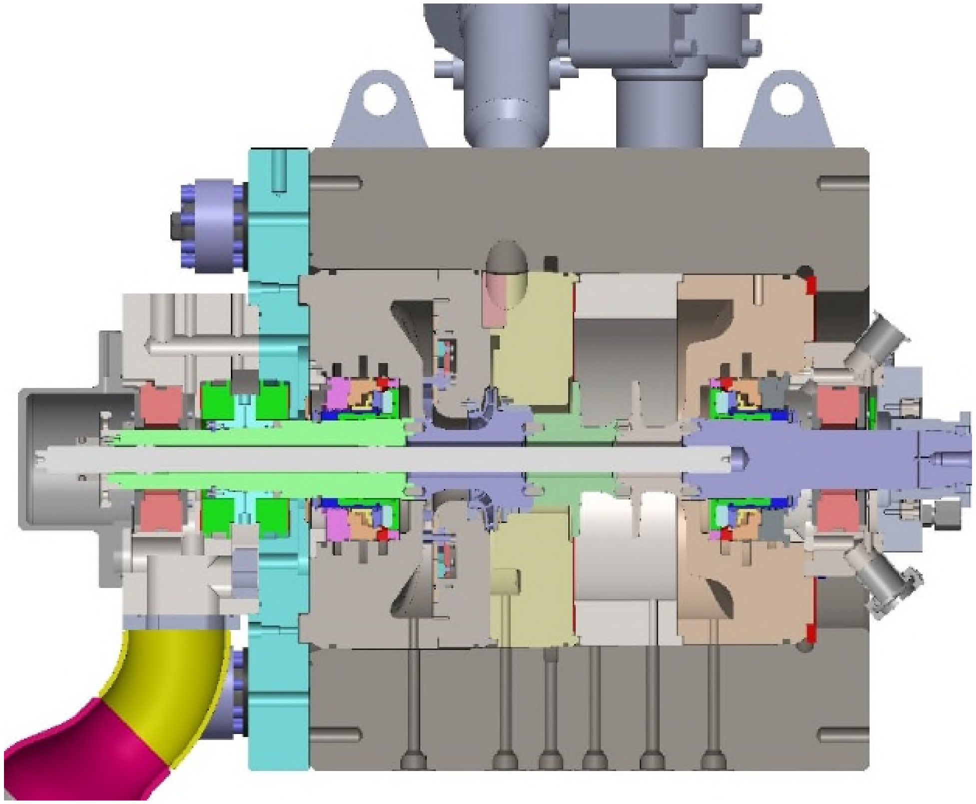
Example SCO2 Compressor



1. High speed coupling [Min Shaft Diameter Limit]
2. Tilting pad journal bearings [Max Shaft Diameter Limit]
3. End Seal (Dry Gas Seals)
4. Balance Piston [Thrust + Damping]
5. Main Compressor [Min Hub Diameter]
6. Bypass Compressor
7. Thrust Runner

Operating Conditions

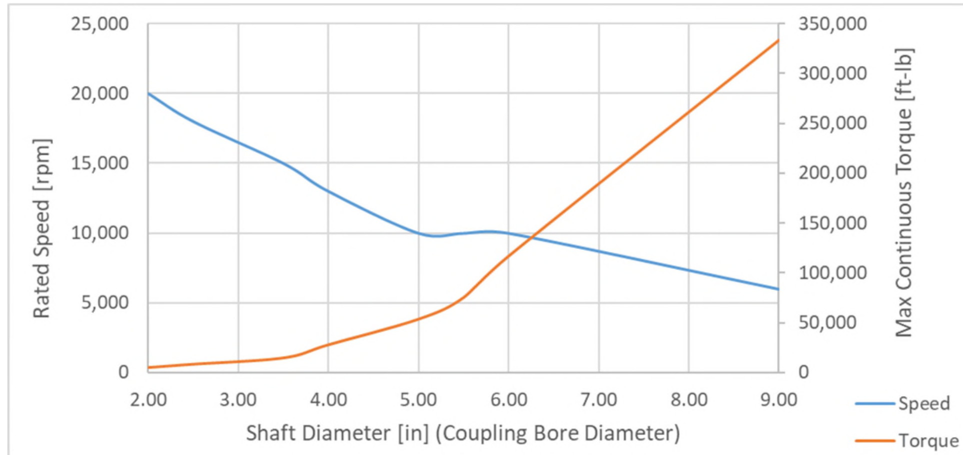
- Power: 6,570 hp (4.9 MW)
- Max Discharge Pressure: 4,850 psia (334 bar)
- Max Temperature: 400°F (200°C)
- Max Speed: 28,350 rpm (match turbine) [5% over Nominal]
- 2X swing in volume flow
- Suction Densities up to 51 lbm/ft^3 (817 kg/m^3)



Cich, S., Moore, J., Kulhanek, C., Mortzheim, J., 2020, "Development and Testing of a Supercritical CO2 Compressor Operating Near the Dome, Proceedings of 49th Turbomachinery Symposium, Houston, TX, September 2020, Turbomachinery Laboratory, Texas A & M University

Mechanical Design

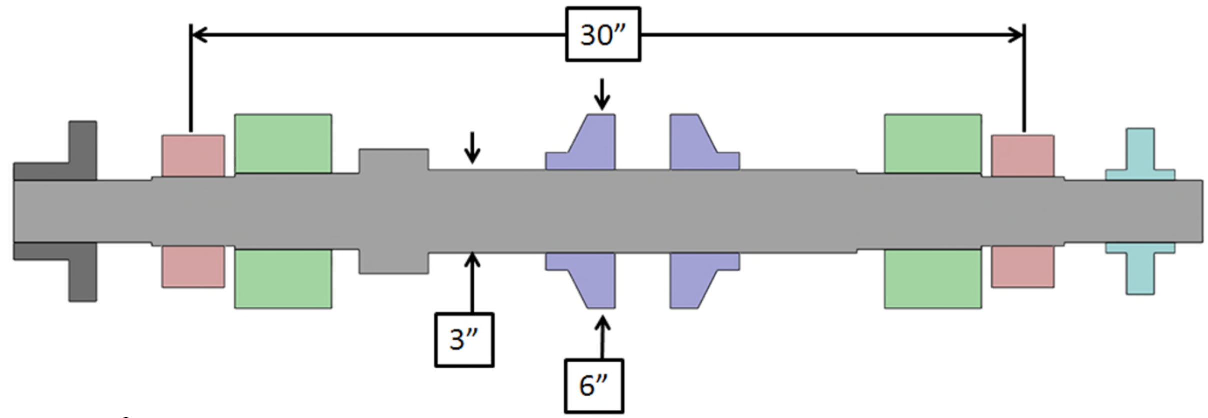
- While aero flow paths are smaller, support equipment (bearings, seals, shaft ends) must support high-power, high-pressure, and high-loads
- Design starts with shafts to look at max diameter based on surface speed and min diameter based on torque



[3]

$T = 63{,}025\frac{P}{w} = 63{,}025\frac{6570}{27000} = 15{,}336$	T = Torque, in-lb P = Power, hp w = Speed, rpm
$\tau = \frac{Tr}{J} = \frac{15336\ x\ 1.125}{2.52} = 6{,}857\ psi$	τ = Shear Stress, psi r = radius, in J= Polar moment of inertia, in^4
$J = \frac{\pi r^4}{2} = 2.52\ in^4$	

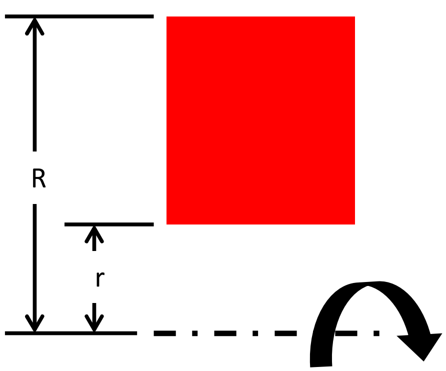
Mechanical Design



- Design Limit Considerations
  - Surface speed limits at Journal Bearings
  - Coupling speed limits
  - Peak tip speeds at impellers based on peak stresses in hub or blades
  - Torsional limits at shaft ends from torque and torsional modes
  - Bearings Span vs Hub Diameter

Mechanical Design – Rotor Stress Considerations

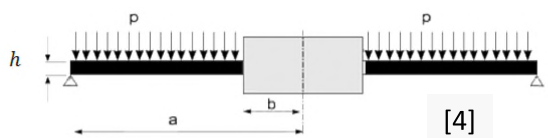
- Larger diameters → More head, more rotating weight, higher hub stress
- Smaller Inner Diameters → reduced tie bolt diameter, more rotating weight, reduced stresses
- Solid shafts have the lowest stresses but required monolithic shafts or more complicated joints
- Aero hub diameter may be smaller than shaft ends requiring special rotor construction



$\sigma = \frac{3+v}{8}\rho\omega^2\left[r^2 + 2R^2 - \frac{1+3v}{3+v}r^2\right]$	σ is peak stress at the smallest diameter, psi v is Poisson's ratio ρ is material density, lbm/in^3 ω is rotating speed, rps r is the inner diameter, in R is the outer diameter, in

Mechanical Design – Stator Considerations

- Consider stator components impact on rotor layout, especially for high pressure machines
- Look at diaphragm stresses and deflection
- Peak deflection limited by max opening at exit diffuser

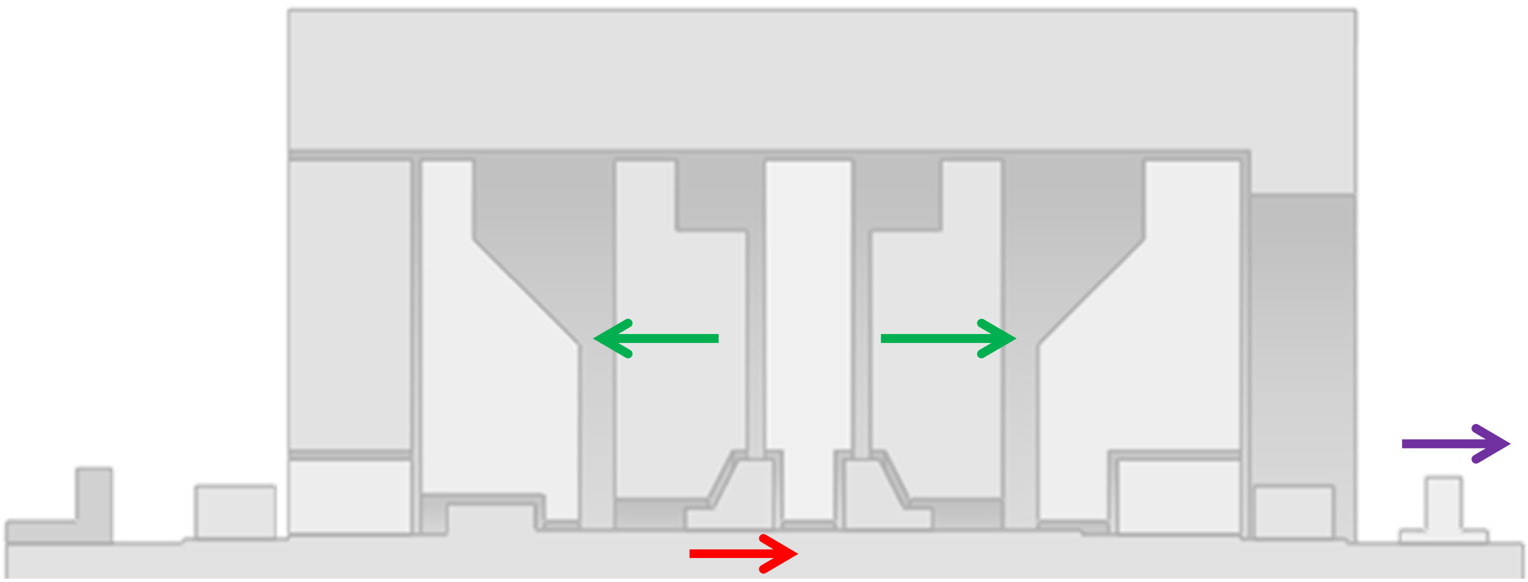


[4]

a/b	Simply Supported		Fixed Support	
	k1	k2	k1	k2
1.25	0.592	0.184	0.105	0.002
1.50	0.976	0.414	0.259	0.014
2.00	1.440	0.664	0.480	0.058
3.00	1.880	0.824	0.657	0.130
4.00	2.080	0.830	0.710	0.162
5.00	2.190	0.813	0.730	0.175

$\sigma_{max} = \frac{k_1 P a^2}{h^2}$	σ_{max} is the peak stress in the diaphragm, psi w_{max} is the max displacement at the ID of the diaphragm, in a is the outer radius, in
$w_{max} = \frac{k_2 P a^4}{E h^3}$	b is the inner radius, in P is the pressure, psi h is thickness, in E is modulus of elasticity, psi

Mechanical Design

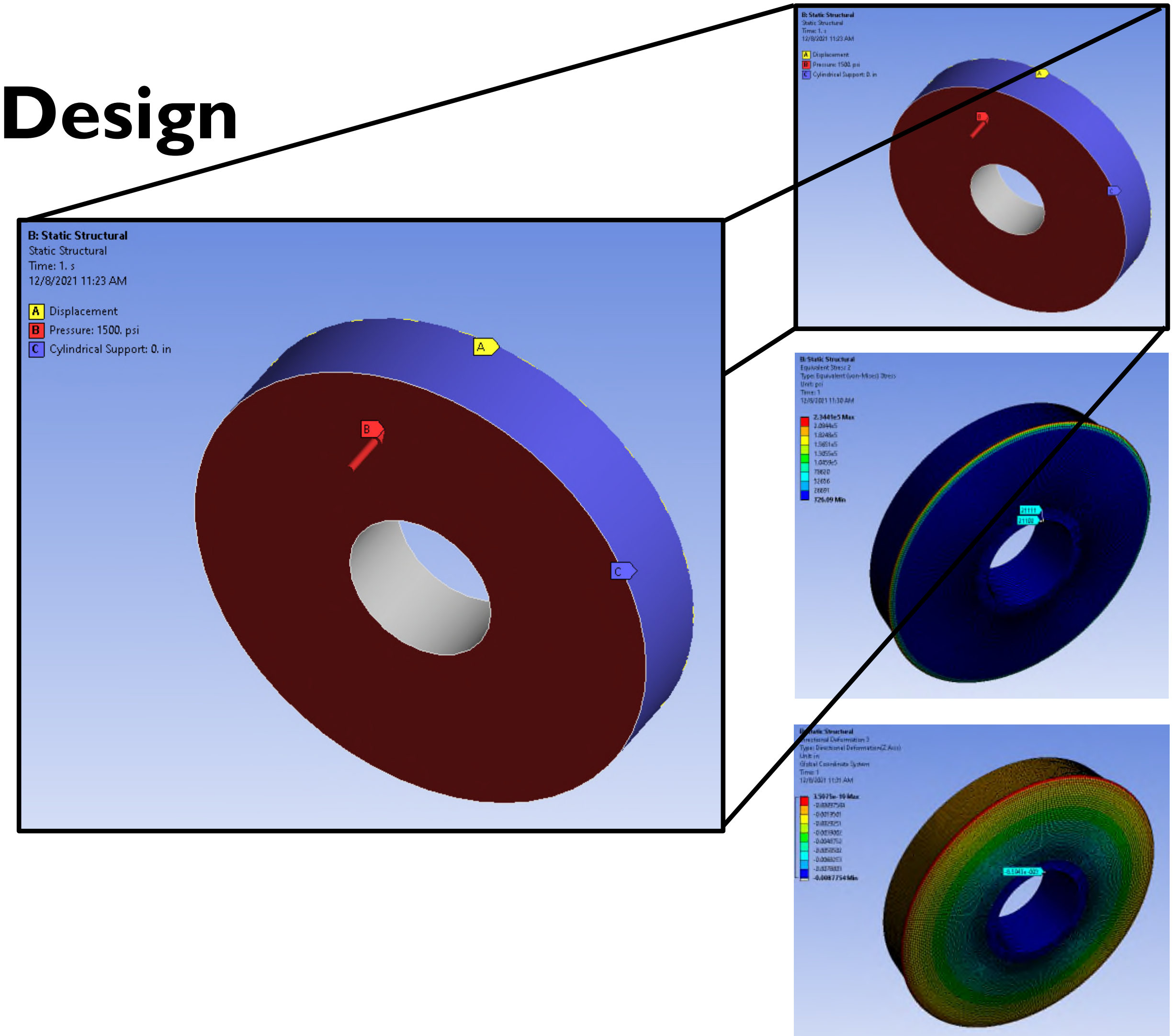


As Pressure Increases from Start Up:

- Housings and Heads Deflect Away from High Pressure – Opening of Diffuser
- Housing Deflection Pulls Rotor at Thrust Bearing
- Impeller Move to the Right

Mechanical Design

- Example Calculation
  - OD (2a): 16”; ID (2b): 5”
  - Pressure: 1,500 psi
  - Modulus: 29,000,000 psi
  - Simply Supported
- Plate Thickness (h): 3”
  - Hand Calcs: 20,550 psi & 0.0065”
  - FEA: 21,100 psi & 0.0086”
- Plate Thickness (h): 2”
  - Hand Calcs: 46,234 psi & 0.0219”
  - FEA: 46,839 psi & 0.0241”



Mechanical Design

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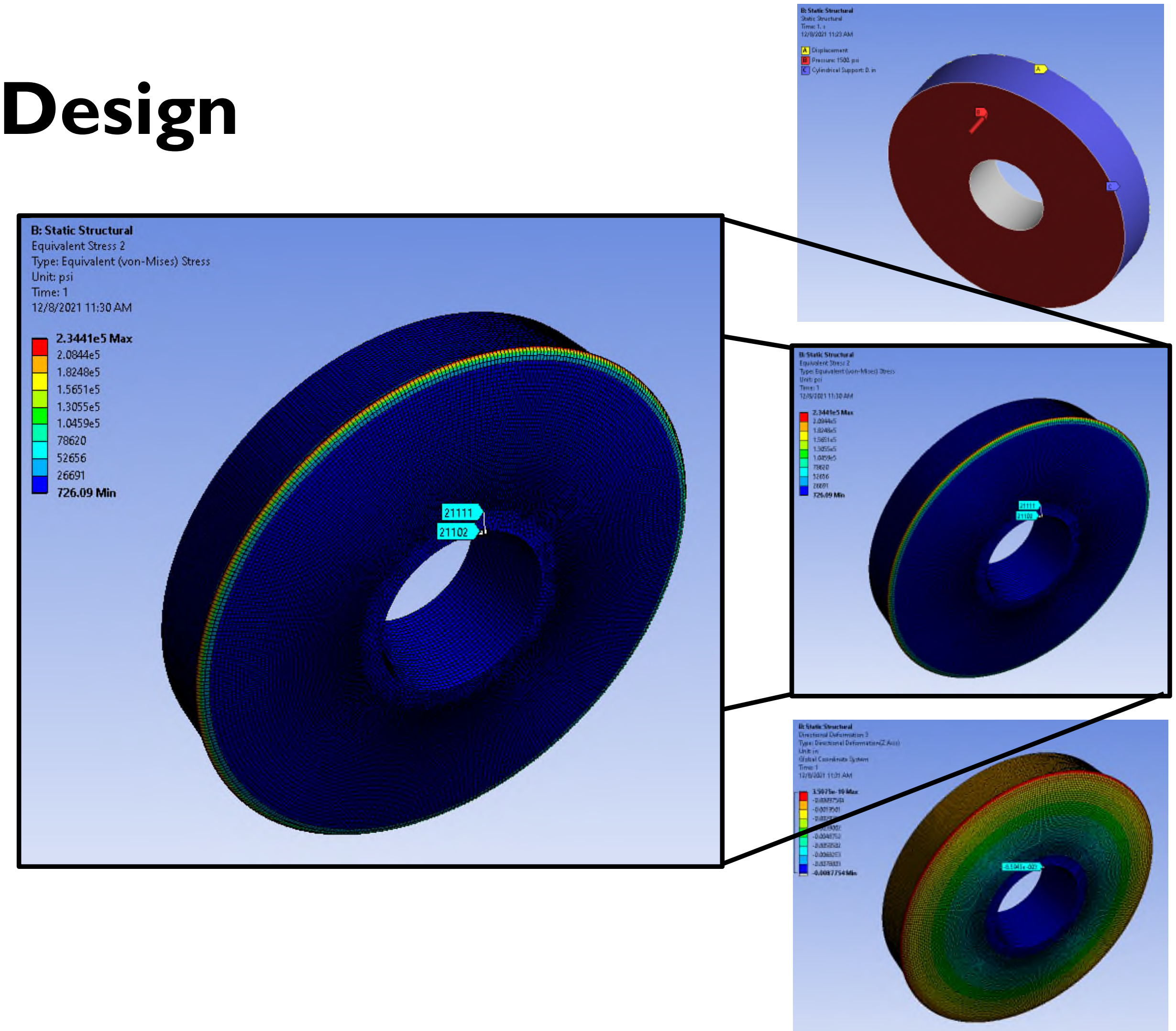

Mechanical Design

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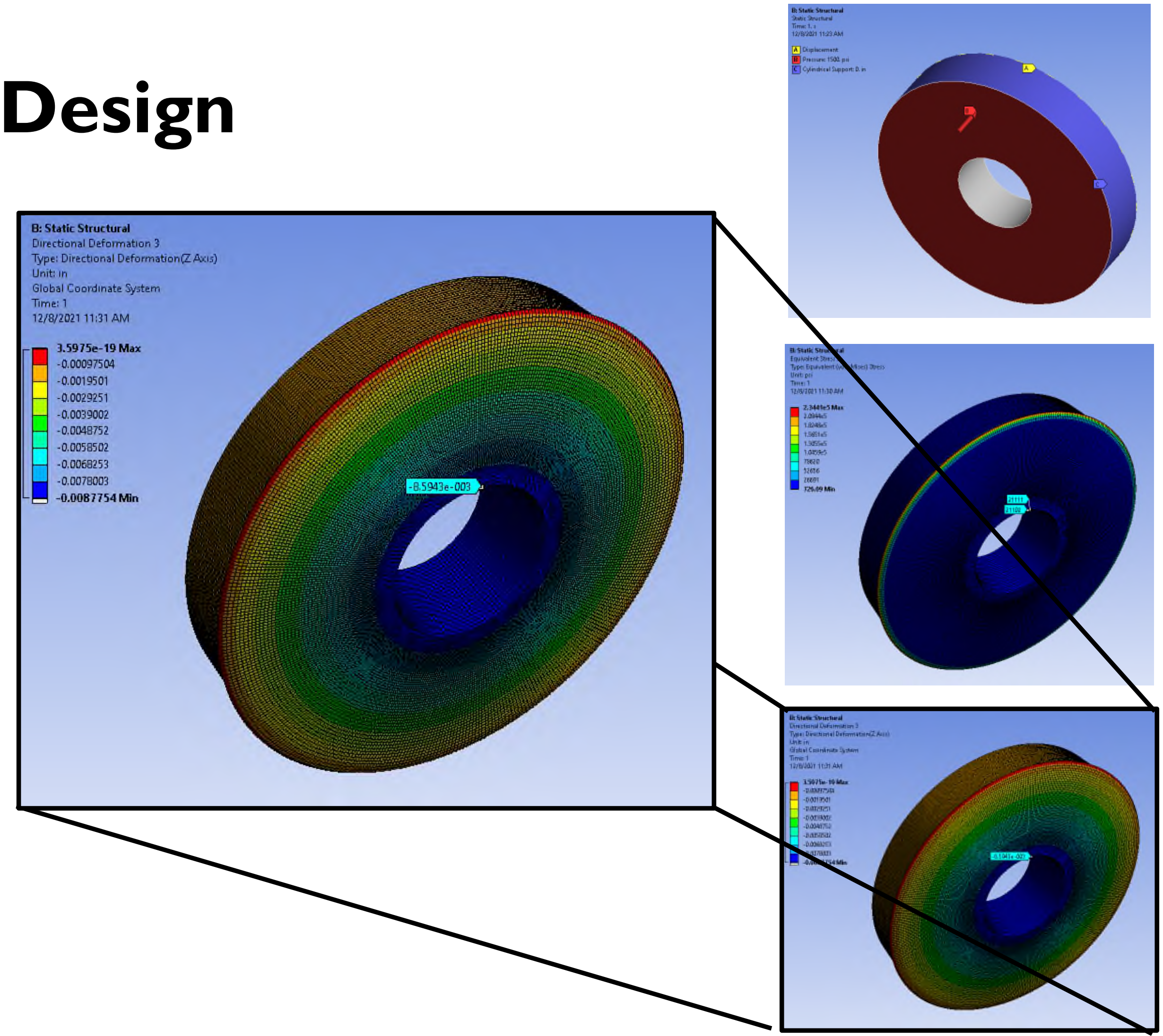
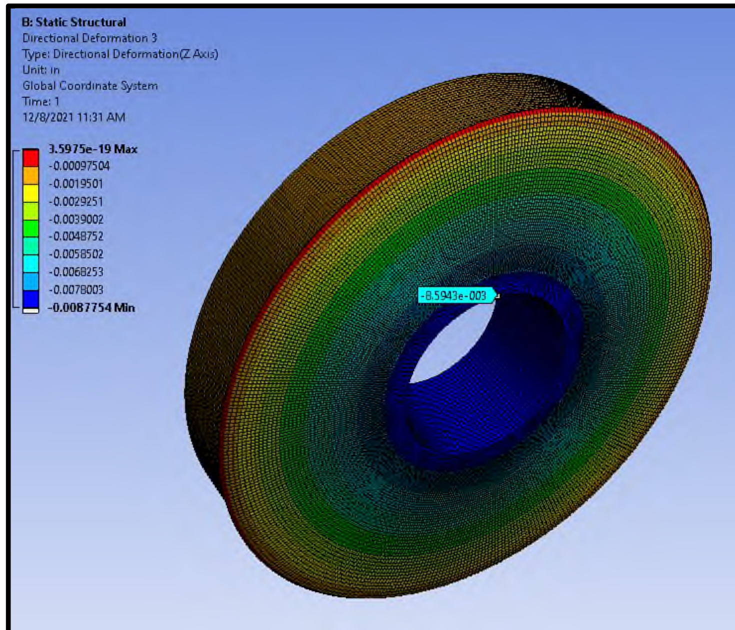
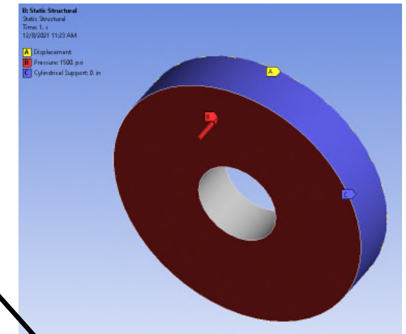
- Plate Thickness (h): 3”
  - Hand Calcs: 20,550 psi & 0.0065”
  - FEA: 21,100 psi & 0.0086”
- Plate Thickness (h): 2”
  - Hand Calcs: 46,234 psi & 0.0219”
  - FEA: 46,839 psi & 0.0241”

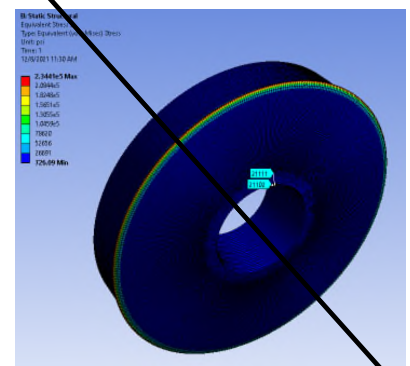


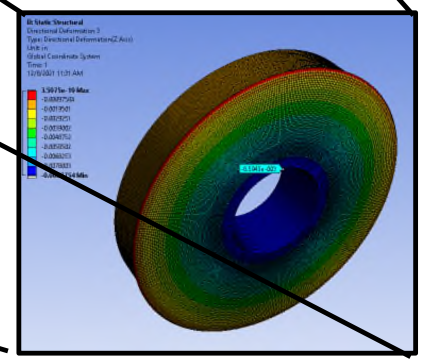
Mechanical Design

- Example Calculation
  - OD (2a): 16”; ID (2b): 5”
  - Pressure: 1,500 psi
  - Modulus: 29,000,000 psi
  - Simply Supported
- Plate Thickness (h): 3”
  - Hand Calcs: 20,550 psi & 0.0065”
  - FEA: 21,100 psi & 0.0086”
- Plate Thickness (h): 2”
  - Hand Calcs: 46,234 psi & 0.0219”
  - FEA: 46,839 psi & 0.0241”

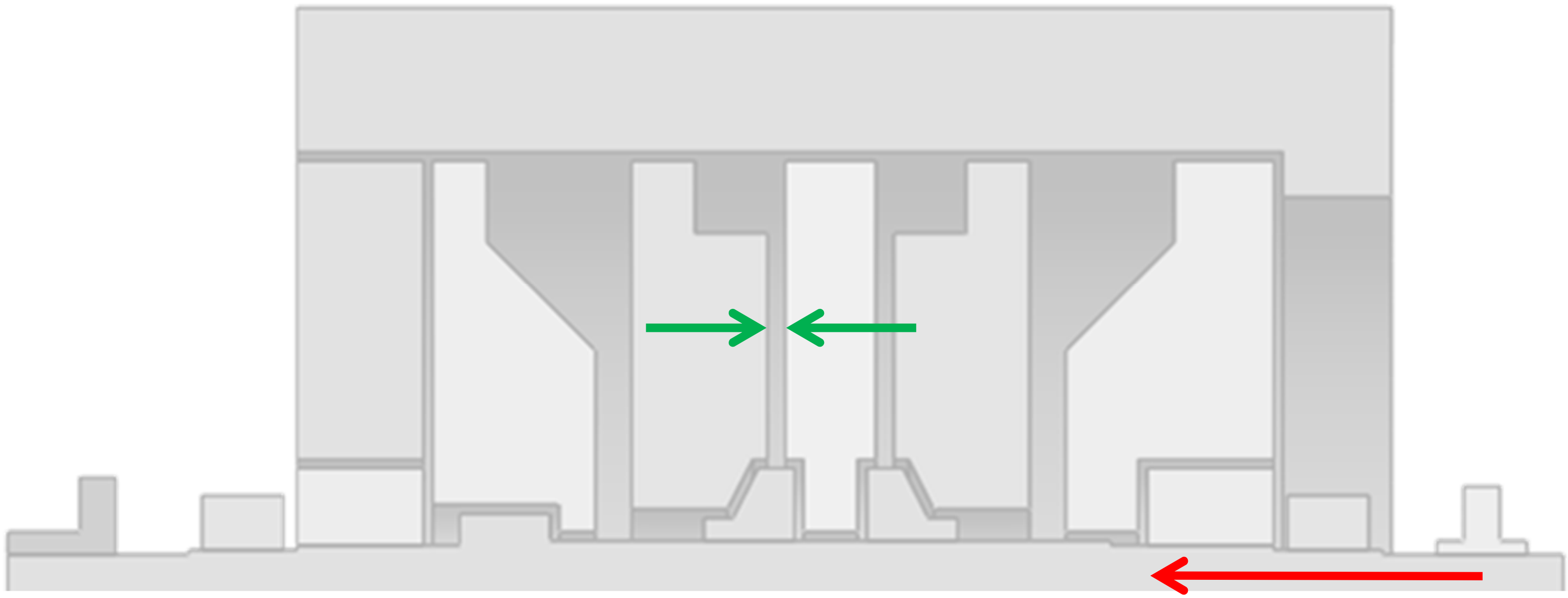








Mechanical Design



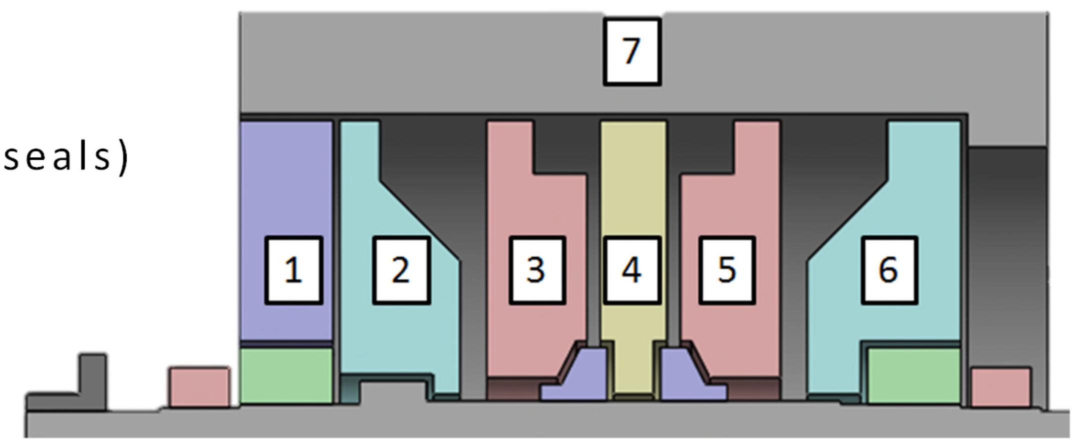
As Temperature Increases from Start Up:

- **Diffuser Closes Up at Designed Thermal Gap**
- **Rotor Grows Away from Thrust Bearing and Moves Impeller to the Left**

Mechanical Design

Design considerations for diaphragms

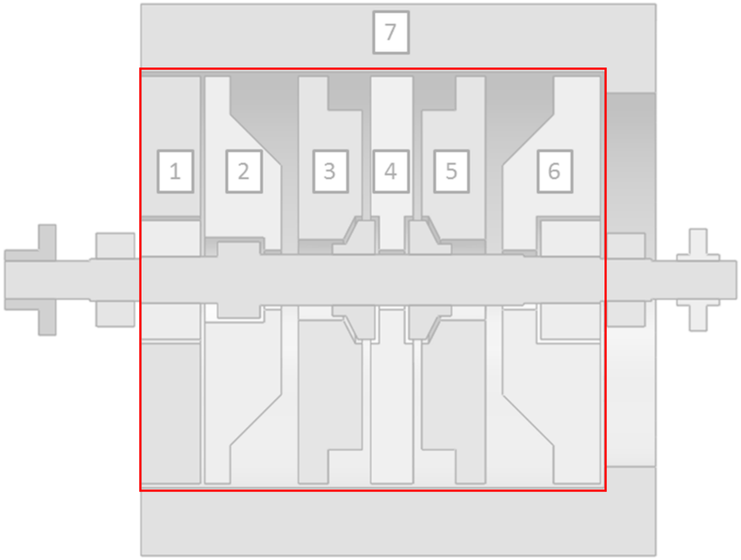
- If designing for MAX Displacement / Stress
  - Larger OD → Increased Thickness→ Longer Bearing Span → Reduced Rotor Modes
  - Higher Pressure → Increased Thickness → etc.
- Features that will impact OD
  - End Seal Diameter
  - Impeller Diameter
  - Diffuser Length
  - Exit Plenum flow area
  - Assembly features (bundle bolts, face seals)
  - Pilot fits



Mechanical Design

- Material Selection and Sizing
- Larger Bundle OD and lower Design stresses → Longer Rotor Span

OD	Material (Allowable Stress)			
in	20 ksi	30 ksi	40 ksi	50 ksi
10	11.13	9.15	7.98	7.21
12	14.92	12.27	10.69	9.77
14	18.54	15.25	13.29	12.27
16	22.06	18.15	15.81	14.72
18	25.54	21.01	18.36	17.16
20	29.01	23.87	21.00	19.63
22	32.49	26.73	23.67	22.13
24	35.89	29.53	26.33	24.63
26	39.06	32.14	28.85	26.99
28	41.67	34.29	30.96	28.96
30	43.15	35.50	32.09	29.98

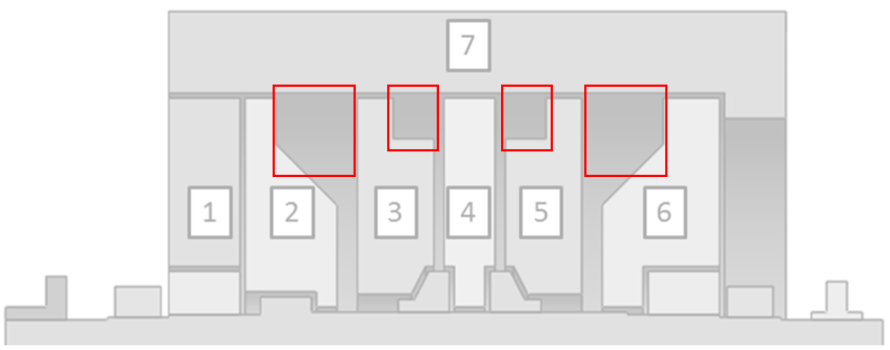


Mechanical Design

Flow path Sizing

- Lower velocity limits → larger flow areas → Increase bundle diameter (or case complexity) → reduced pressure losses
- Higher velocity limits → smaller flow areas → Reduced bundle diameter → erosion concerns & increased pressure loss

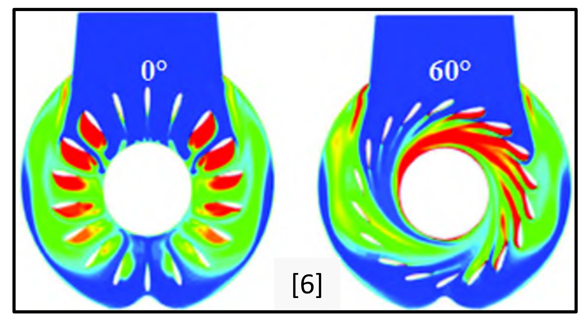
Section		Pressure		Temperature		Density	Mass Flow		Max Vel.	Min Dia.
		Mpa	psi	C	F	lbm/in^3	kg/s	lbm/s	ft/s	in
Main	Inlet	8.55	1,240	35	95	0.0224	70.3	155.0	80	3.03
	Exit	24.13	3,500	78	172	0.0247	70.3	155.0	80	2.88
Bypass	Inlet	8.69	1,260	88	190	0.0061	34.2	75.4	80	4.04
	Exit	23.99	3,479	194	381	0.0116	34.2	75.4	80	2.94
Main	Inlet	8.55	1,240	35	95	0.0224	70.3	155.0	150	2.21
	Exit	24.13	3,500	78	172	0.0247	70.3	155.0	150	2.11
Bypass	Inlet	8.69	1,260	88	190	0.0061	34.2	75.4	150	2.95
	Exit	23.99	3,479	194	381	0.0116	34.2	75.4	150	2.15



Mechanical Design

- How to manage large swings in suction density?
- Case Treatment (Passive)
  - Internal recirculation to maintain discharge flow
- Chilling / Heating (Active with Increased Power)
  - Reduce or Increase fluid density to match design
- Variable Inlet Guide Vanes (Active / Control Logic)
  - Increasing inlet swirl to reduce volume flow
  - Radial or Axial (typical for overhung)

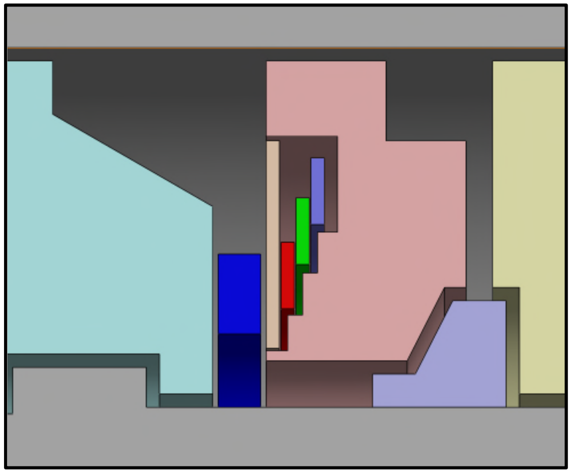
[5]

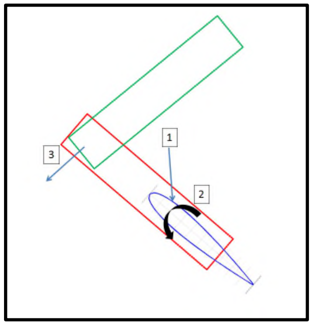


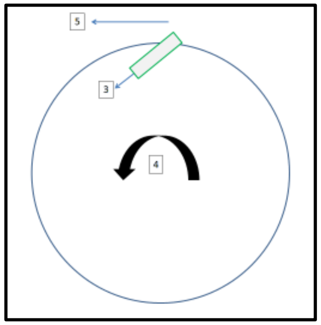
[6]

Mechanical Design

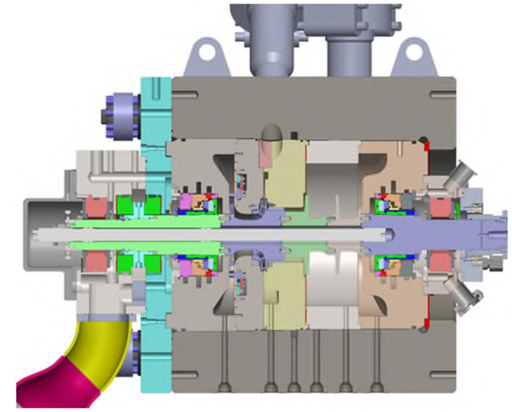
- Variable IGVs Selected
- How to Package an IGV System?
- Important to consider where the IGV mechanisms are housed and impact on diaphragm design
- Extra case penetrations and need to quantify loads on actuator control rod





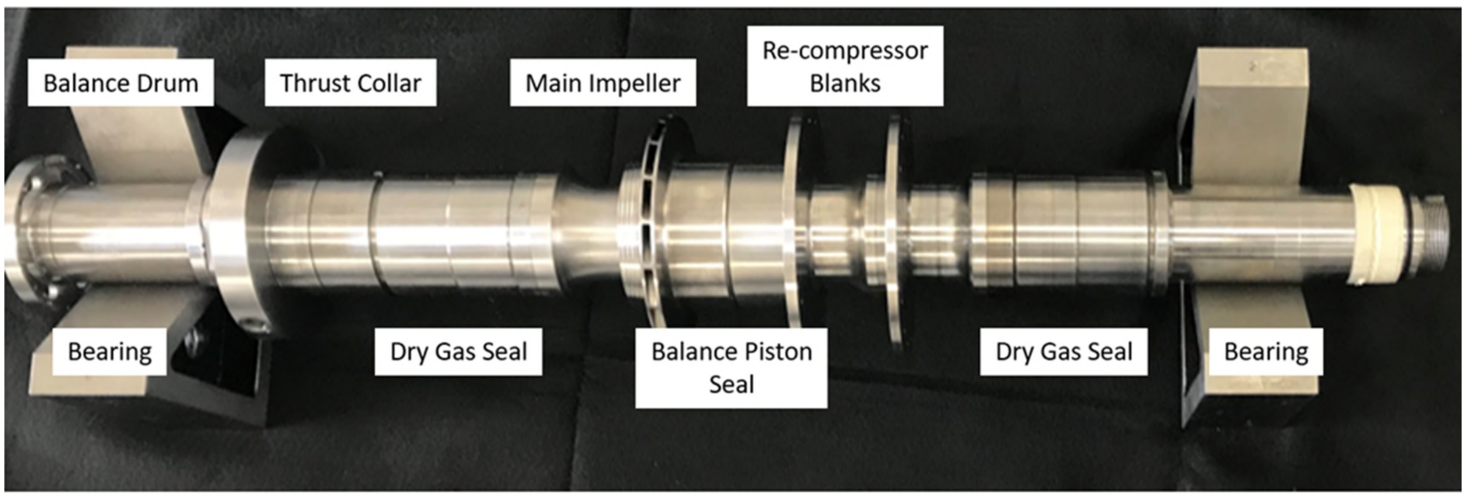


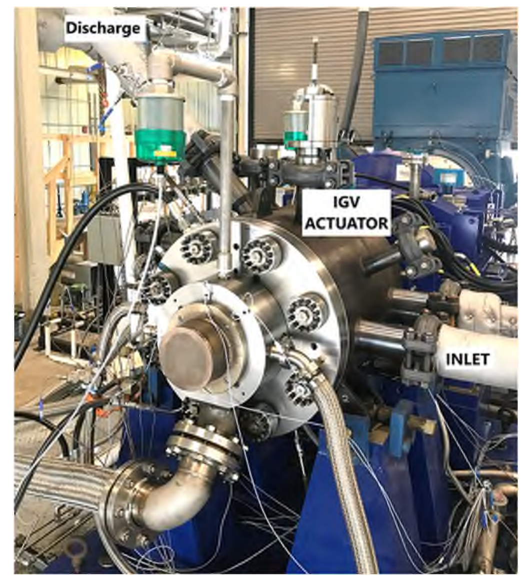
Final Embodiment – SwRI/GE Apollo Compressor



- Integrated main and bypass compressor in back-to-back arrangement
- Built-up rotor construction
- Variable IGVs
- Tested to full load, speed, and pressure

- One of highest density compressor in literature (720 kg/m3)





Cich, S., Moore, J., Kulhanek, C., Mortzheim, J., 2020, "Development and Testing of a Supercritical CO2 Compressor Operating Near the Dome, Proceedings of 49th Turbomachinery Symposium, Houston, TX, September 2020, Turbomachinery Laboratory, Texas A & M University

U.S. DEPARTMENT OF ENERGY | Energy Efficiency & Renewable Energy

GE Global Research

SwRI

Supercritical CO_2 Cycles Pressure Containment

Supercritical CO_2 Pressure Containment

Pressure Safety Specifications for Power Plant and Rotating Machinery

- ASME Section 8, Div 1, 2, 3

- API 610, “Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries”
  - References to ASME Section 8

- API 617, “Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Service”
  - Elements Regarding to Pressure Safety
  - Does not Cover Hot Gas Expanders Over 300°C
  - References to ASME Section 8

- EN 13445 “Unfired Pressure Vessels” and Pressure Equipment Directive 97/23/EC

ASME Section 8 Summary

A BRIEF DISCUSSION ON ASME SECTION VIII DIVISIONS 1 AND 2 AND THE NEW DIVISION 3

K.T.Lau, Ph.D., P.Eng., 3rd Annual Pressure Industry Conference, Banff, Alberta, Canada, February 1999.
Last Update - October, 2000

	Section VIII Division 1	Section VIII Division 2	Section VIII Division 3
	"Unfired" Pressure Vessel Rules	Alternative Rules	Alternative Rules for High Pressure
Published	< 1940	1968	1997
Pressure Limits	Normally up to 3000 psig	No limits either way, usually 600+ psig	No limit; Normally from 10,000 psig
Organization	General, Construction Type & Material U, UG, UW, UF, UB, UCS, UNF, UCI, UCL, UCD, UHT, ULT	General, Material, Design, Fabrication and others AG, AM, AD, AF, AR, AI, AT, AS	Similar to Division 2 KG, KM, KD, KF, KR, KE, KT, KS
Design Factor	Design Factor 3.5 on tensile (4* used previously) and other yield and temperature considerations	Design Factor of 3 on tensile (lower factor under reviewed) and other yield and temperature considerations	Yield based with reduction factor for yield to tensile ratio less than 0.7
Design Rules	Membrane - Maximum stress Generally Elastic analysis Very detailed design rules with Quality (joint efficiency) Factors. Little stress alaysis required; pure membrane without consideration of discontinuities controlling stress concentration to a safety factor of 3.5 or higher	Shell of Revolution - Max. shear stress Generally Elastic analysis Membrane + Bending. Fairly detailed design rules. In addition to the design rules, discontinuities, fatigue and other stress analysis considerations may be required unless exempted and guidance provided for in Appendix 4, 5 and 6	Maximum shear stress Elastic/Plastic Analyses and more. Some design rules provided; Fatigue analysis required; Fracture mechanics evaluation required unless proven leak-before-burst, Residual stresses become significant and maybe positive factors (e.g. autofrettage)
Experimental Stress Analysis	Normally not required	Introduced and may be required	Experimental design verification but may be exempted
Material and Impact Testing	Few restrictions on materials; Impact required unless exempted; extensive exemptions under UG-20, UCS 66/67	More restrictions on materials; impact required in general with similar rules as Division 1	Even more restrictive than Division 2 with different requirements.Fracture toughness testing requirement for fracture mechanics evaluation Crack tip opening displacement (CTOD) testing and establishment of KIc and/or JIc values
NDE Requirements	NDE requirements may be exempted through increased design factor	More stringent NDE requirements; extensive use of RT as well as UT, MT and PT.	Even more restrictive than Division 2; UT used for all butt welds, RT otherwise, extensive use of PT and MT
Welding and fabrication	Different types with butt welds and others	Extensive use/requirement of butt welds and full penetration welds including non-pressure attachment welds	Butt Welds and extensive use of other construction methods such as threaded, layered, wire-wound, interlocking strip-wound and others

ASME SECTION VIII-For Rotating Machinery

- Useful for Defining Safety Margins
  - 1.5X on Yield Strength, 3.5X on Ultimate Tensile Strength
- Useful for Defining Hydrostatic Test Requirements
  - 1.3X MAWP (Temperature Rated)
- Useful for Material Selection and Temperature/Stress De-rating

- Not Cognizant of Complicated Geometry Found in Turbomachinery (Can use Div 2 or 3 for FEA)
- Transient Thermal Stresses

U-1(c)(1) The scope of this Division has been established to identify the components and parameters considered in formulating the rules given in this Division. Laws or regulations issued by municipality, state, provincial, federal, or other enforcement or regulatory bodies having jurisdiction at the location of an installation establish the mandatory applicability of the Code rules, in whole or in part, within their jurisdiction. Those laws or regulations may require the use of this Division of the Code for vessels or components not considered to be within its Scope. These laws or regulations should be reviewed to determine size or service limitations of the coverage which may be different or more restrictive than those given here.

U-1(c)(2) Based on the Committee's consideration, the following classes of vessels are not included in the scope of this Division; however, any pressure vessel which meets all the applicable requirements of this Division may be stamped with the Code U Symbol:

(a) those within the scope of other Sections;

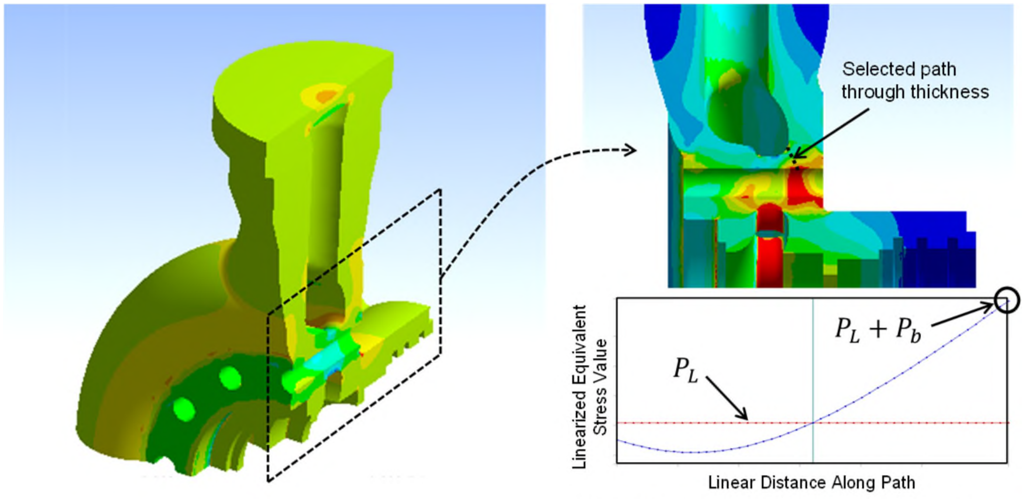
(b) fired process tubular heaters;

(c) pressure containers which are integral parts or components of rotating or reciprocating mechanical devices, such as pumps, compressors, turbines, generators, engines, and hydraulic or pneumatic cylinders where the primary design considerations and/or stresses are derived from the functional requirements of the device;

SunShot Casing Example

- P_m is the general primary membrane equivalent stress, equal to the average value through the section
- P_L is the local primary membrane equivalent stress.
- P_b is the primary bending equivalent stress. The value of $P_L + P_b$ is equal to the highest value through the section, excluding secondary and peak stresses.
- σ_1, σ_2, and σ_3 are the calculated principal stresses at each point

$$P_m \leq S$$
$$P_L \leq 1.5S$$
$$P_L + P_b \leq 1.5S$$
$$\sigma_1 + \sigma_2 + \sigma_3 \leq 4S$$

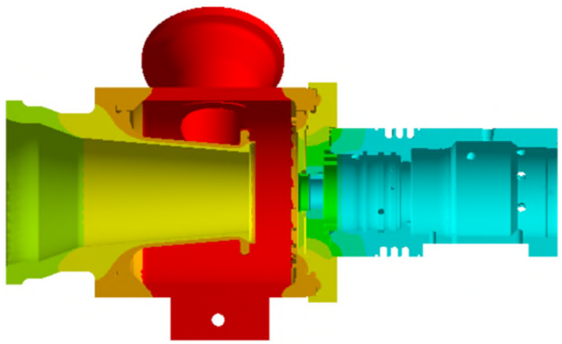


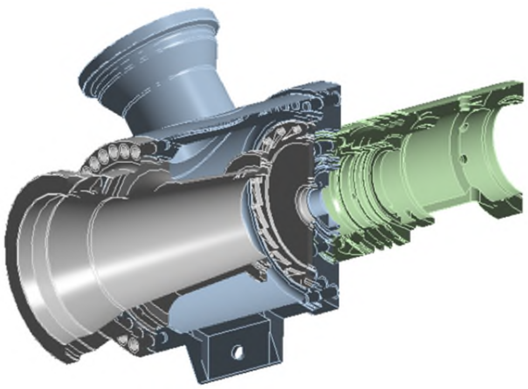
(a) Sum of Principal Stresses

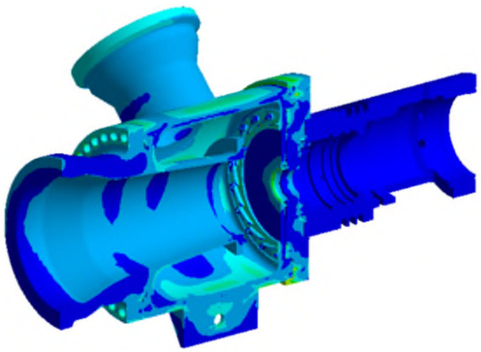
(b) Equivalent (von Mises) Stress and Linearized Stress Components Along Selected Path

Radial Turbine Housing – Operating Stress Example

- Use FEA for operating temperature
  - Use appropriate film coefficients
- Use FEA for operating stresses
  - Pressures
  - Nozzle Loads
- Define limits using material allowable stresses
  - ASME Allowable Stresses or Other
- Iterate the Design to Satisfy Requirements

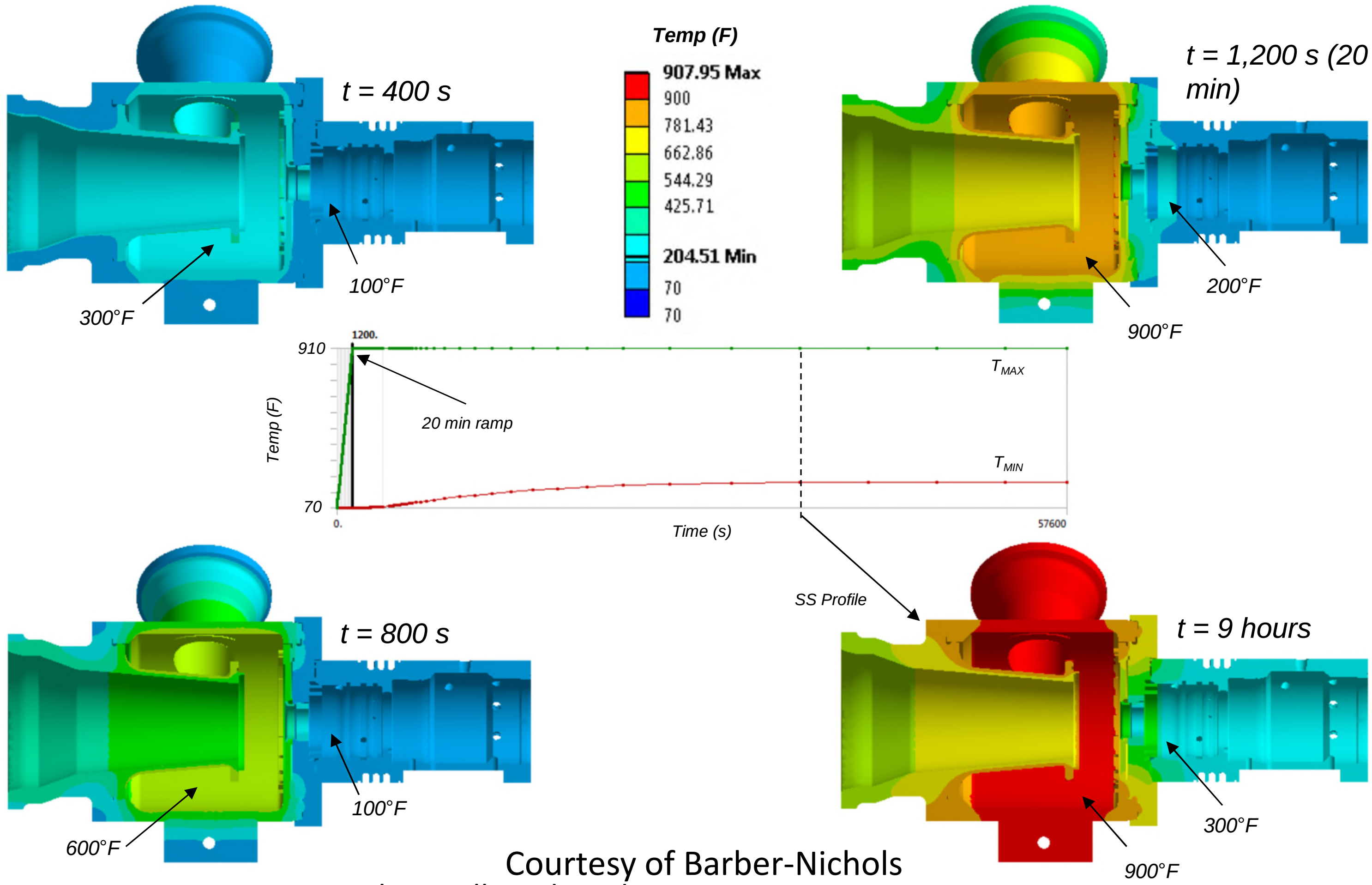






Courtesy of Barber-Nichols

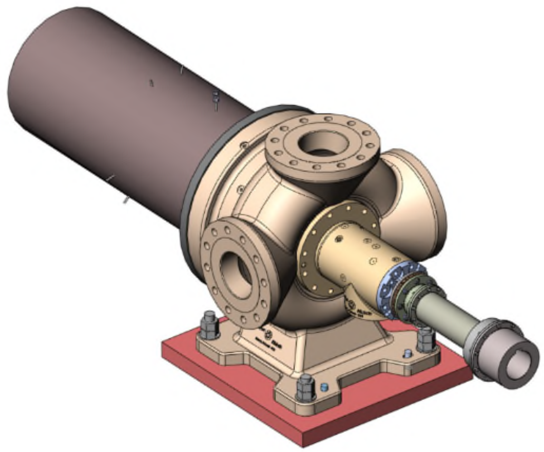
Radial Turbine Housing – Transient Thermal Profile

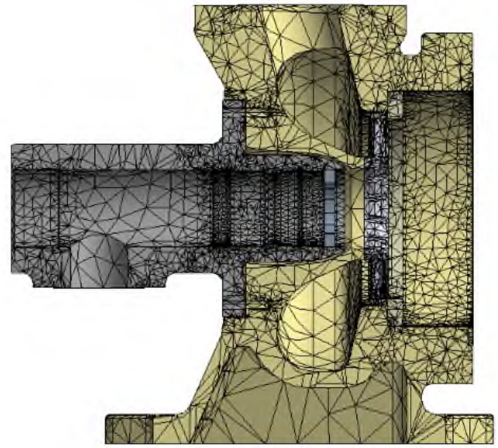


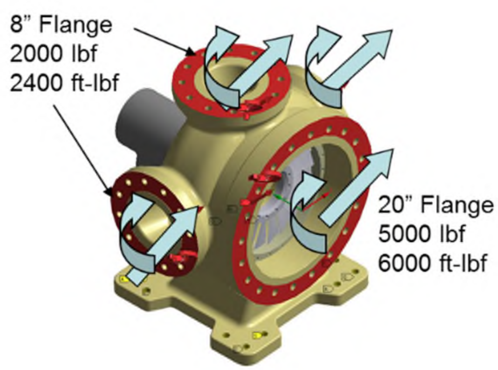
Courtesy of Barber-Nichols

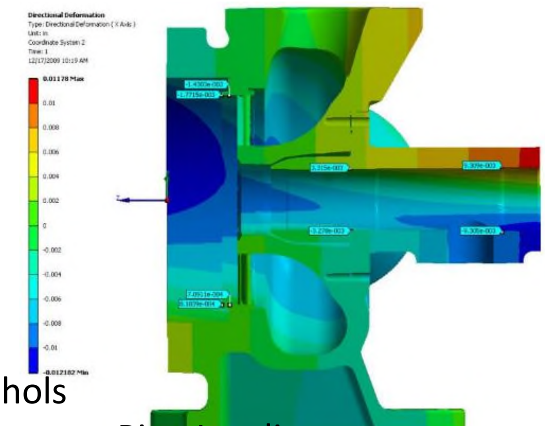
Thermally Induced Stresses Can Limit Startup Time

Single Stage Axial Turbine Example









Courtesy of Barber-Nichols

Combined Thermal, Pressure, Pipe Loading

sCO_2 Turbomachinery Materials

Supercritical CO_2 Cycles Material Selection

- CO_2 Metal Compatibility/Corrosion
  - Low Temperature -40°C to 150°C
    - Medium Chrome Steels
  - Medium Temperature 150°C to 300°C
  - High Temperature 300°C+
- CO_2 Seal Material Compatibility
  - Elastomeric
  - Rotating Shaft Seals
  - High Temperature Seals

Material Selection CO_2 Corrosion

- Oil Business
  - Pipeline Corrosion
    - Usually due to water or other constituents
- Specific to sCO_2 Power
  - MIT
  - Oakridge NL
  - Sandia NL
  - University of Wisconsin

Material Selection CO_2 Corrosion

Gibbs, MIT 2010, for Nuclear Reactor Use

- 610C and 20 MPa, 3,000 hour test
- F91, HcM12A, 316SS, 310SS, AL-6XN, Haynes 230, Alloy 625, PE-16, PM2000

- Highest Chromium and Nickel Content are Best

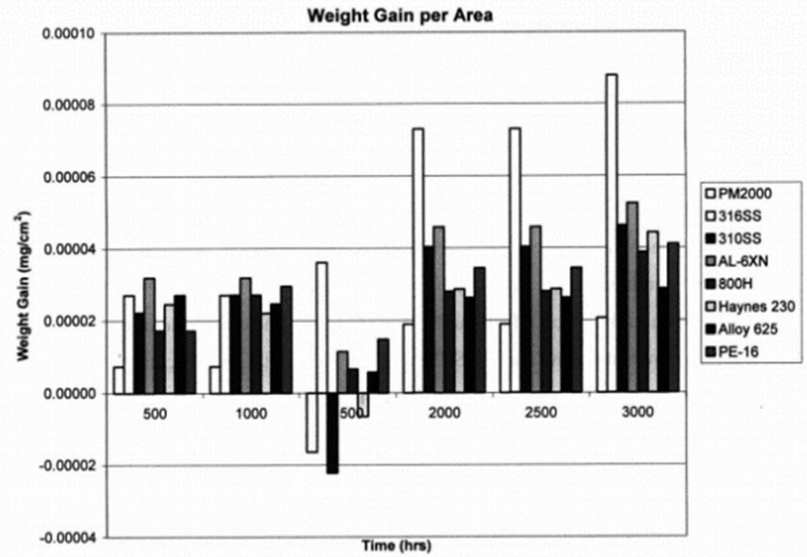


Figure 3.8: Weight gain per area, Alloys F91 and HCM12A not present

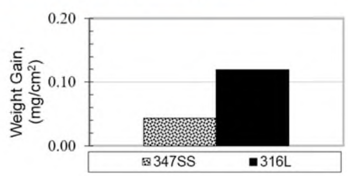
Table 3-2: 3000 hr weight gain

Alloy	Weight gain at 3000 hours (mg/cm^2)
F91	4.1×10^{-3}
HCM12A	5.5×10^{-3}
PM2000	2.1×10^{-5}
316SS	8.7×10^{-5}
310SS	4.6×10^{-5}
AL-6XN	5.2×10^{-5}
800H	3.9×10^{-5}
Haynes 230	4.4×10^{-5}
Alloy 625	2.9×10^{-5}
PE-16	4.1×10^{-5}

Alloy Corrosion Tests (UW-Madison)

Alloy	C	Fe	Cr	Ni	Mn	**Nb**	**Mo**	Si	Cu	Co
316L	0.045	64.3	17.4	13.3	1.7	**-**	**2.7**	0.43	-	-
347ss	0.051	68.5	17.7	9.62	1.66	**0.72**	**0.38**	0.77	0.38	0.20

200 hours exposure to CO_2 at 650C and 200 bar:



- Oxide Formation Increases
- Material Spalls (Corrosion and Erosion)

Alloy Corrosion Tests (UW-Madison)

Alloy	C	Fe	Cr	Ni	Mn	**Nb**	**Mo**	Si	Cu	Co
316L	0.045	64.3	17.4	13.3	1.7	**-**	**2.7**	0.43	-	-
347ss	0.051	68.5	17.7	9.62	1.66	**0.72**	**0.38**	0.77	0.38	0.20

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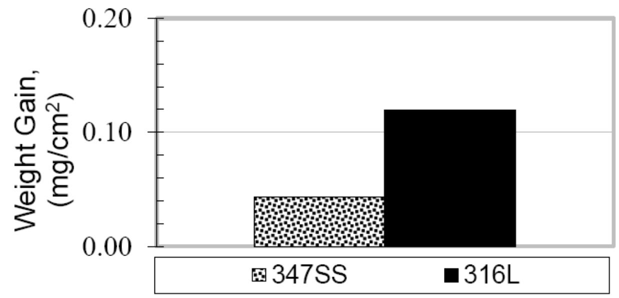
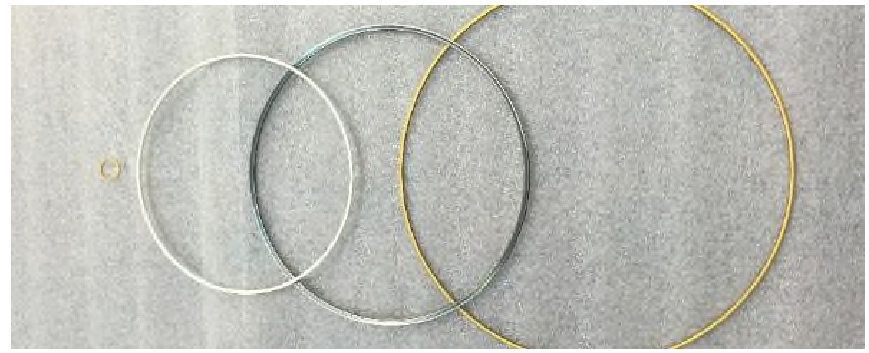
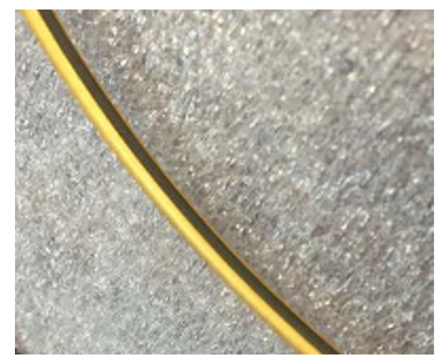


Materials Selection CO_2 Seals

- Static Seals, Elastomeric Seals Can Absorb High Pressure CO2. Rapid Depressurization Can Then Destroy the Seals
  - XNBR, HNBR, Available Bulk Purchase Only
  - EPDM, Widely Available, less suitable
  - Kalrez
- Rotating Shaft Seals
  - Aluminum, Teflon, PEEK, Graphite for Labyrinth Seals
  - Graphite, Si-Carbide, and Si-Nitride Liftoff Gas Seals
- High Temperature Static Seals
  - Silver Plated Inconel “C” Seals
- Electric Machines (rapid decompression testing)
  - Most Common Insulation Materials withstand sCO_2 Operation
  - MW35C wire insulation tested
  - Epoxy Type Varnish Works Best

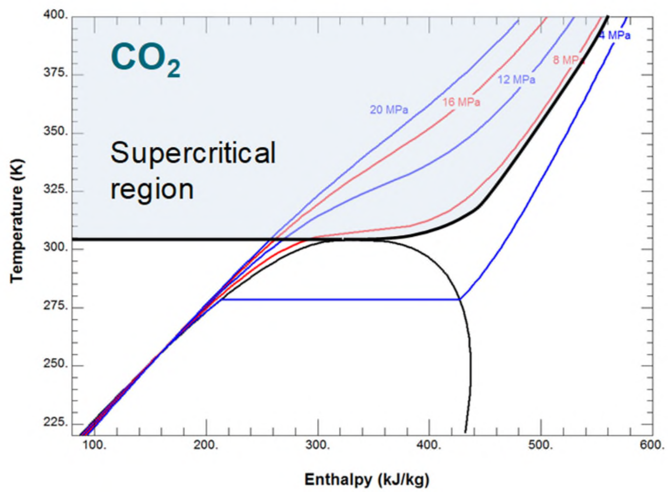


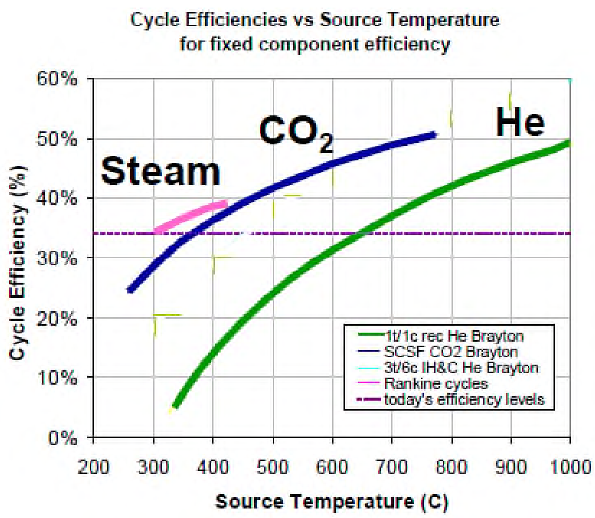


Summary

Both supercritical power cycles and the use of sCO_2 are not new concepts

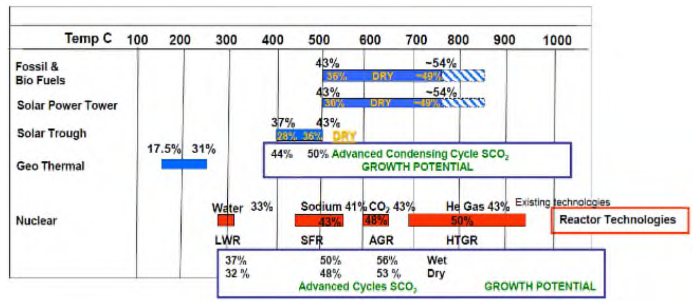
sCO_2 is desirable for power cycles because of its near-critical fluid properties



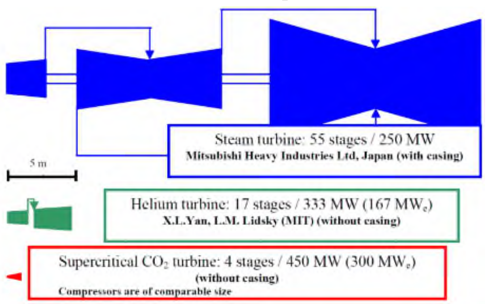


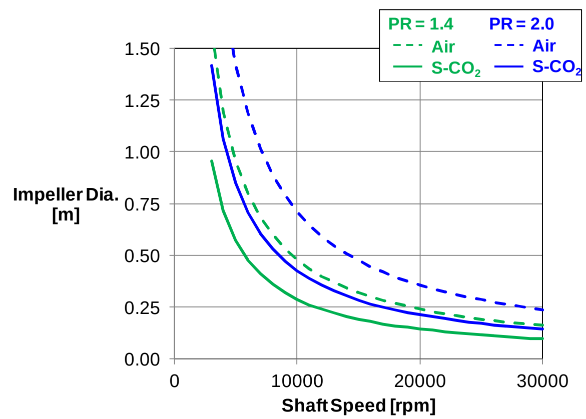
sCO_2 power cycles can be applied to many heat sources and have a small footprint

The near ambient critical temperature of CO_2 allows it to be matched with a variety of thermal heat sources



The combination of favorable property variation and high fluid density of sCO_2 allows small footprint of machinery





SunShot is helping to achieve these goals.

Summary

- sCO_2 Cycle can provide over 50% thermal efficiency
- sCO_2 Turbomachinery Require many additional considerations
- Real gas properties important for aero prediction and rotordynamics
- Gas density high – rotordynamics and blade dynamics
- High heat transfer – thermal management and pressure containment
- Material compatibility – high temperature and seals
- Requires design that can accommodate high thermal gradients with high pressure containment
- High power density results in challenges in packaging and driven equipment matching