Tutorial: Turbo Machinery Design for Supercritical CO2 Applications

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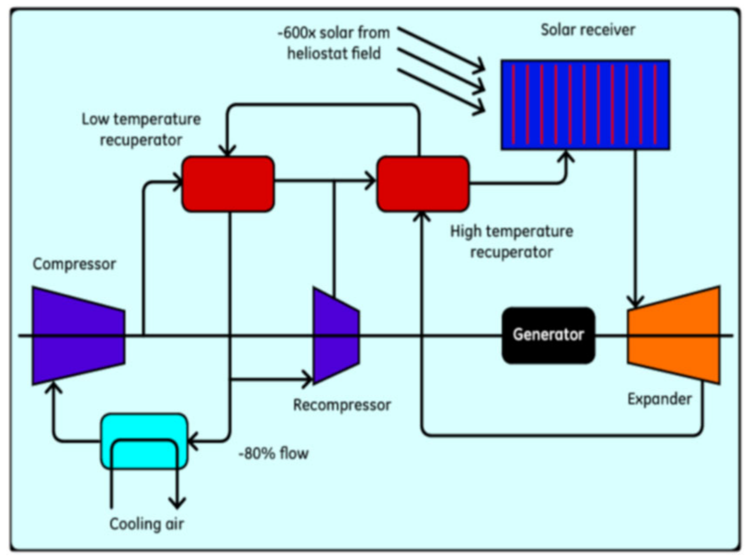


SCO2 Workshop

San Antonio, TX

February 21, 2022

Outline

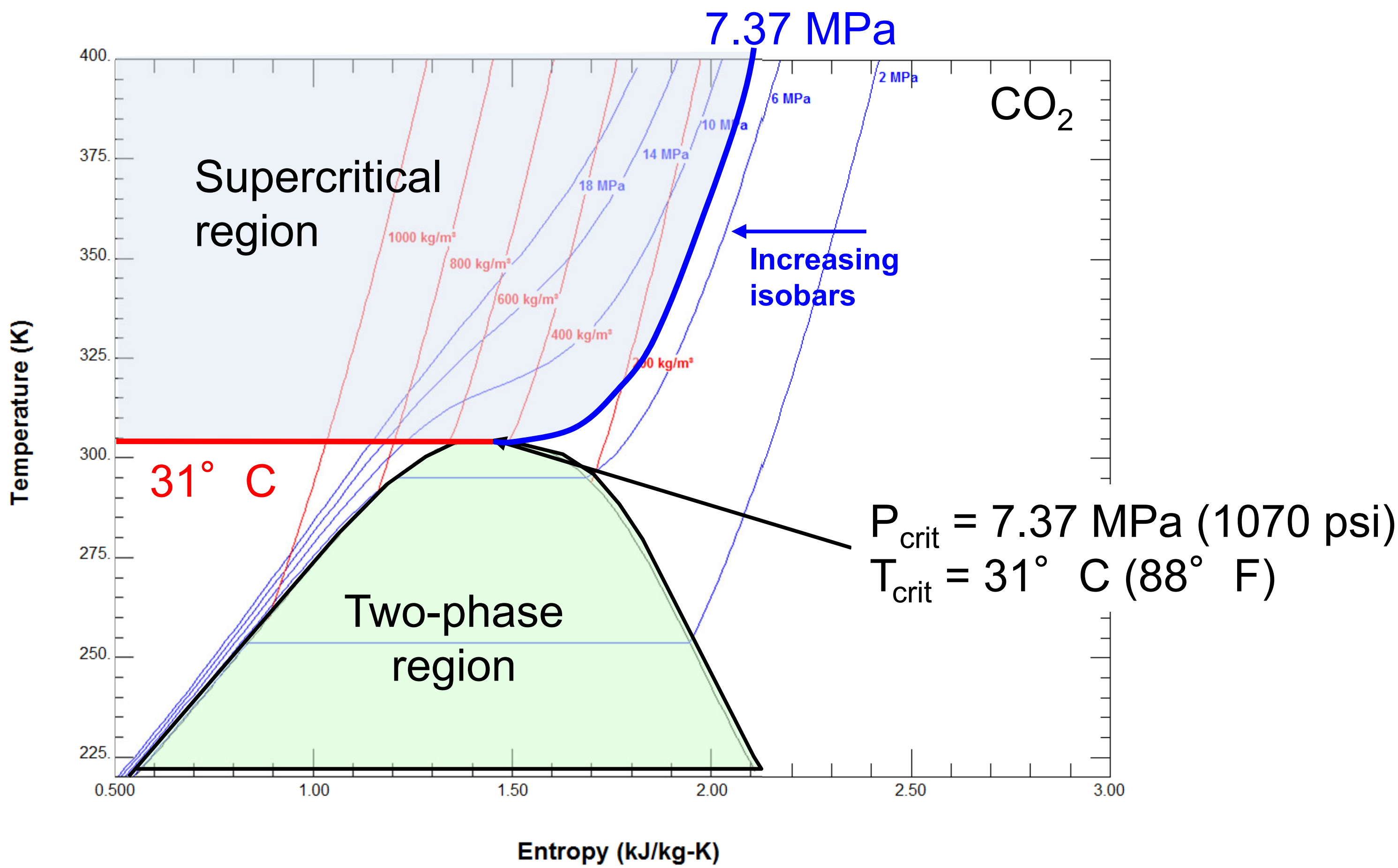
- Supercritical CO2 gas properties SCO2
- Closed Loop Brayton Cycle
- Seals and Seal design – damper seals, dry gas seals
- Bearings: Gas and Oil Hydrodynamics
- Rotordynamics
- Blade Loading and Dynamics
- Thermal management – Blade cooling, rotor, casing, dry gas seals
- Packaging considerations – KW to 100 MW, high speed generator, gearbox experience

Outline Cont.

- Pumps/Compressor/Turbine Aero Designs
- Materials For CO2
- Pressure containment

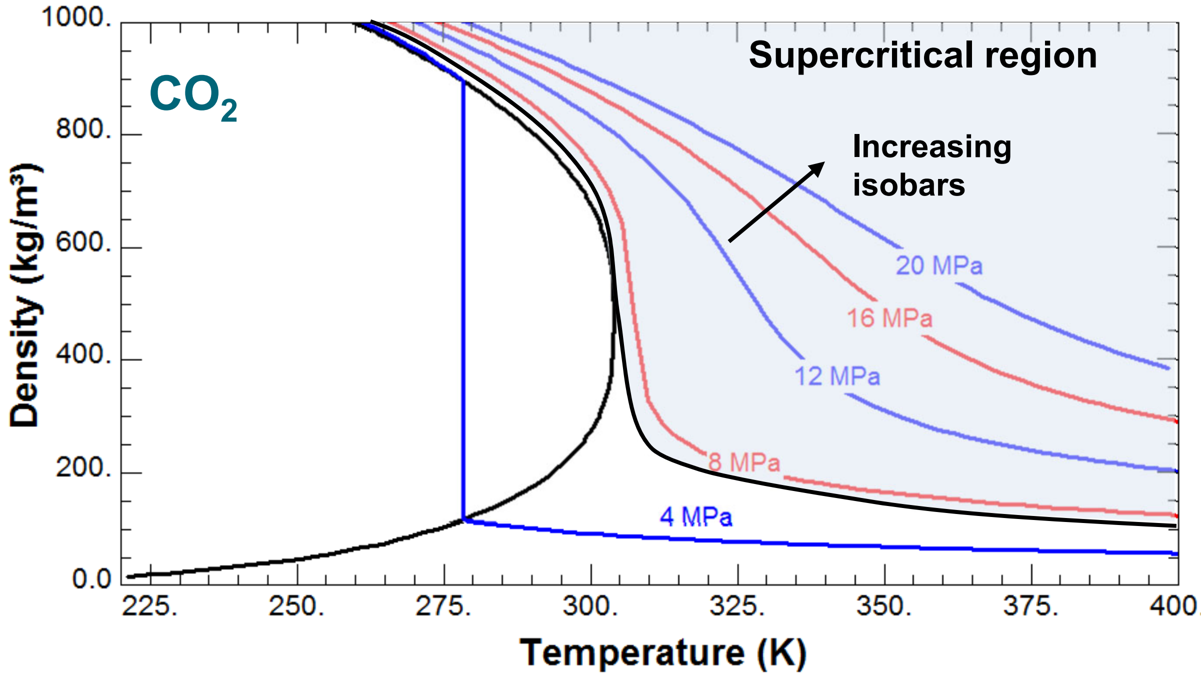
Supercritical CO_2 gas properties
SCO_2 Close Loop Brayton Cycle

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



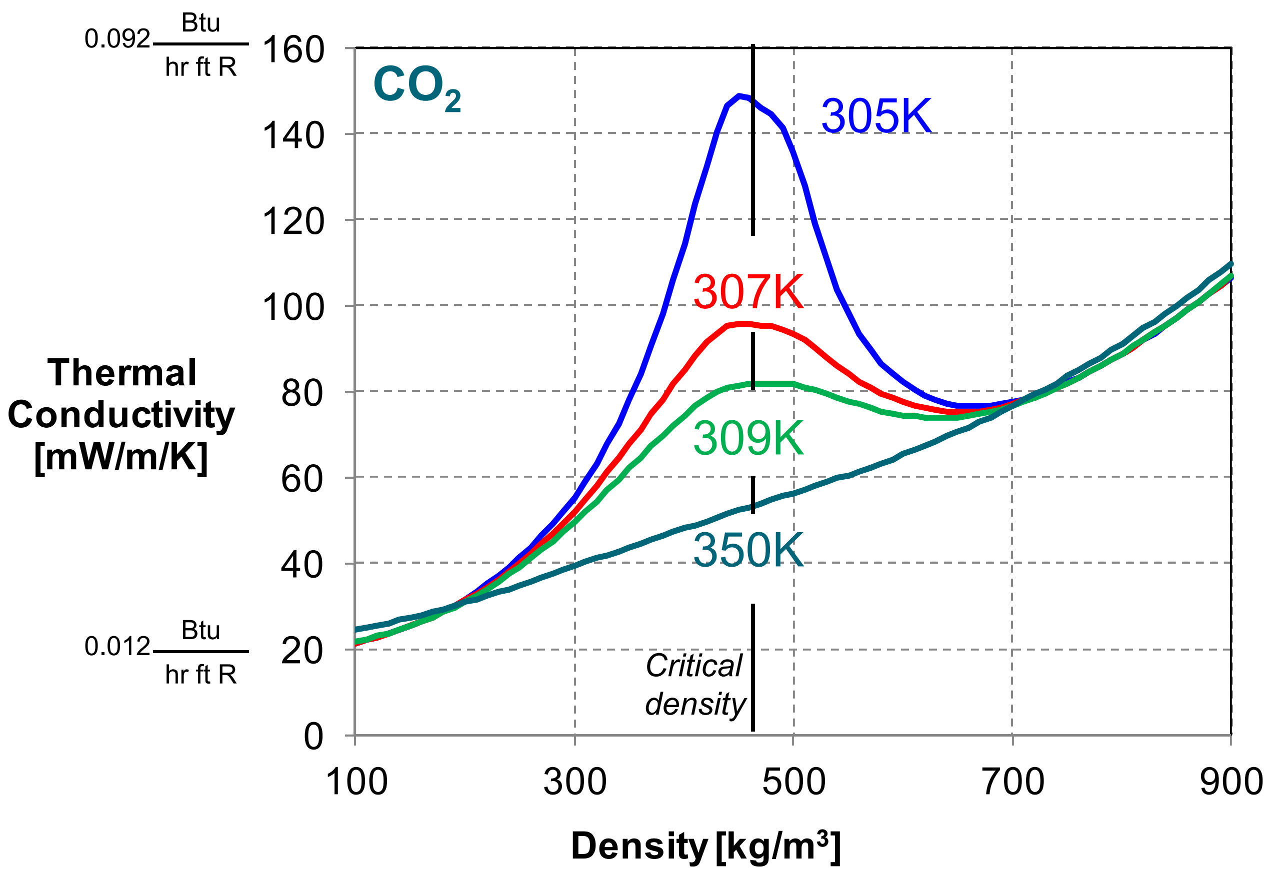
REFPROP (2007), EOS CO_2: Span & Wagner (1996)

Fluid density sharply decreases near the critical point



REFPROP (2007)

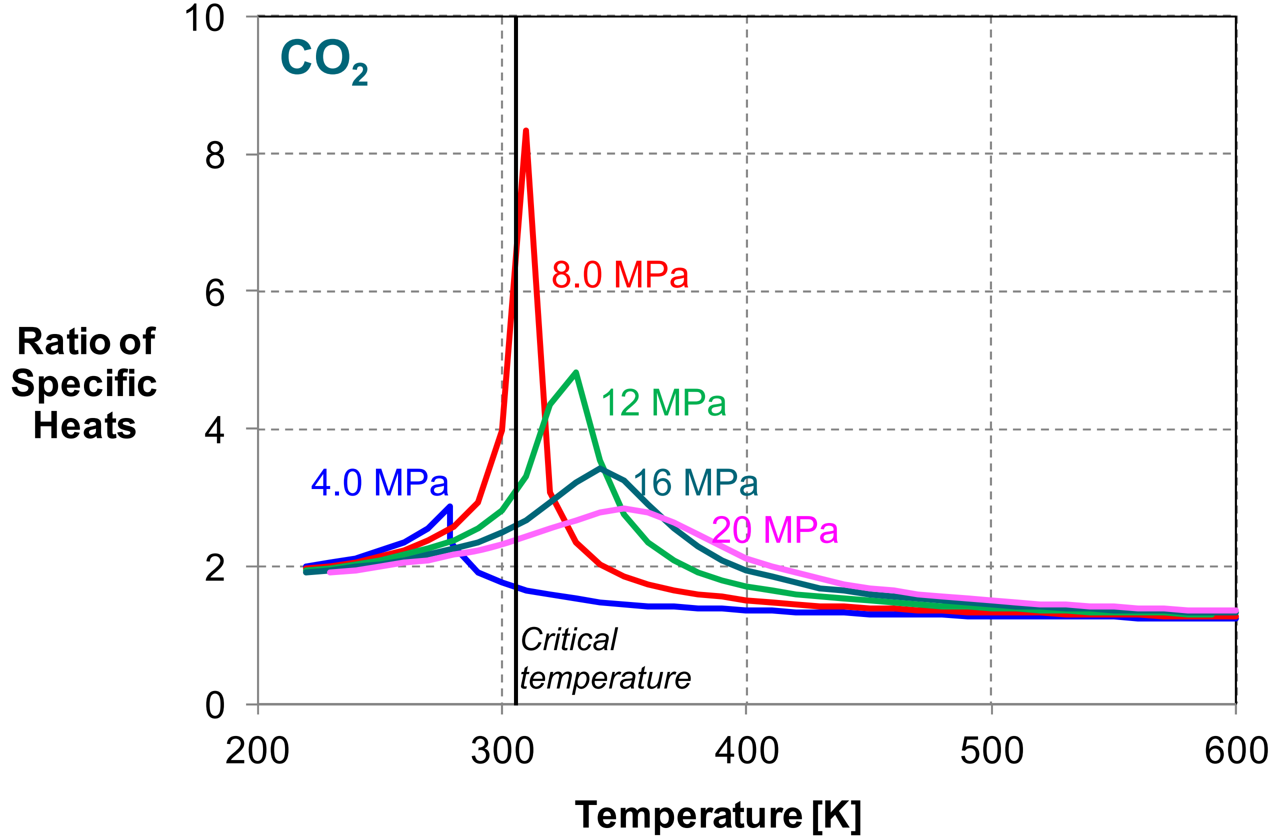
Fluid thermal conductivity is enhanced near the critical region



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)

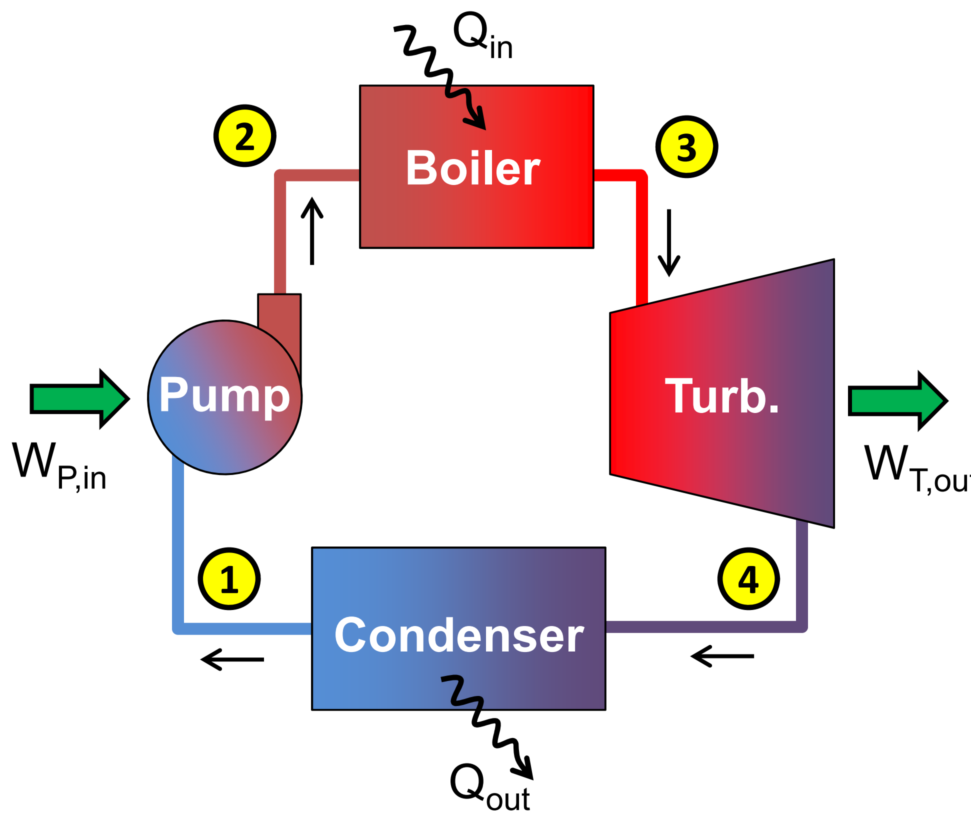
Air = 1.4

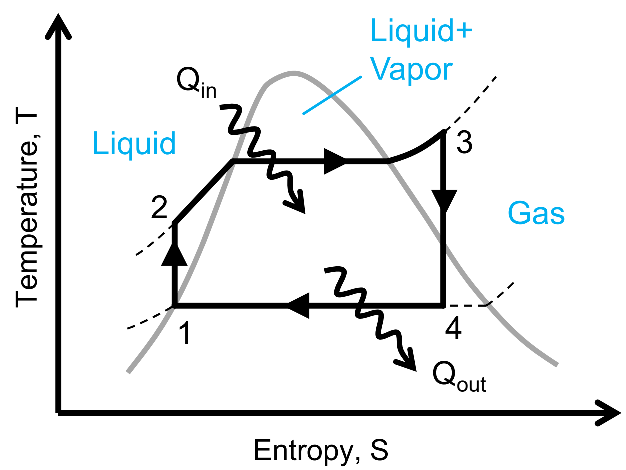
Super Critical Carbon Dioxide Cycles

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
- Phase changes
- E.g., steam cycle

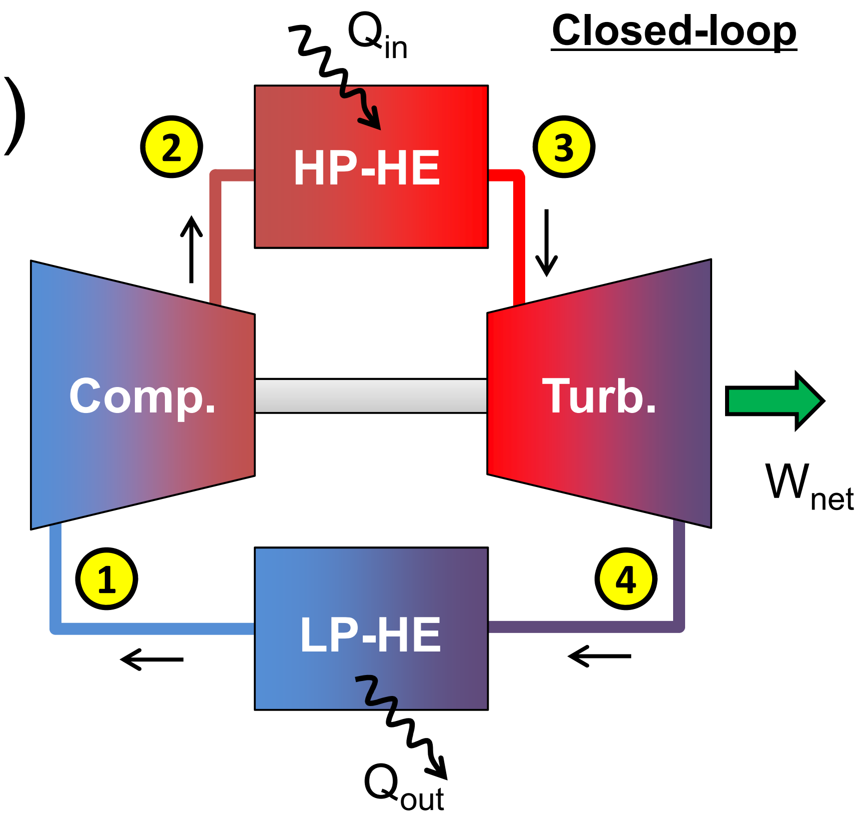
$$\eta_{th} = 1 - Q_{in}/Q_{out}$$





Brayton Cycle (Ideal)

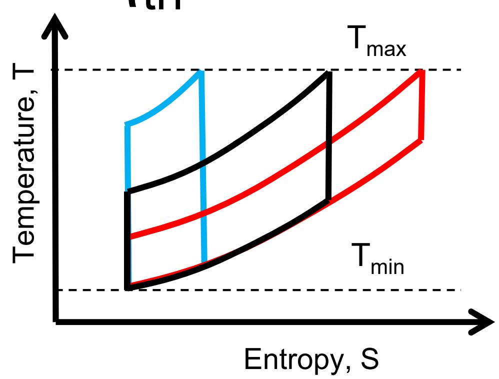
- Processes
  - (1-2) Isentropic compression
  - (2-3) Const. pres. heat addition
  - (3-4) Isentropic expansion
  - (4-1) Const. pres. heat reject.
- Open- or closed-loop

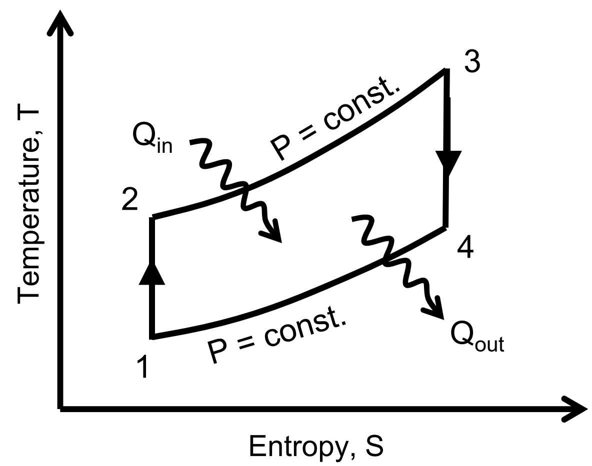


$$\eta_{th,Brayton} = 1 - PR^{(1-k)/k}$$

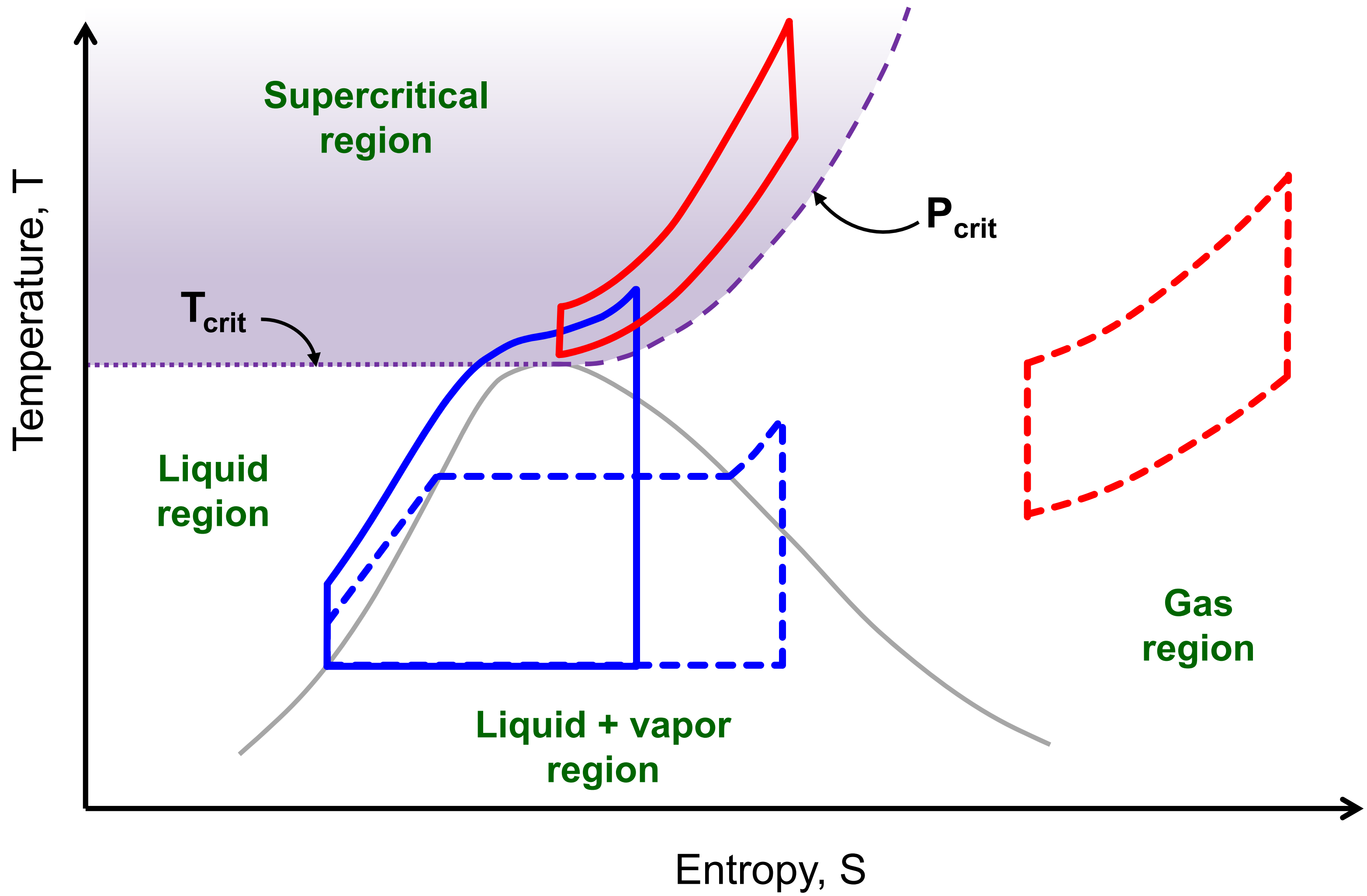
$\uparrow$PR, $\uparrow$k : $\uparrow\eta_{th}$

Optimal PR for net work





What is a Supercritical Power Cycle?



SCO2 Cycles

- Motivation for different configurations
  - Best Efficiency
    - Most Often Considered
    - Cost of Heat Source, Fuel, Solar Mirrors, etc.
  - Best $/kW
    - Capital Cost Is Most Important
    - More Difficult to Optimize
  - Best $/kw-hr
    - Intermittent Heat Source
    - Variable Duty Heat Source
  - Compromising Cycle Efficiency for other reasons
    - Materials
    - Seals
    - Turbomachinery Issues
    - Controllability (Ship Propulsion and others)
    - Building Scaled Test Machines

Supercritical CO2 Cycles Historical Perspective

- Sulzer, 1948, Power Plant
- Feher 1962, 1967, Space Power
- Angelino 1967-1969, Nuclear Power
- Petr, et al. 1997-1999, Nuclear Power
- Kato, et al. 2001, Nuclear Power
- U. S. DOE Nuclear, 2000-present
  - MIT
  - INEEL
  - Argonne National Laboratory
  - Sandia National Laboratory
- Echogen, present, Power Plant WHR
- NetPower Oxy-Fuel
- Sunshot (1 MW scale test loop, 10 MWe turbine) 2013-2017
- Apollo (3 MW SCO2 compressor) 2018-2020
- Supercritical Transformational Electric Power (STEP) 2017-Pres
- Others…

Initial Cycle Schemes Dominated by Nuclear Power
Same Cycles are Beneficial for Solar, Fossil etc.
Bottoming Cycles Tend to Be Unique

“Compound” Brayton Cycle Examples

“Dual Recompression Brayton”

Dostal, 2004

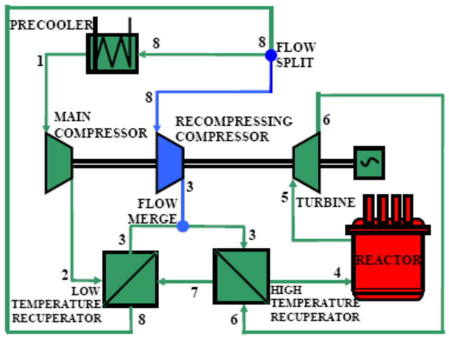


Figure 6.1 Recompression Brayton cycle layout

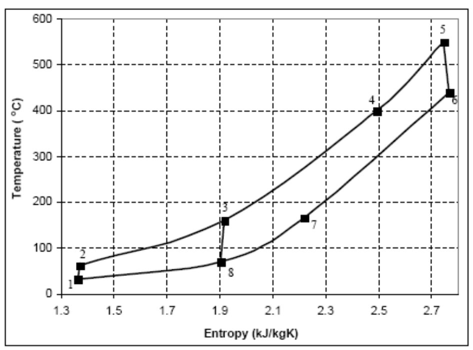
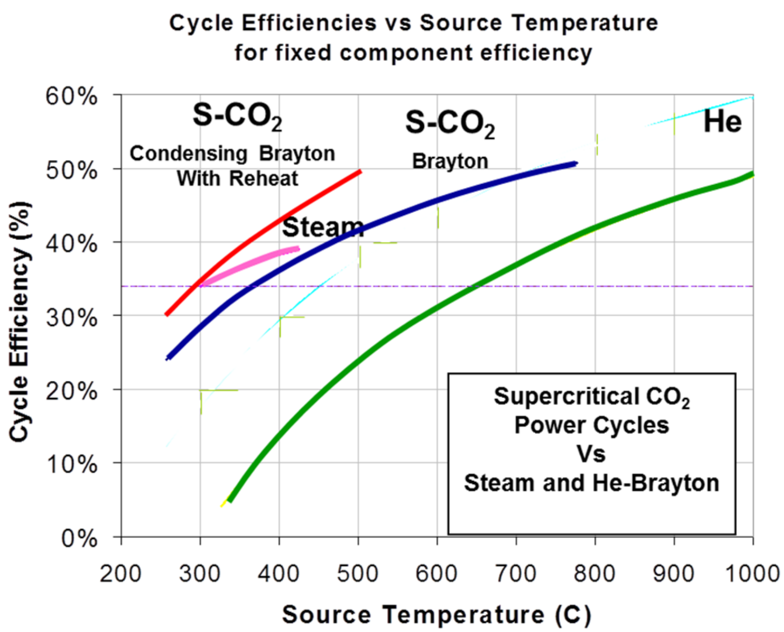


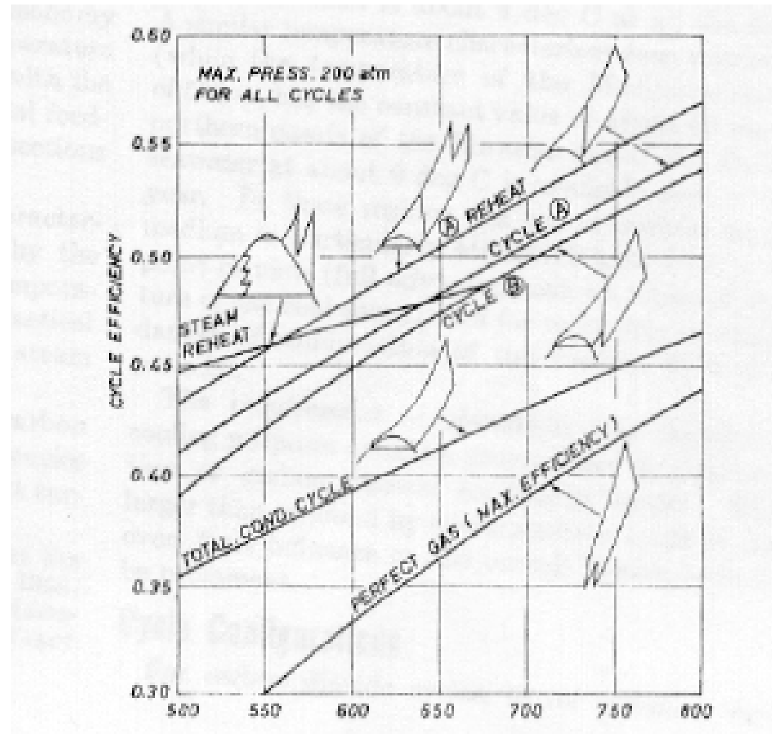
Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle

Cycle Used for Sandia National Laboratories Scaled Test Loop

Comparison of SCO2 Cycles with Alternatives



Wright, et. al. 2010



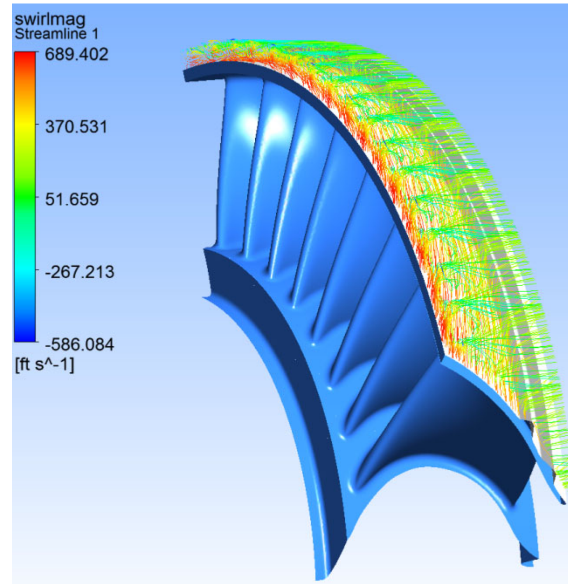
Angelino, 1969

Seals: Internal and Shaft End

Seals

Labyrinth:

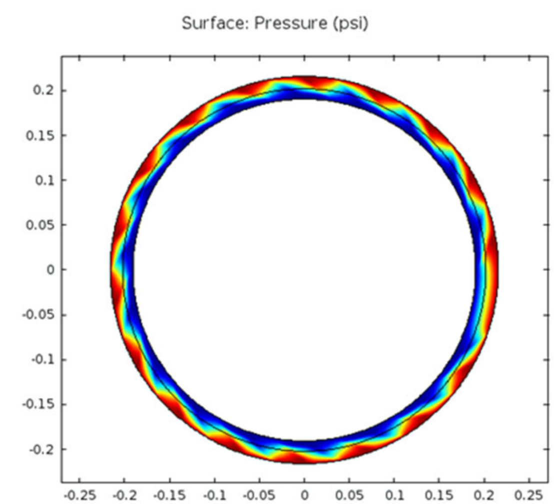
- Labyrinth seals used at blade tip and interstage locations
- Swirl brakes used to minimize swirl entering seal
  - CFD used to optimize and evaluate swirl brake performance



CFD analysis of interstage laby seal flow in CO_2

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



DGS Face Pressure Distribution from CFD

SwRI

Annular Gas Seals in Compressor

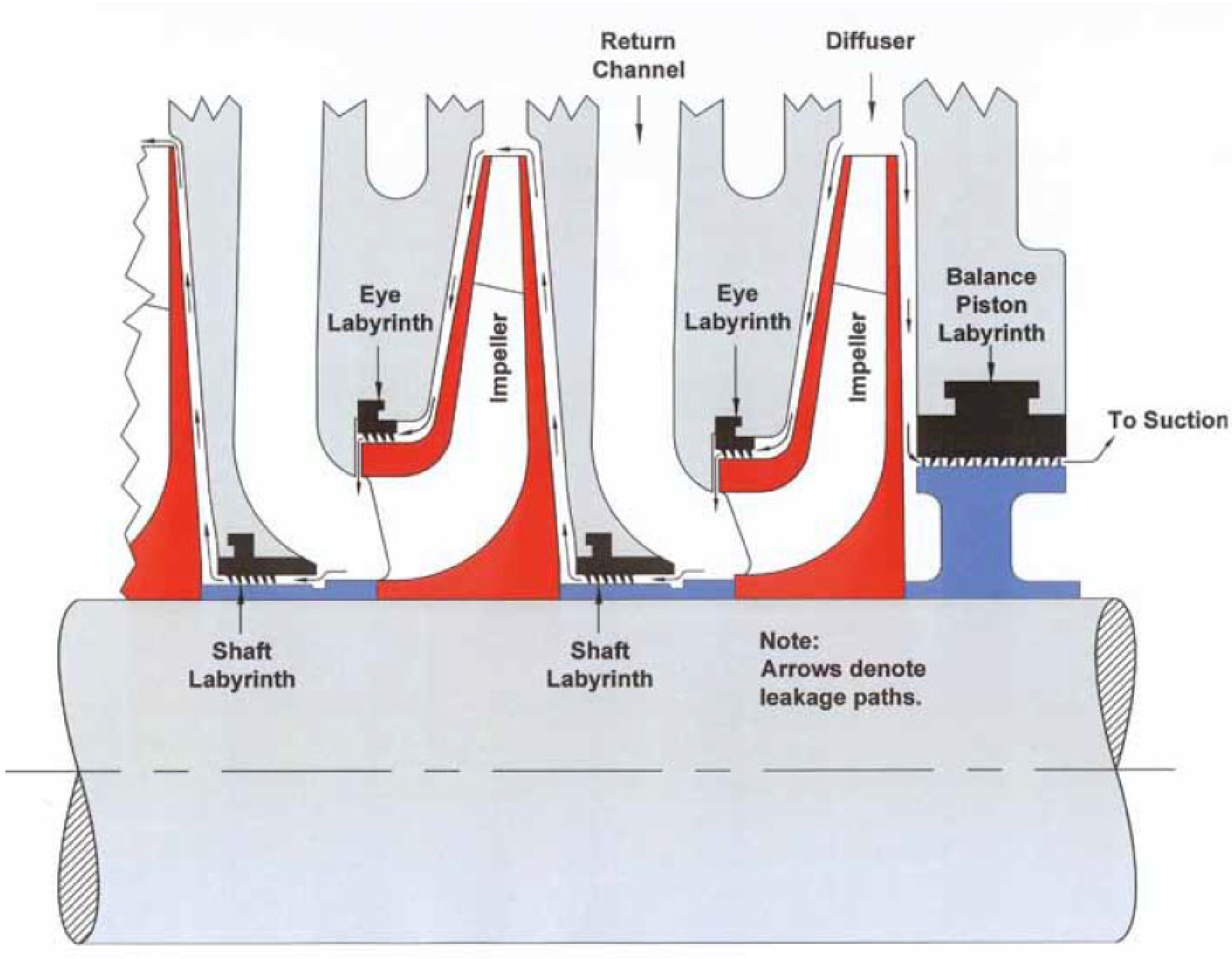
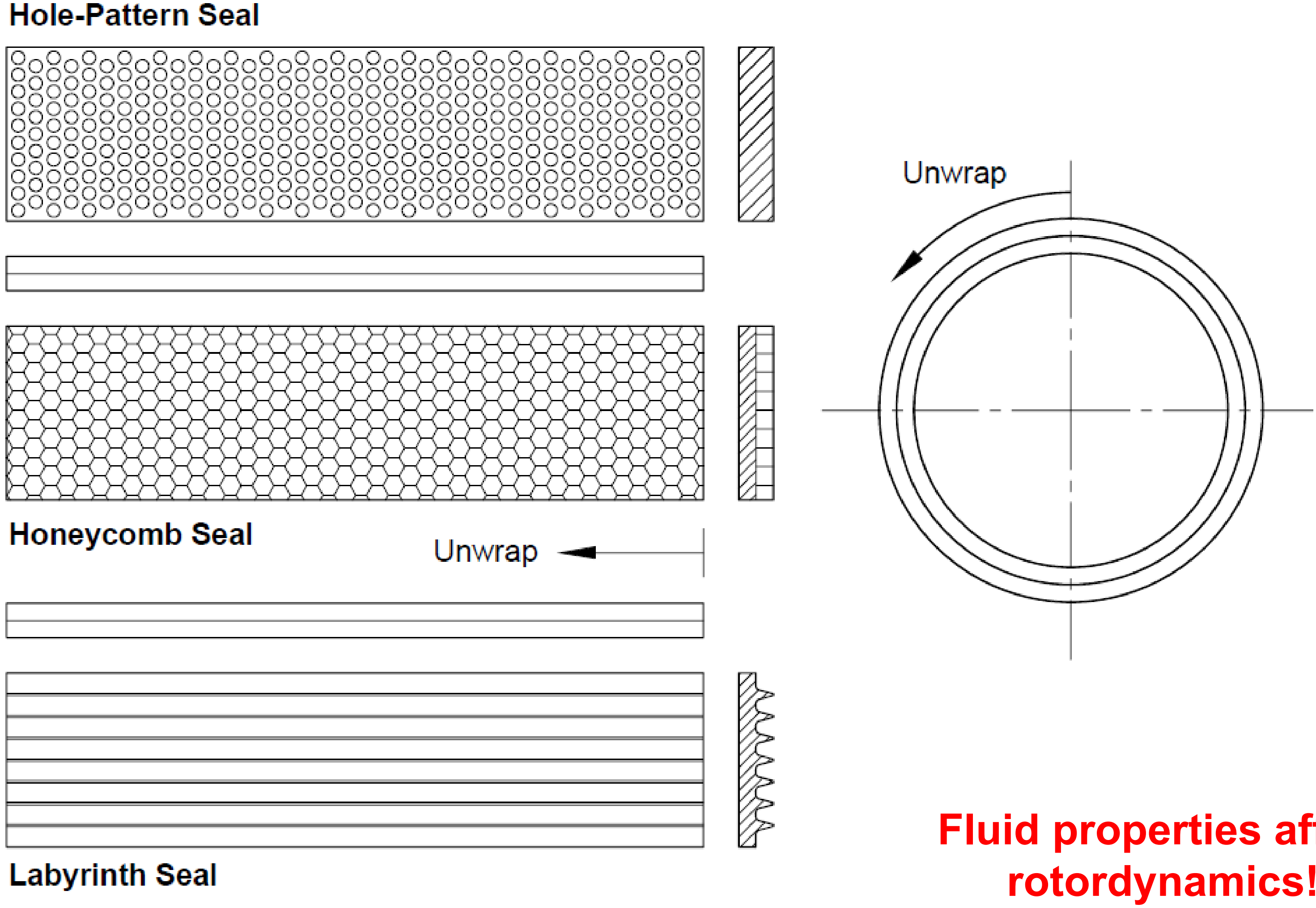


Image source [7-1]

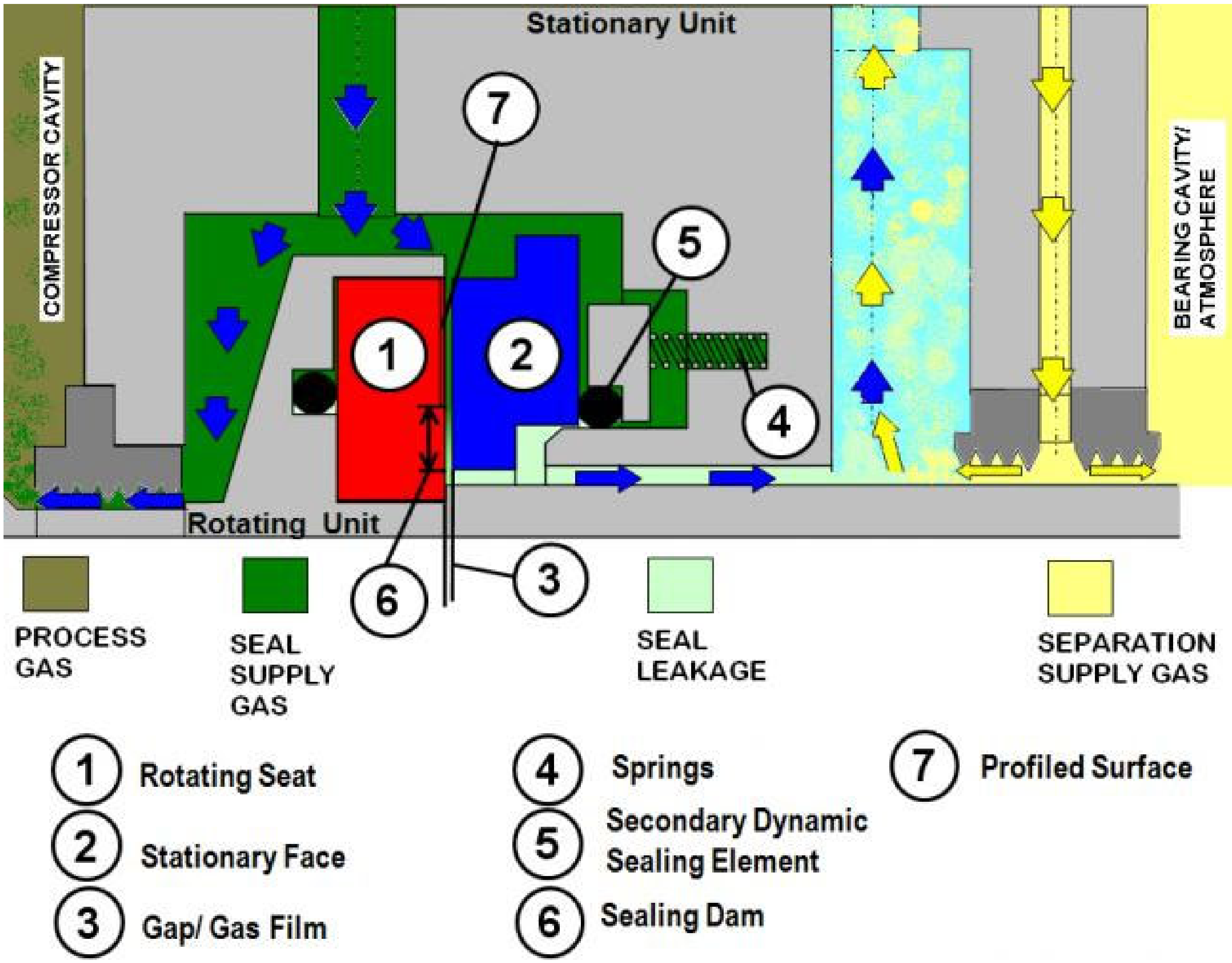
Different Seal Geometries



Fluid properties affect rotordynamics!

Image source [7-2]

Dry Gas Seals



Rotating seal surface…

Image source [7-4]

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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

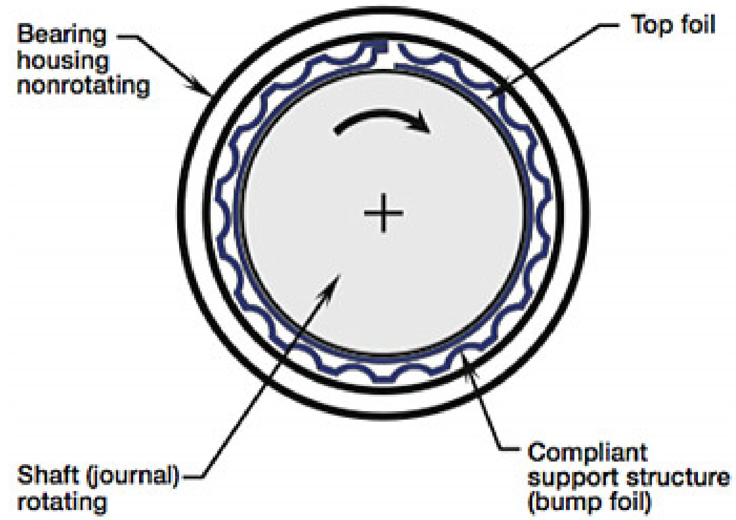
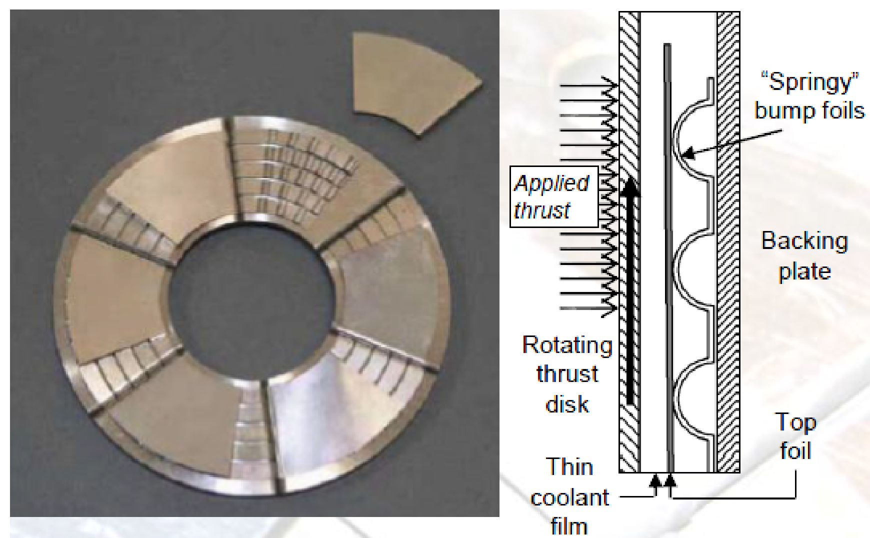


Image source [7-5]



Source: Milone (2011)

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)

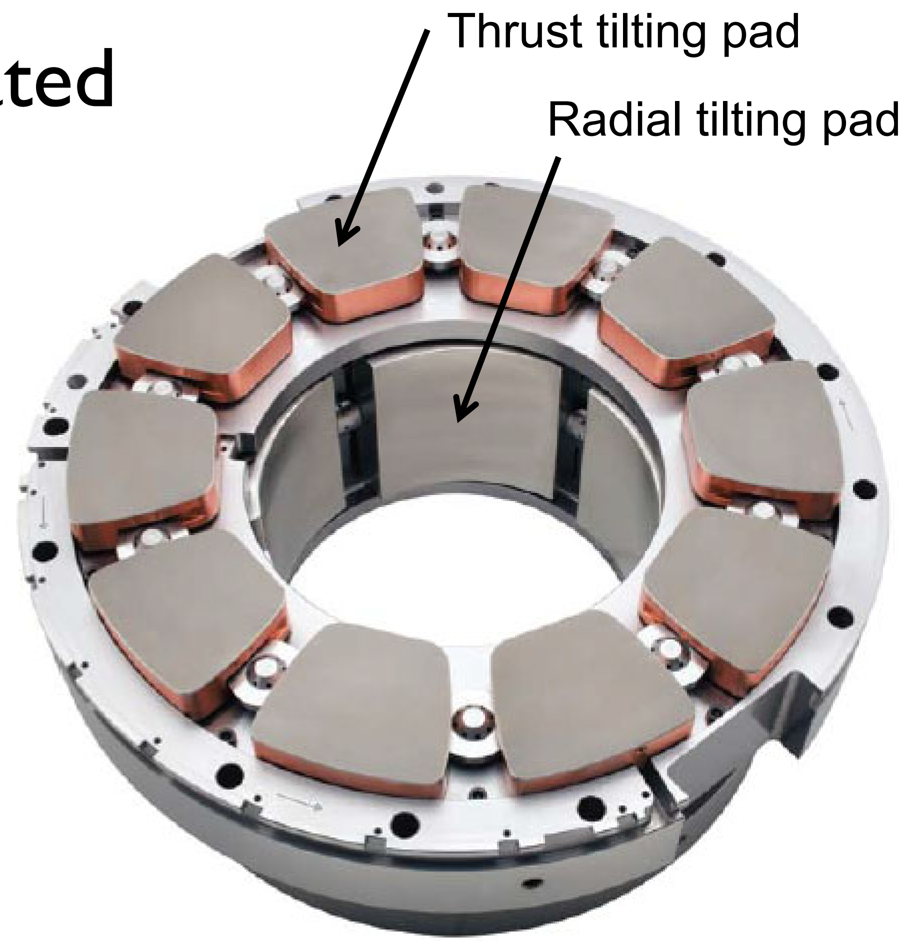


Image source [7-6]

Rotordynamics

Rotordynamics

Challenges

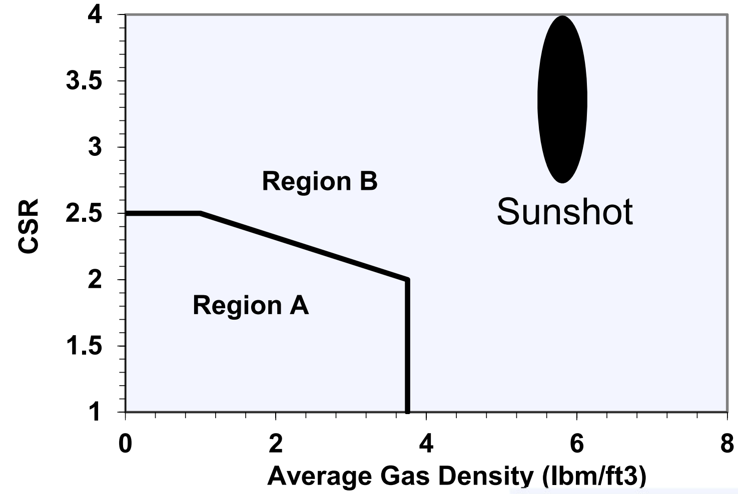
- High gas density
- High operating speed
- Low critical speed (large L/D)
- Similar design methodology as high pressure turbocompressors

Interstage laby seals

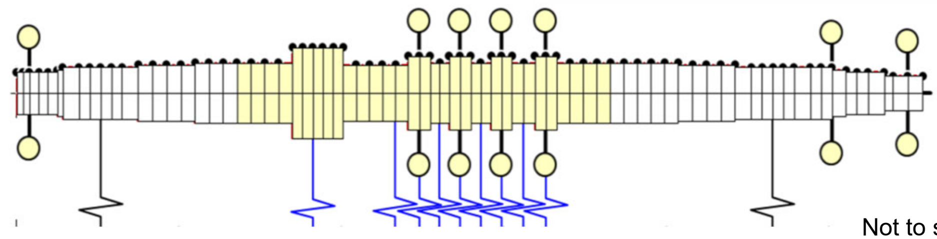
- Texas A&M XLTRC code
- Real gas CO2 properties

Balance piston seal

- Texas A&M code
- Perfect gas properties



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



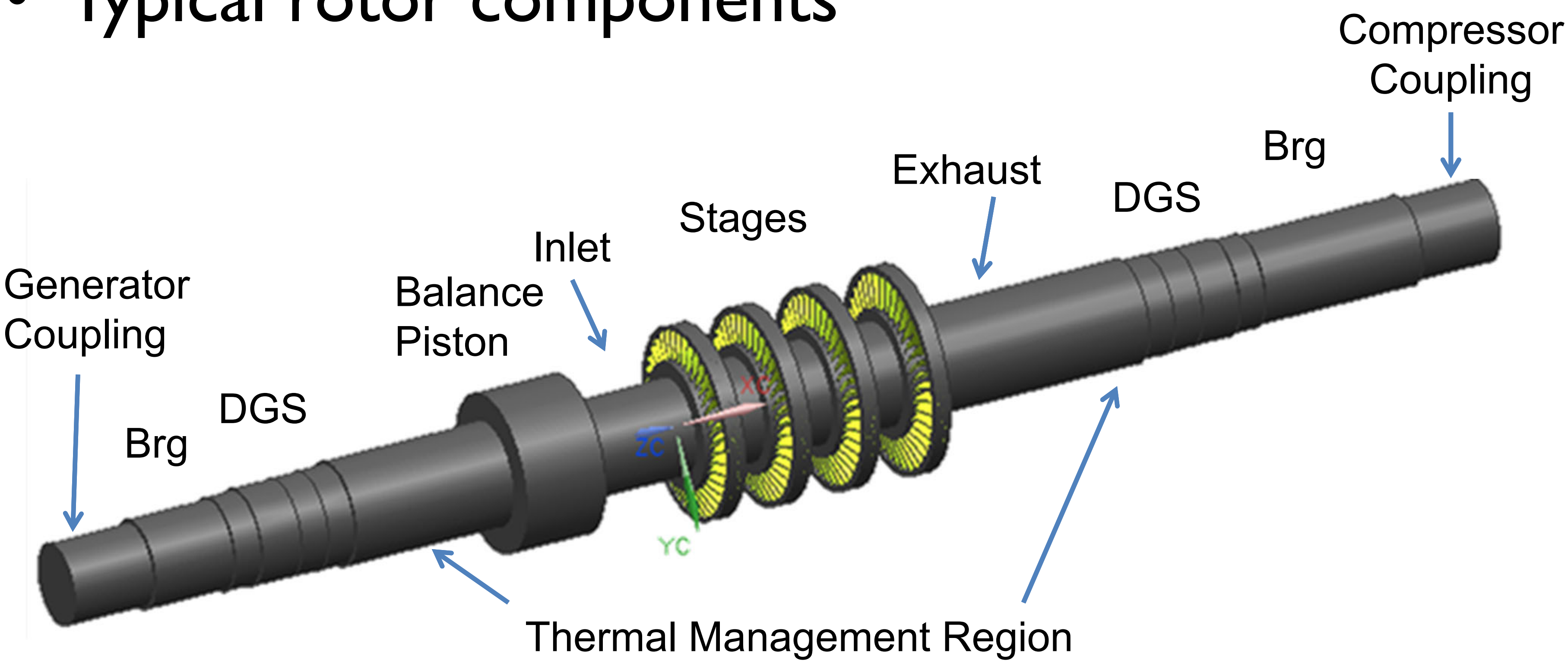
Not to scale

Result

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

SCO2 Turbine Rotor Features

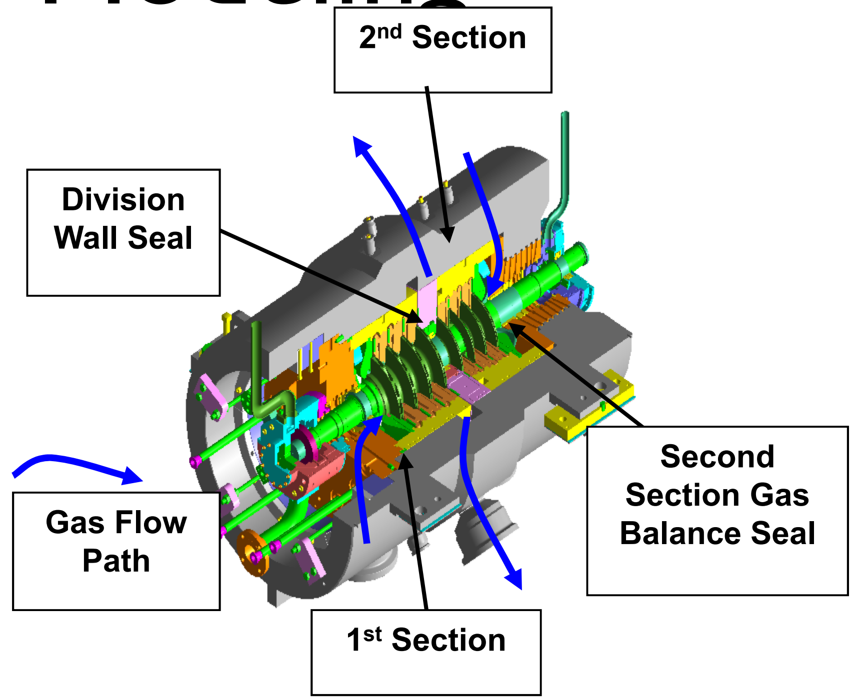
- Typical rotor components



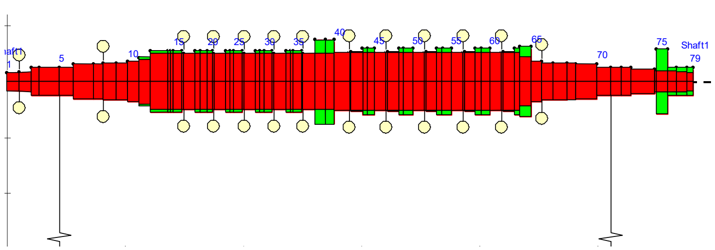
Rotordynamic Modeling

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

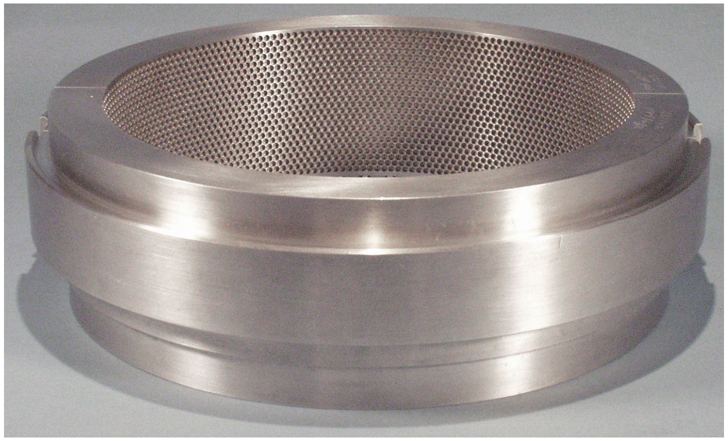
Reference: Moore, J.J., Soulas, T.S., 2003, "Damper Seal Comparison in a High-Pressure Re-Injection Centrifugal Compressor During Full-Load, Full-Pressure Factory Testing Using Direct Rotordynamic Stability Measurement," Proceedings of the DETC '03 ASME 2003 Design Engineering Technical Conference, Chicago,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

SwRI

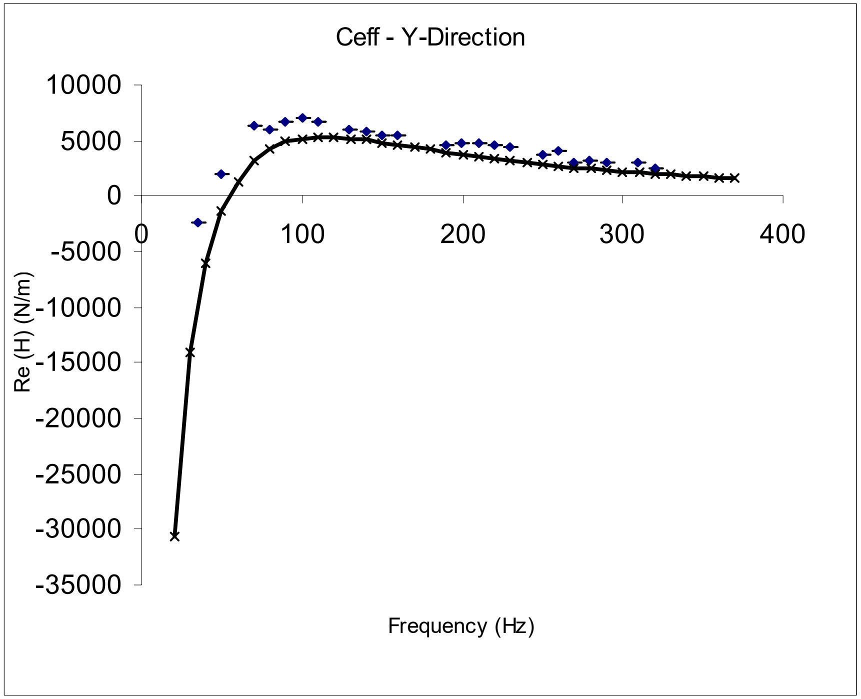
Rotordynamic Modeling

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

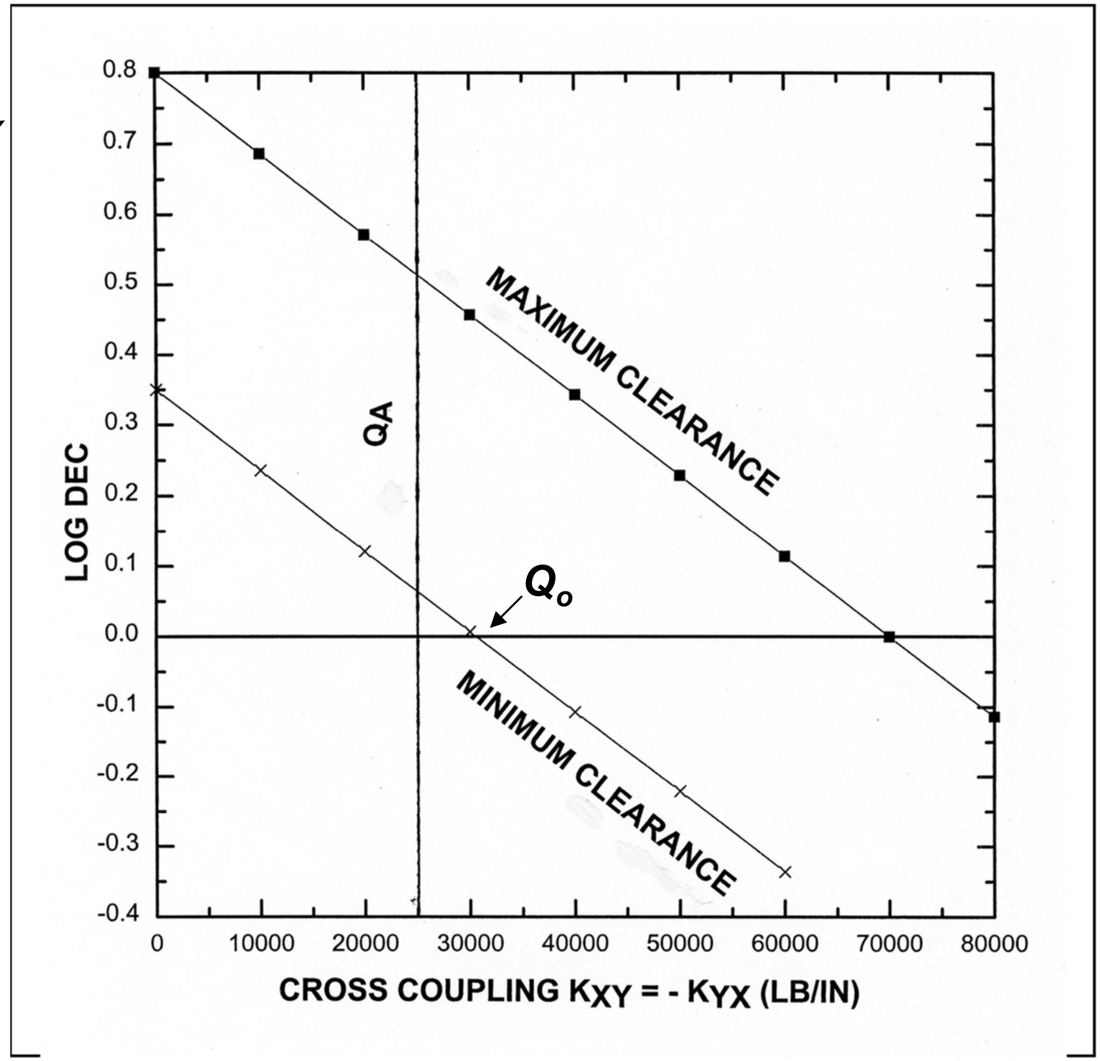


Reference: Camatti, M., Vannini, G., Fulton, J.W., Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," *Proc. of the Thirty-Second Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station,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

API 617 Requirements

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

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



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

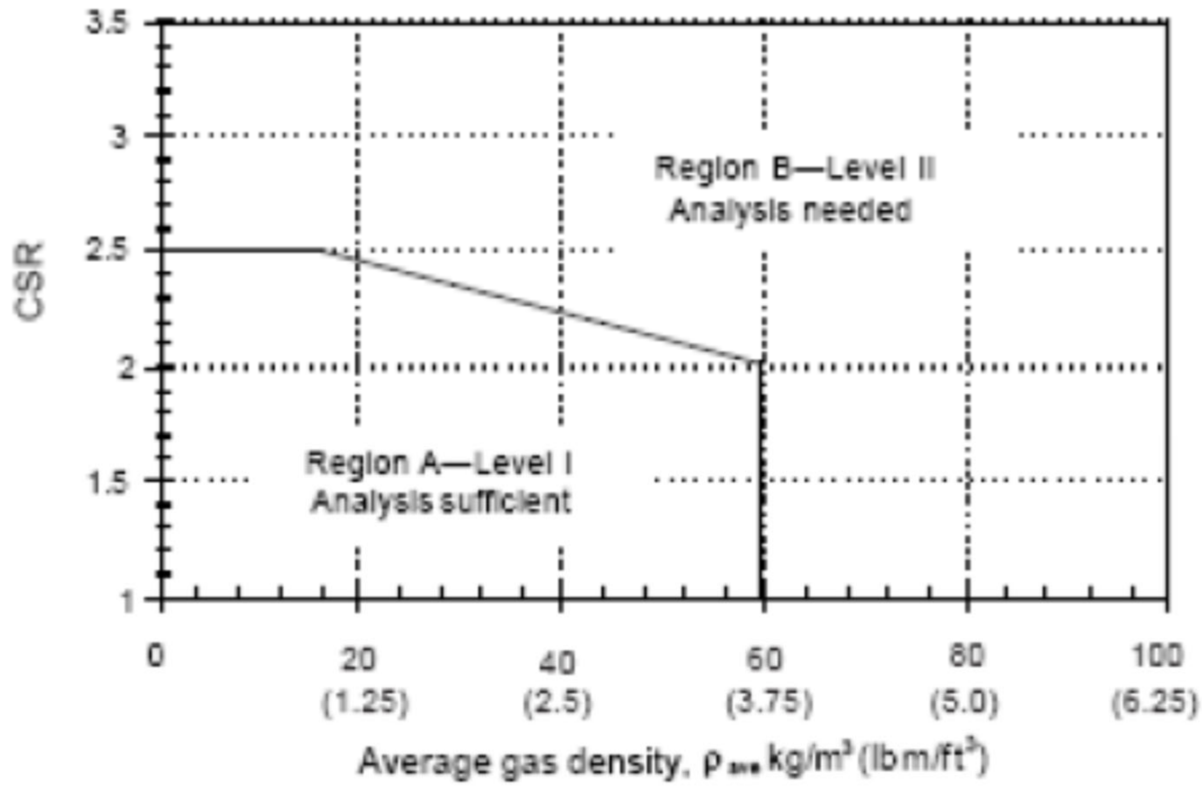


Figure 1.2-5—Level I Screening Criteria

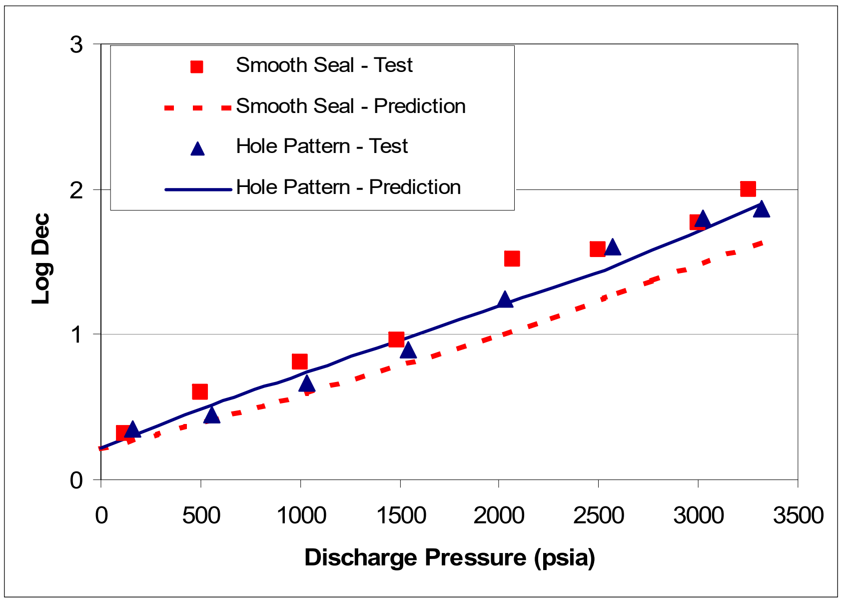
API 617 Requirements

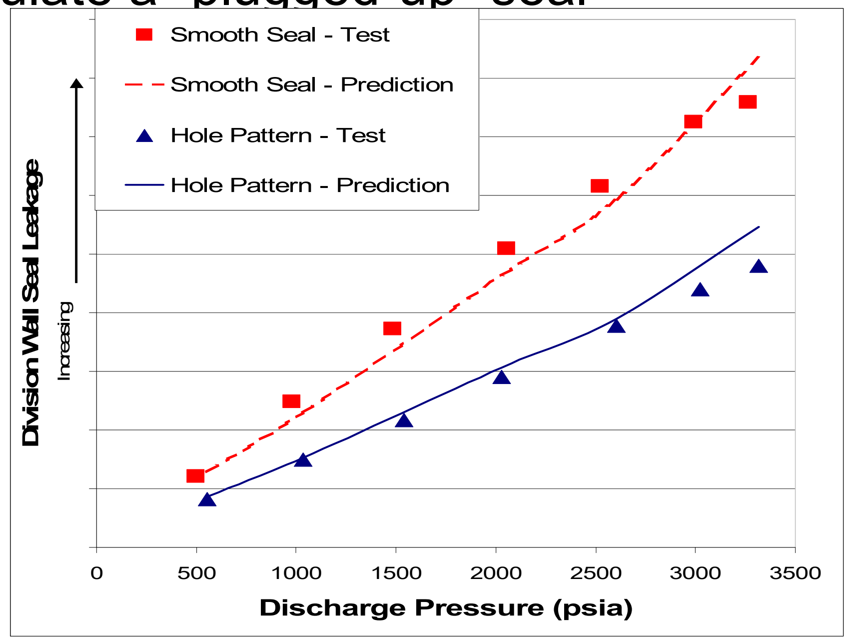
- If any of the following is not met, then a Level 2 analysis is required
  - SM < 2.0
  - $\delta_A < 0.1$
  - 2.0 < SM < 10 if in Region B
- Level 2 Analysis includes the effect of:
  - All labyrinth/damper seals
  - Balance piston seals
  - Impeller/blade row (some believe that only labyrinths are 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

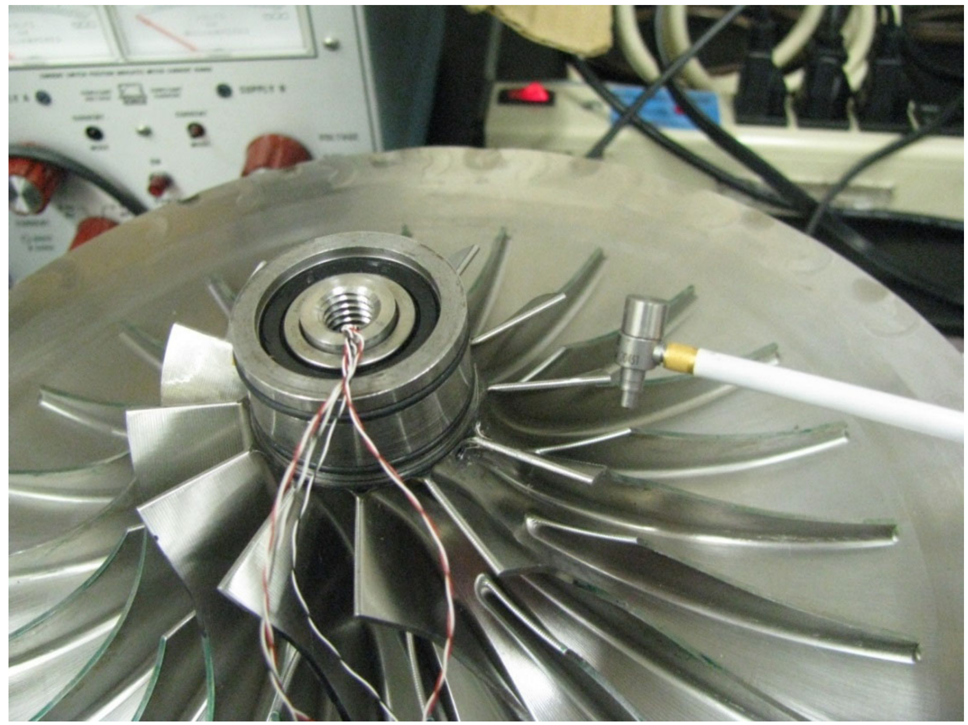
Blade Dynamics

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

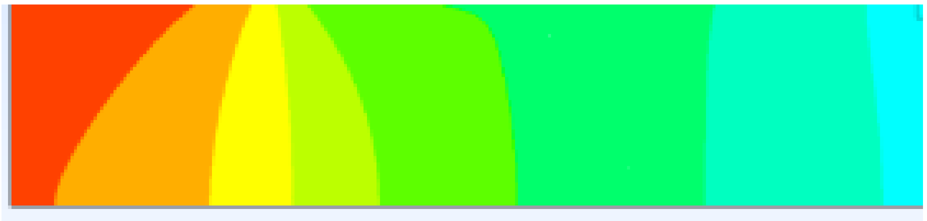
- Modal testing used to validate design
- Effect of gas density and temperature dependent material properties must be considered



Thermal Management

Thermal Management

- Temperature between hot inlet (up to 700C) and dry gas seal (~100C) 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



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

Packaging Considerations

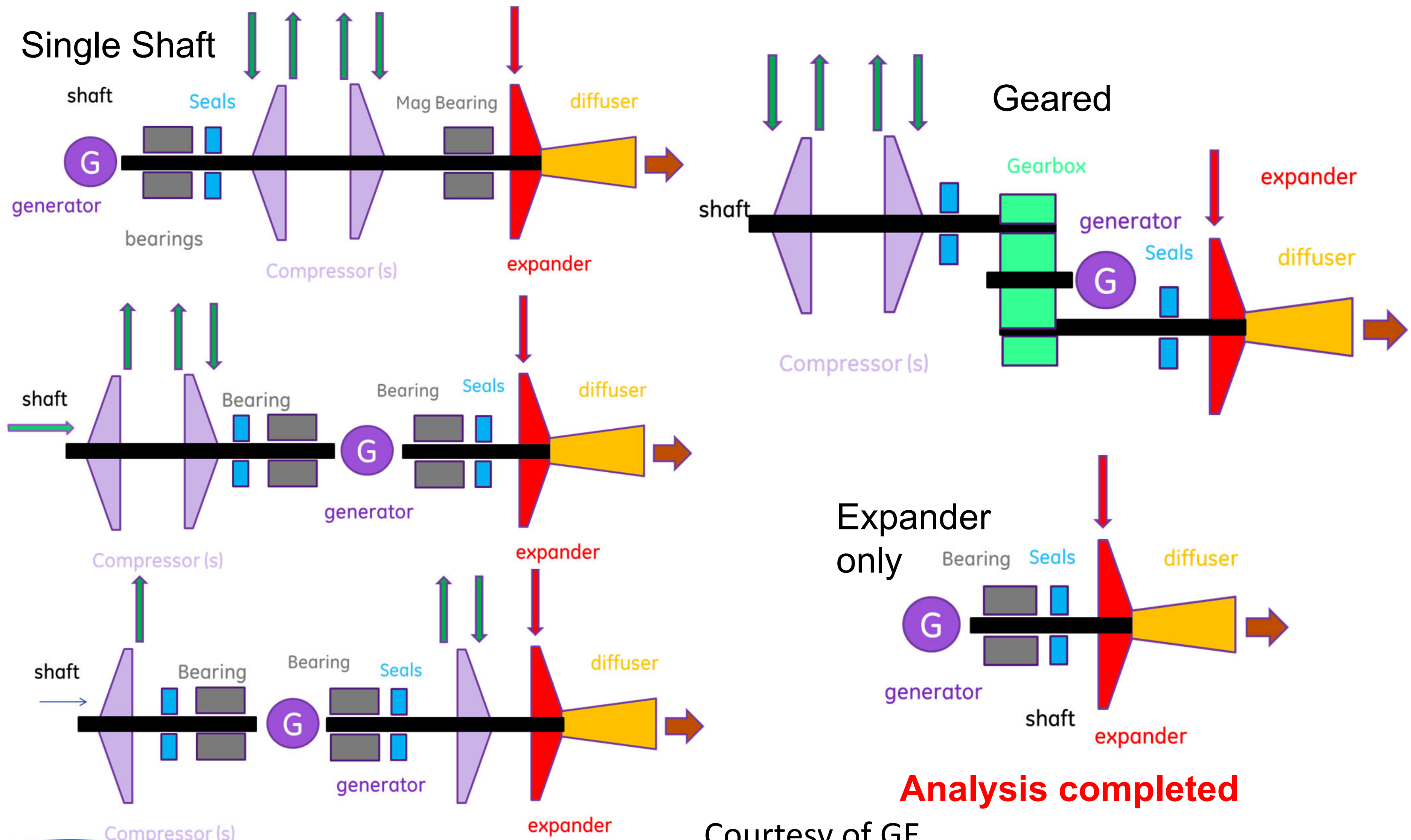
Turbomachinery Concepts

- Concepts explored include high-speed, low-speed, and geared layouts.

Option	Generator	Compressor	Turbine	RPM
High-speed, Optimal	A. IC B. PM	A. Single-stage centrifugal B. Multi-stage pump	A. Radial B. Axial	Optimized for compressor
High-speed, expander only	A. IC B. PM	None	A. Radial B. Axial	Optimized for expander
High speed, Geared	A. IC B. PM C. 3,600 rpm	A. Single-stage centrifugal B. Multi-stage pump	A. Radial B. Axial	Both expander and compressor run at optimal speed
3,600 rpm – integrated	3,600 rpm	Multi-stage pump or compressor	Multi-stage Axial at 3,600 rpm	3,600 rpm
3,600 rpm – expander only	3,600 rpm	None	Multi-stage Axial at 3,600 rpm	3,600 rpm

Courtesy of GE

Turbomachinery Concepts



Analysis completed

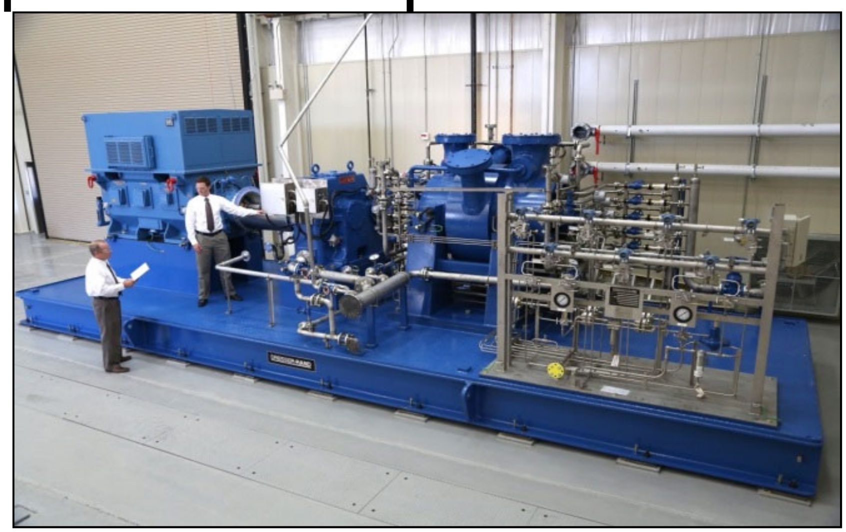
Courtesy of GE

Generator Design

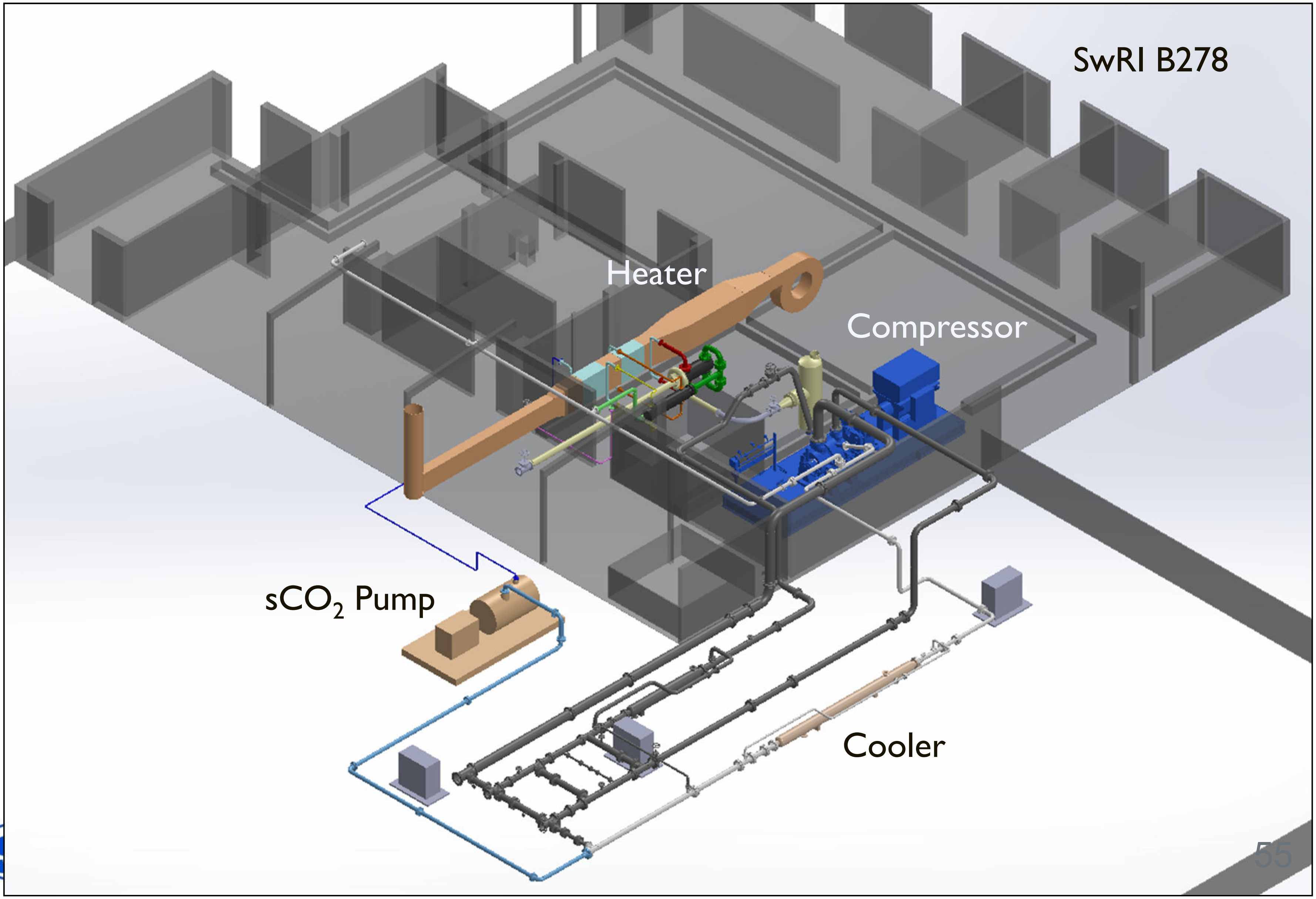
- Low-speed Synchronous
  - Low cost
  - Commercially available
  - Need to be combined with gearbox
- High-speed
  - May drive turbomachinery directly or use small gearbox if needed
  - Limited to 20,000 rpm
  - Lighter than low-speed option
  - More expensive

CO2 Compression Loop at SwRI

- 3 MW CO_2 compression under construction
- May be used to perform aerodynamic testing of 10 MW SCO_2 turbine prototype
- High pressure portion of the loop will be used for full pressure-temperature testing of turbine

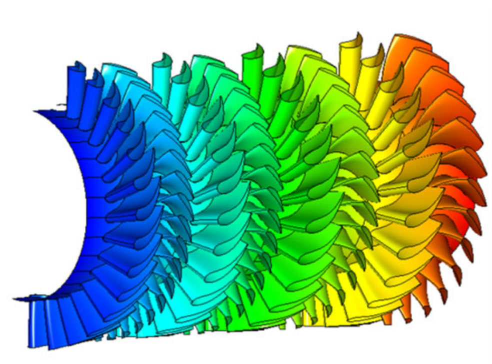


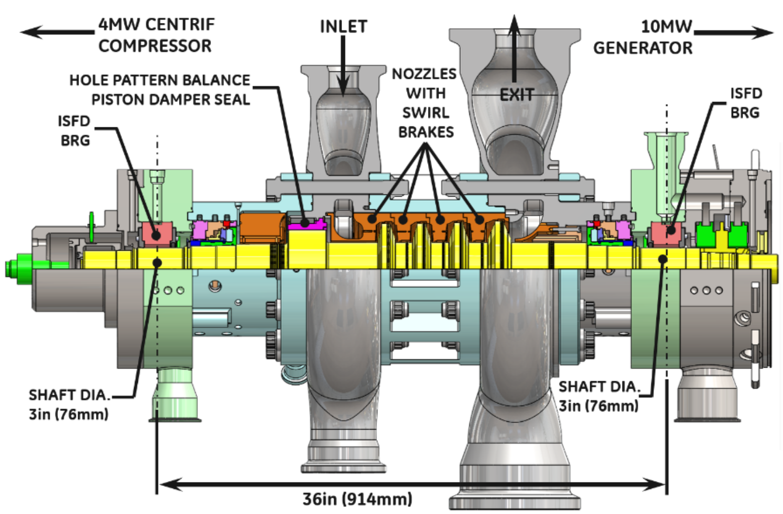
Sunshot: SCO2 Test Loop at SwRI

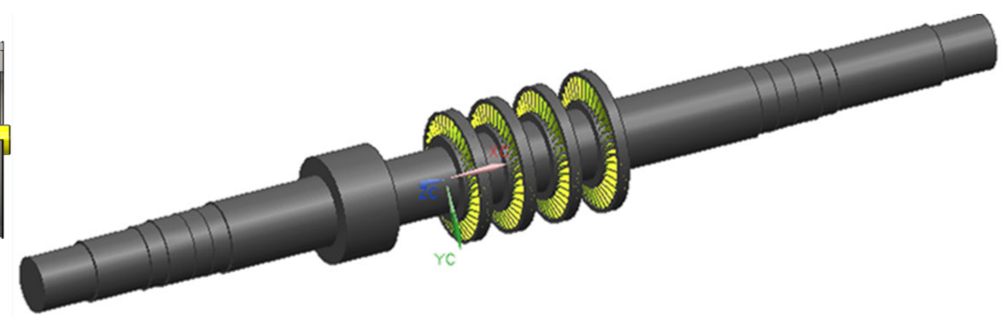


SwRI/GE 10 MWe SCO2 Turbine

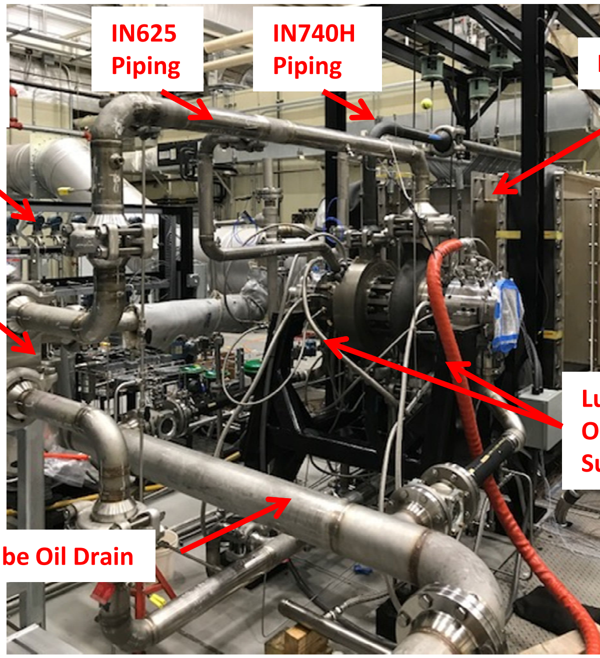
- ~14MW shaft power
- >700C inlet temp
- >85% aero efficiency
- Multi-stage axial
- Dry gas seals
- Fluid-film bearings
- Scalable to 100+ MW utility scale turbine

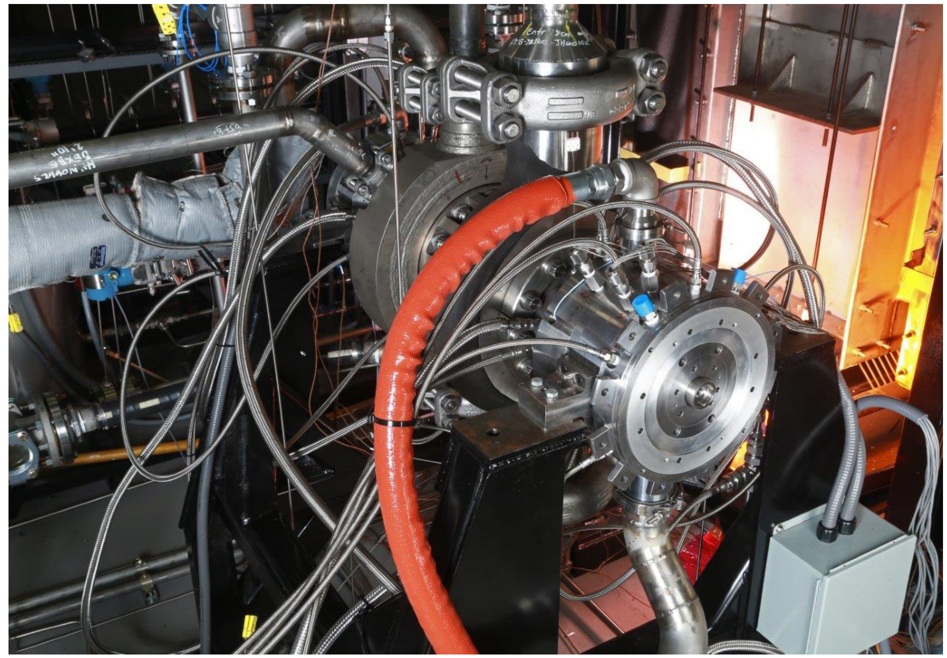






Sunshot Test Loop Components





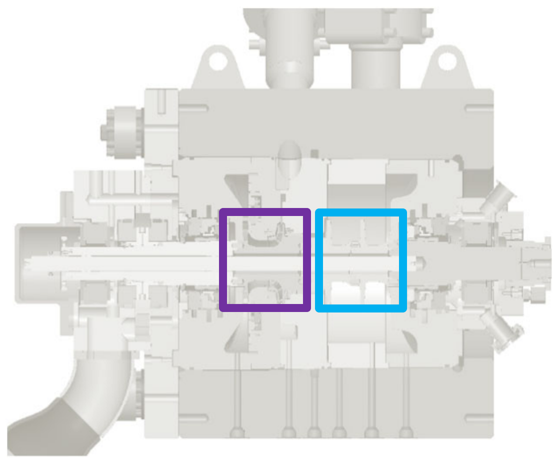
	Speed (rpm)	Turbine Inlet Temp. °C (°F)	Turbine Inlet Pressure bar (psi)	Turbine Exit Pressure bar (psi)
1st Design Point	21,000	550°C (1022°F)	~200 bar (3000 psi)	80 bar (1160 psi)
2nd Design Point	27,000	715°C (1319°F)	~250 bar (3625 psi)	80 bar (1160 psi)

Sunshot Turbine Summary

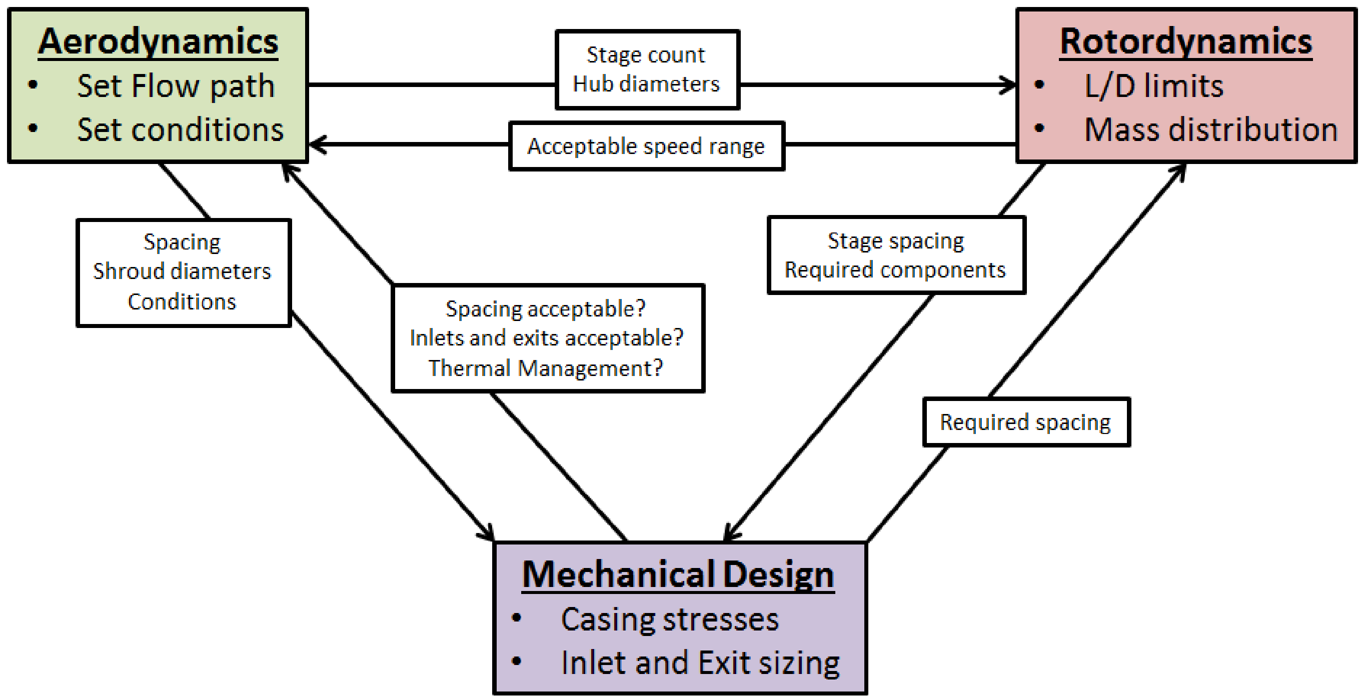
- Turbine performance met mechanical and performance objectives.
  - Achieved design temperature of 715C, design speed of 27000 rpm, and near design pressure of 250 bar.
  - Highest temperature SCO_2 turbine in the literature.
  - Thermal seal maintained acceptable dry gas seal operating temperature with near linear profile.
  - Vibration well less than 0.5 mils with no signs of instability
  - Low critical speed response (good bearing damping and balance)
  - Good thrust balance and low thrust bearing temperature
  - Low radial bearing temperatures following clearance modification
  - Many shutdown transients tolerated
    - Some leakage experienced out case joints due to loss of bolt preload
    - Being addressed with single piece case design with STEP
  - Modified dry gas seal panel maintained warm seal gas preventing dry ice formation

Example SCO2 Compressor

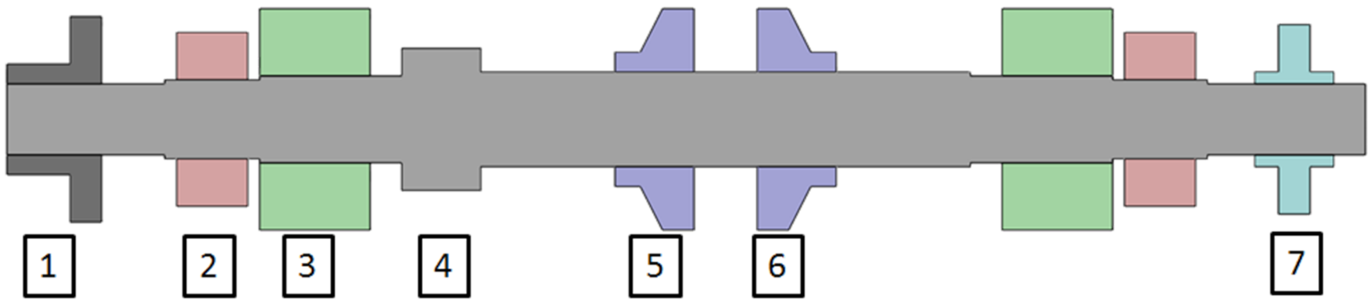
- Design a sCO_2 compressor for use in a closed loop Recompression Brayton Cycle (RCBC) to achieve thermal efficiencies > 50% with both **MAIN** and **BYPASS** compressor
- Compressor must handle suction temperatures from 68 to 122^{o}C and pressures from 1,090 to 1,540 psia to maintain near steady mass-flow to turbine
- High-pressure, high-density, large swings in volume flow, high-speed, packaging



Example SCO2 Compressor



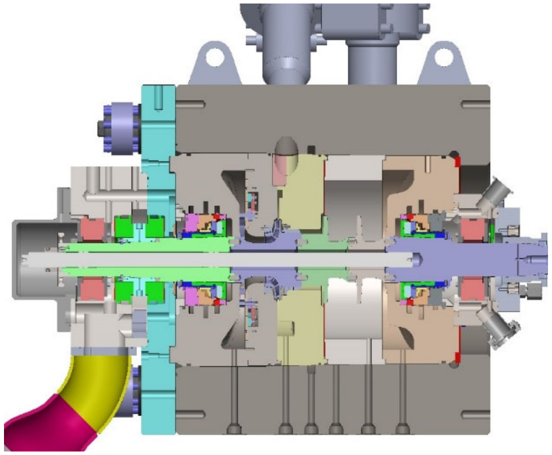
Example SCO2 Compressor



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

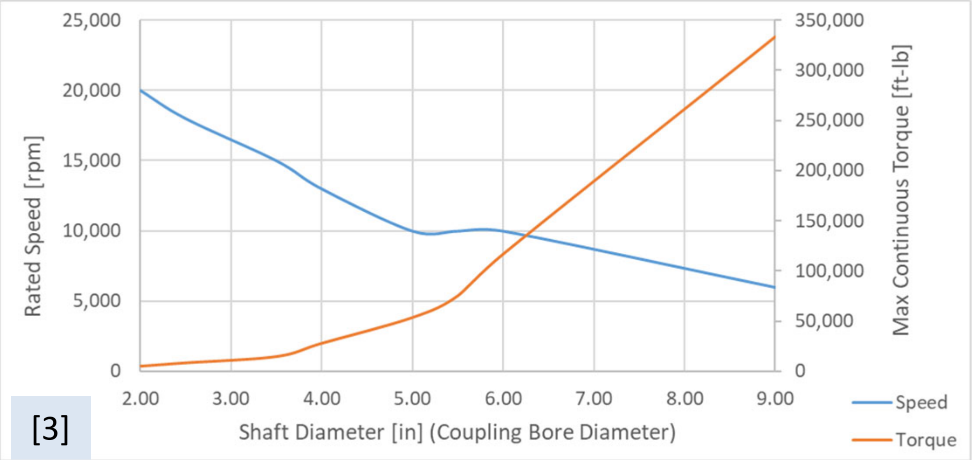
Operating Conditions

- Power: 6,570 hp
- Discharge Pressure: 4,850 psia
- Max Temperature: 400^{o}F
- Max Speed: 28,350 rpm (match turbine) [5% over Nominal]
- 2X swing in volume flow
- Suction Densities from 29 to 51 lbm/ft^3



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

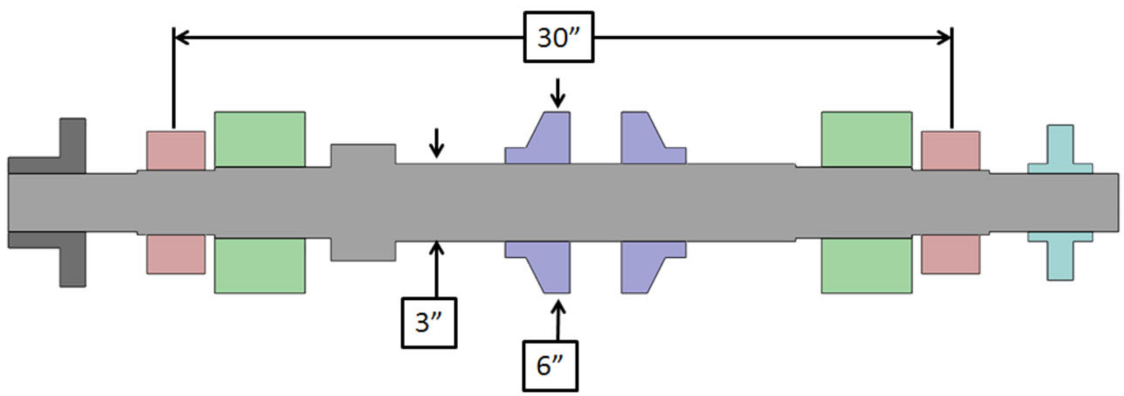


[3]

$T = 63{,}025\frac{P}{w} = 63{,}025\frac{6570}{27000} = 15{,}336$	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
$J = \frac{\pi r^4}{2} = 2.52\ in^4$	

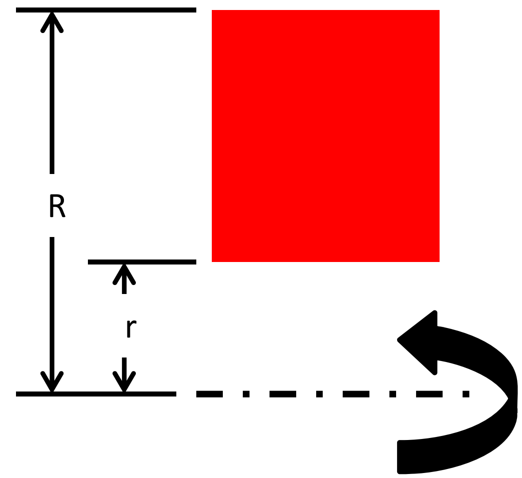
Mechanical Design

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



Mechanical Design

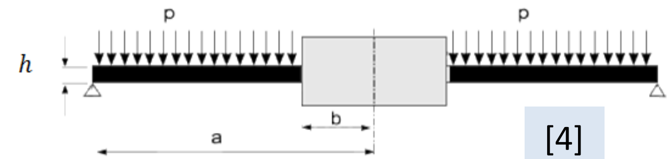
- Larger diameters → More head, more rotating weight, higher hub stress
- Smaller Inner Diameters → reduced tie bolt diameter, more rotating weight, reduced stresses
- Solid shafts have the lowest stresses but required monolithic shafts or more complicated joints



$\sigma = \frac{3+v}{8}\rho\omega^2\left[r^2 + 2R^2 - \frac{1+3v}{3+v}r^2\right]$	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

- 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

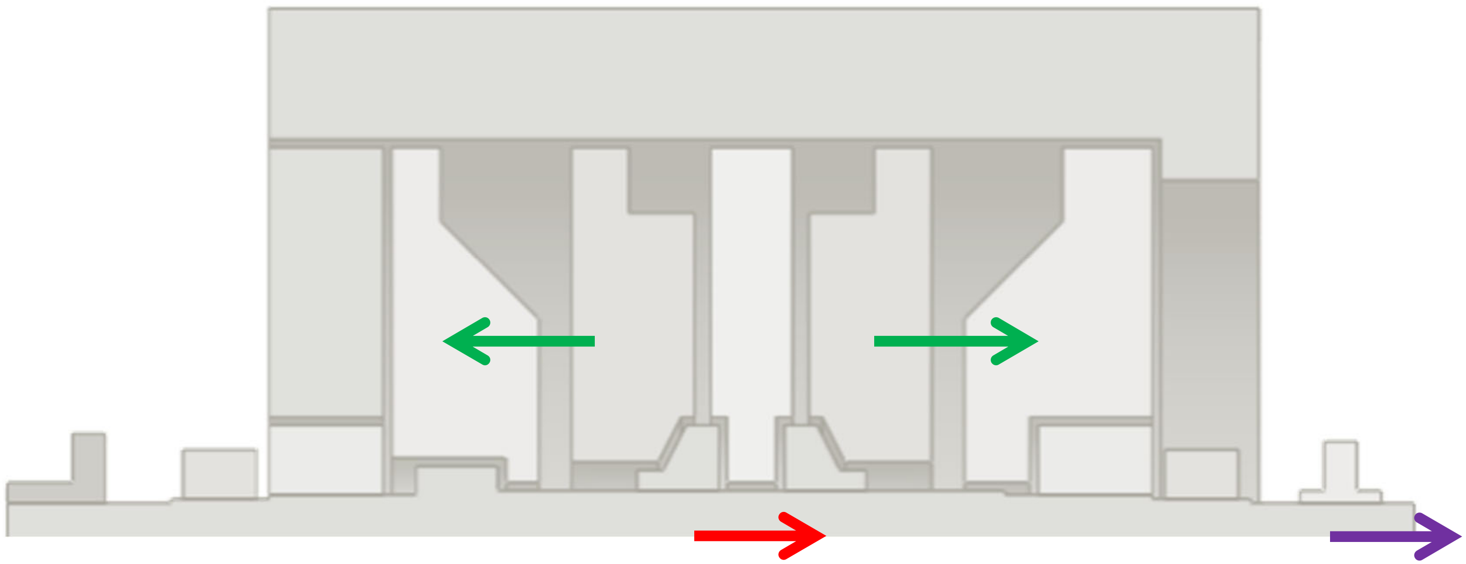


[4]

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

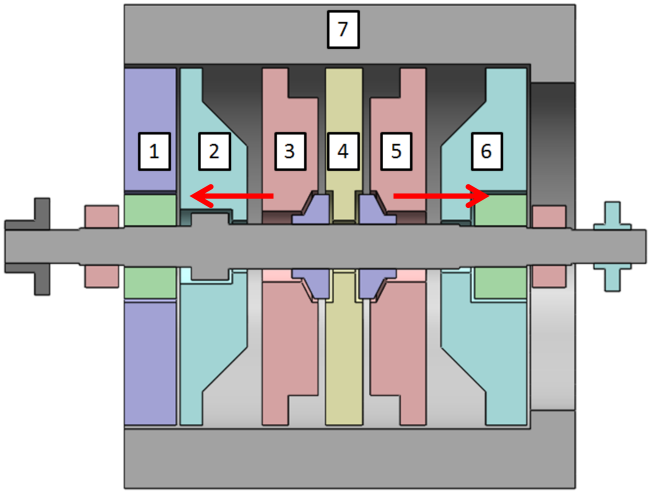
Housings and Heads Deflect Away from High Pressure – Opening of Diffuser

Housing Deflection Pulls Rotor at Thrust Bearing

Impeller Move to the Right

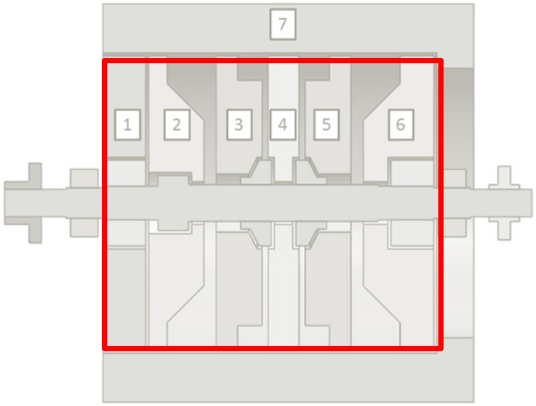
Mechanical Design

- 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 sea
    - Pilot fits



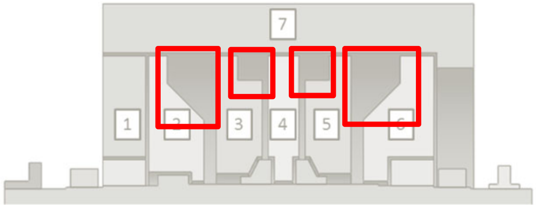
Mechanical Design

- 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

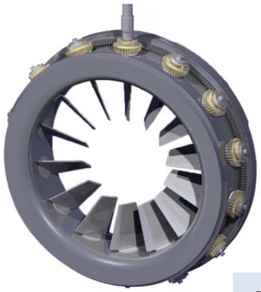


- Flow path Sizing
- Lower velocity limits → larger flow areas → Increase bundle diameter (or case complexity) → reduced pressure losses
- Higher velocity limits → smaller flow areas → Reduced bundle diameter → erosion concerns & increased pressure loss

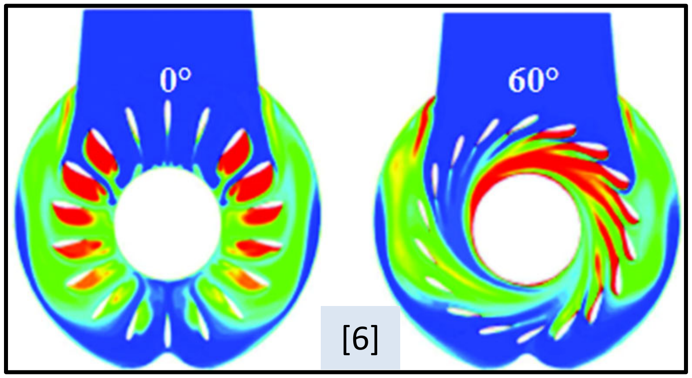
Section		Pressure		Temperature		Density	Mass Flow		Max Vel.	Min Dia.
		Mpa	psi	C	F	lbm/in^3	kg/s	lbm/s	ft/s	in
Main	Inlet	8.55	1,240	35	95	0.0224	70.3	155.0	80	3.03
	Exit	24.13	3,500	78	172	0.0247	70.3	155.0	80	2.88
Bypass	Inlet	8.69	1,260	88	190	0.0061	34.2	75.4	80	4.04
	Exit	23.99	3,479	194	381	0.0116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

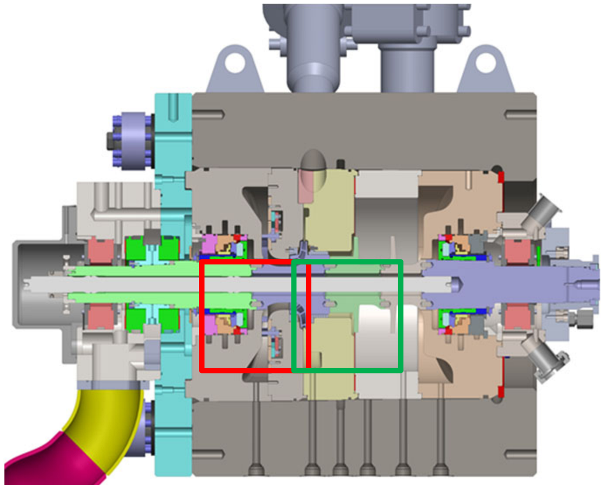


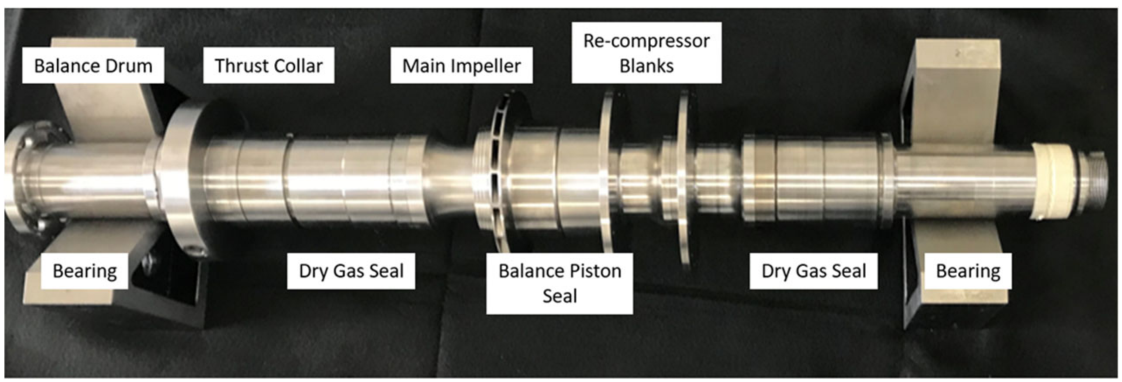
[6]

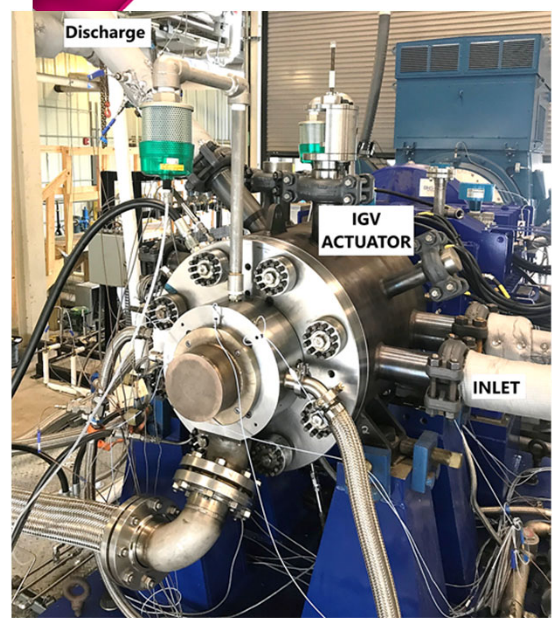
Final Embodiment – SwRI/GE Apollo Compressor

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

- Highest density compressor in literature (720 kg/m3)







Supercritical CO2 Cycles Pumps/Compressors/Turbines Aero Design

Turbomachinery Elements for Super Critical CO2

- 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

Turbomachinery Attributes for Super Critical CO2

- Extremely Compact and Power Dense
- Relatively Low Peripheral Speeds
- High Blade Loading
- High Shaft Speeds for
- Difficult to Balance or Accommodate Thrust

Turbomachinery Elements-Main Compressor

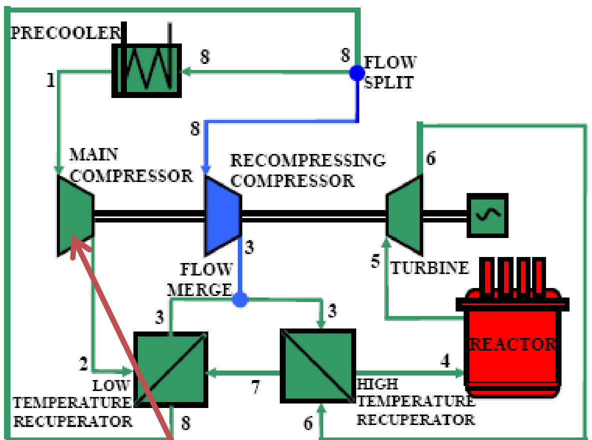
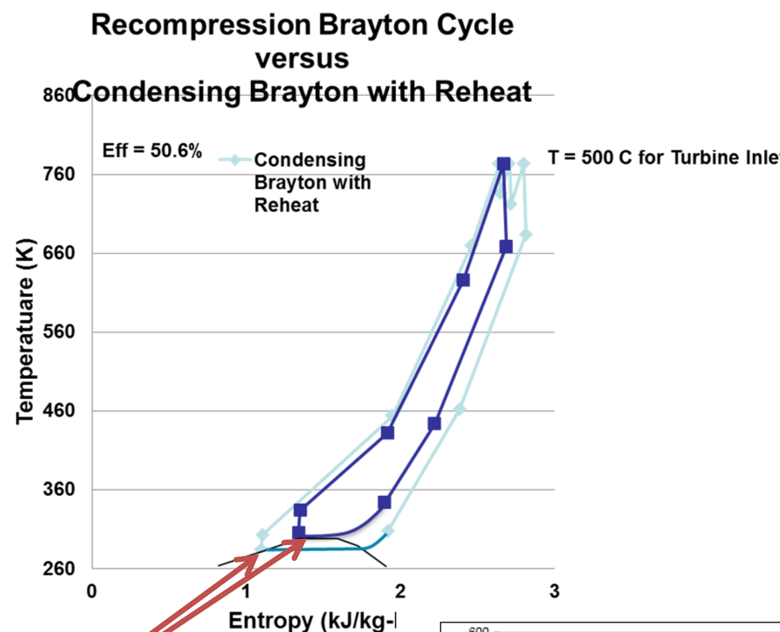


Figure 6.1 Recompression Brayton cycle layout



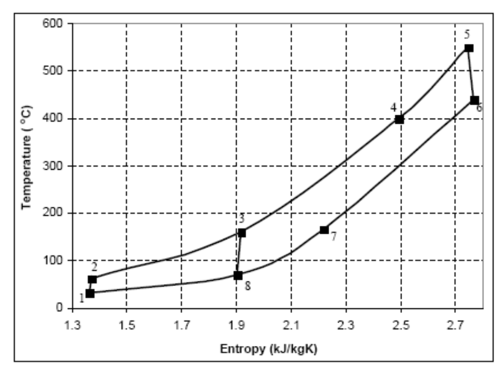
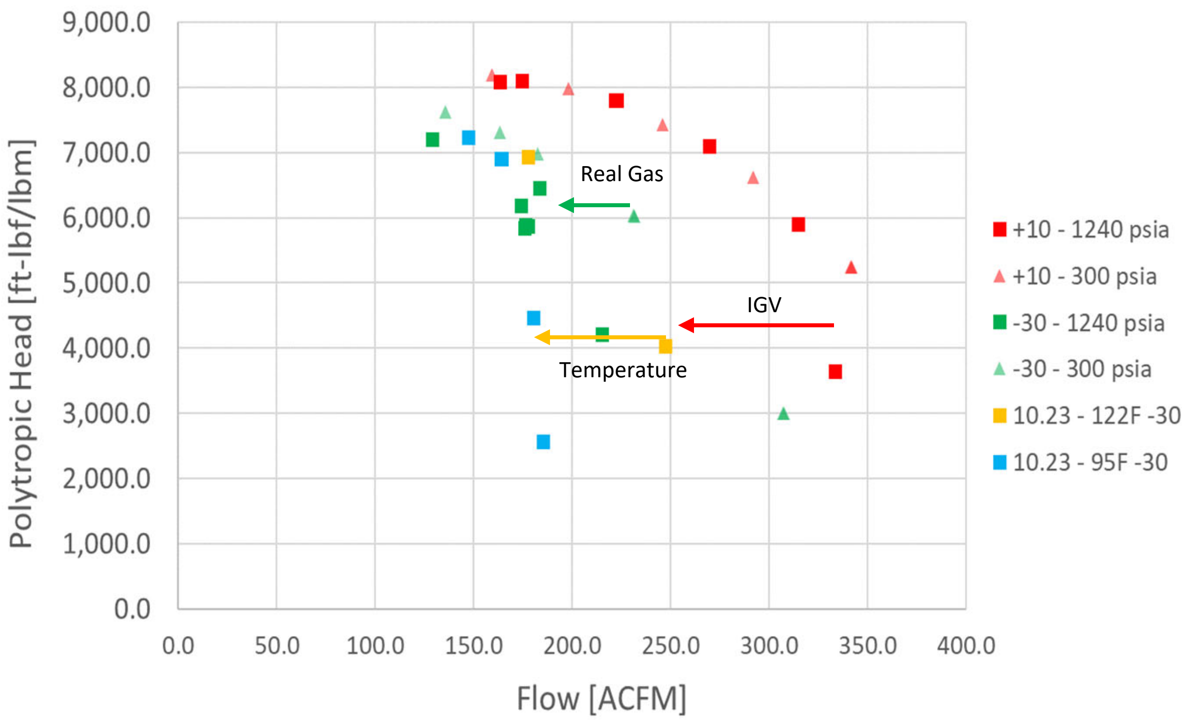


Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle

Main Compressor/Pump-Saturated Liquid or Vapor Inlet

Apollo SCO_2 Main Compressor Performance

- Red and Green triangle highlight low pressure with ideal gas like behavior
- Squares indicate high pressure speed line
- Impact from IGV position and fluid density of compressor head and flow



Supercritical CO2 Cycles Materials

Supercritical CO2 Cycles Material Selection

- CO2 Metal Compatibility/Corrosion
  - Low Temperature -40C to 150C
    - Medium Chrome Steels
  - Medium Temperature 150C to 300C
  - High Temperature 300C+
- CO2 Seal Material Compatibility
  - Elastomeric
  - Rotating Shaft Seals
  - High Temperature Seals

CO2 Corrosion

- Oil Business
  - Pipeline Corrosion
    - Usually due to water or other constituents
- Specific to SCO2 Power
  - MIT
  - Oakridge NL
  - Sandia NL
  - University of Wisconsin

Two Types-Aqueous (Carbonic Acid) and Dry
For the Chemists/Materials Scientists-

3.1 Corrosion Mechanism and Passivation in CO_2

Dry CO_2 is considered practically inert at low temperatures. However, in the presence of even small amounts of water carbonic acid forms that is quite corrosive to many metals. The mechanism of CO_2 corrosion of iron and mild steel in an aqueous environment is given by:

$$CO_{2\,(g)} + H_2O_{(l)} \leftrightarrow H_2CO_{3\,(aq)}$$

$$H_2CO_3 \leftrightarrow H^+ + HCO_3^-$$

$$HCO_3^- \leftrightarrow H^+ + CO_3^{2-}$$

$$2H^+ + 2e^- \rightarrow H_2$$

$$Fe \rightarrow Fe^{2+} + 2e^-$$

$$Fe^{2+} + CO_3^{2-} \rightarrow FeCO_3$$

$$Fe^{2+} + 2HCO_3^- \rightarrow Fe(HCO_3)_2$$

$$Fe(HCO_3)_2 \rightarrow FeCO_3 + CO_2 + H_2O$$

Below temperatures of 360ºC magnetite, $FeCO_3$ acts as a protective layer on the steel. Above this temperature and with high CO_2 pressures, breakaway oxidation can occur through the formation of a duplex layer and carburization (Madina, 2008). Carburization is a potentially significant mechanism of metal degradation and is discussed in detail in proceeding sections of this report.

Although it is well established that CO_2 is corrosive in the presence of water, recent data indicates dry CO_2 may be corrosive under certain conditions. Based on computer simulations, Glezakou at al., 2000 indicated that corrosion of metal surfaces can happen in the presence of very small amounts of water or complete absence of water when very high temperatures exist. The authors state in the absence of water, CO_2 can dissociate and react with other CO_2 molecules to form CO_3^{2-}.

For austenitic steels, defined as having a maximum C content of 0.15% and minimum Cr content of 16% with a face centered cubic structure, several reactions
commonly occur in a CO2 and O2 environment given by:

M + CO2 → MO + CO

2M + CO2 → MO + C

M + CO → MO + C

where M is a metal in the steel. The formation of C can lead to carbide formation.
However, once a protective oxide layer has been established these reactions cease.
The high Cr content in austenitic steels enables the formation of a Cr2O3 layer that is
highly protective (Gibbs, 2008).

Nickel based alloys, such as Inconel and Hastelloy, form a continuous protective oxide layer. Gibbs (2008) states the protective layer is composed of 50%
NiO and 50% Cr2O3. Both of these oxides are highly stable.

From, "Milestone Report"
METAL CORROSION IN A SUPERCRITICAL CARBON DIOXIDE – LIQUID SODIUM POWER CYCLE
Moore, Conboy 2012

CO2 Corrosion

Gibbs, MIT 2010

- Gibbs, MIT 2010, for Nuclear Reactor Use
  - 610C and 20 MPa, 3000 hour test
  - F91, HcM12A, 316SS, 310SS, AL-6XN, Haynes 230, Alloy 625, PE-16, PM2000
  - Highest Chromium and Nickel Content are Best

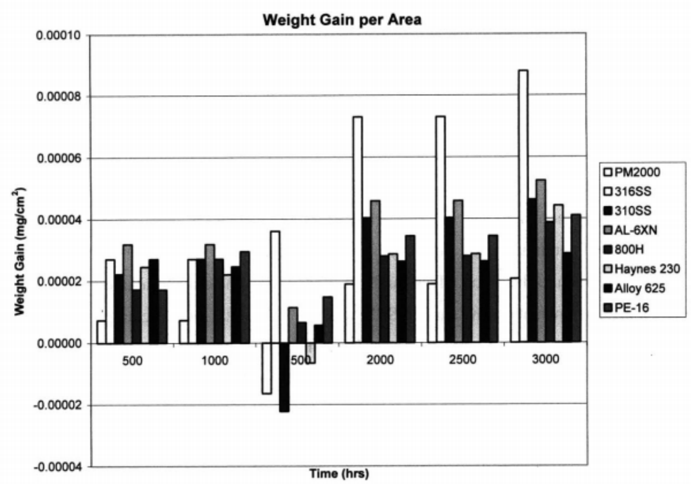


Figure 3.8: Weight gain per area, Alloys F91 and HCM12A not present

Table 3-2: 3000 hr weight gain

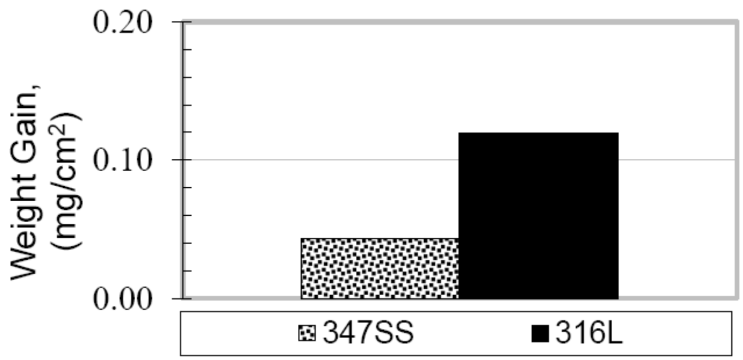
Alloy	**Weight gain at 3000 hours (mg/cm^2)**
F91	4.1×10^{-3}
HCM12A	5.5×10^{-3}
PM2000	2.1×10^{-5}
316SS	8.7×10^{-5}
310SS	4.6×10^{-5}
AL-6XN	5.2×10^{-5}
800H	3.9×10^{-5}
Haynes 230	4.4×10^{-5}
Alloy 625	2.9×10^{-5}
PE-16	4.1×10^{-5}

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

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



CO2 Materials

Static and Dynamic Seals and Electric Machines

- 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
  - Teflon, PEEK, Graphite for Labyrinth Seals
  - Graphite and Carbide Liftoff Gas Seals
- High Temperature Static Seals
  - Silver Plated Inconel “C” Seals
- Electric Machines (rapid decompression testing)
  - Most Common Insulation Materials withstand SCO2 Operation
  - MW35C wire insulation tested
  - Epoxy Type Varnish Works Best

CO_2 Materials Selection, Seals

- Static Seals, Elastomeric Seals Can Absorb High Pressure CO_2. Rapid Depressurization Can Then Destroy the Seals
  - XNBR, HNBR, Available Bulk Purchase Only
  - EPDM, Widely Available, less suitable
  - Kalrez
- Rotating Shaft Seals
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  - 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

Supercritical CO2 Cycles Pressure Containment

Supercritical CO2 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 300C, 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

Page 1 of 2

KTL - 3rd Annual Pressure Equipment Conference, 1999
Last Update - October, 2000

ASME SECTION VIII-For Rotating Machinery

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;

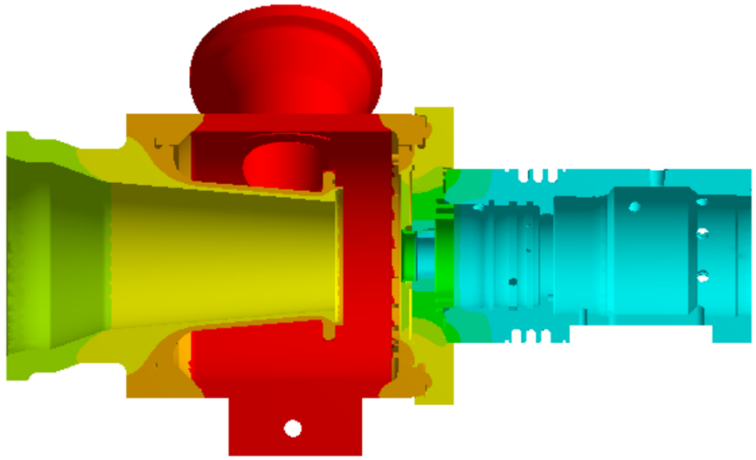
(c) pressure containers which are integral parts or components of rotating or reciprocating mechanical devices, such as pumps, compressors, turbines, generators, engines, and hydraulic or pneumatic cylinders where the primary design considerations and/or stresses are derived from the functional requirements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

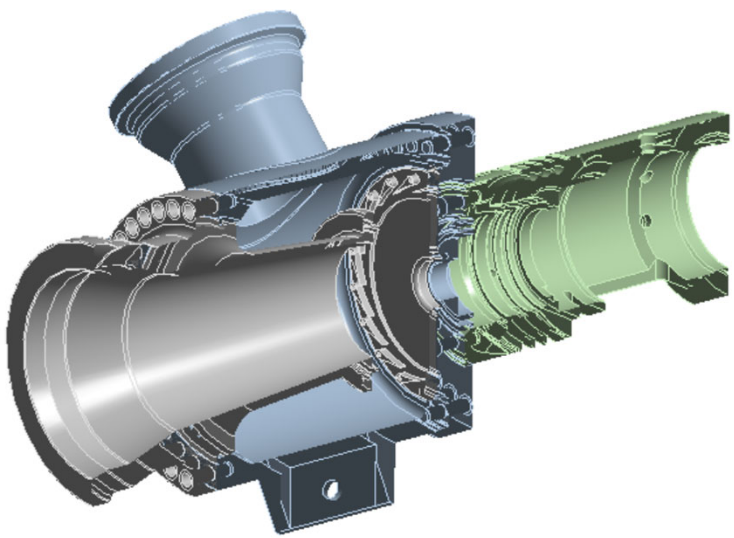
- Useful for Defining Safety Margins
  - 1.5X on Yield Strength, 3.5X on Ultimate Tensile Strength
- Useful for Defining Hydrostatic Test Requirements
  - 1.3X MAWP (Temperature Rated)
- Useful for Material Selection and Temperature/Stress De-rating

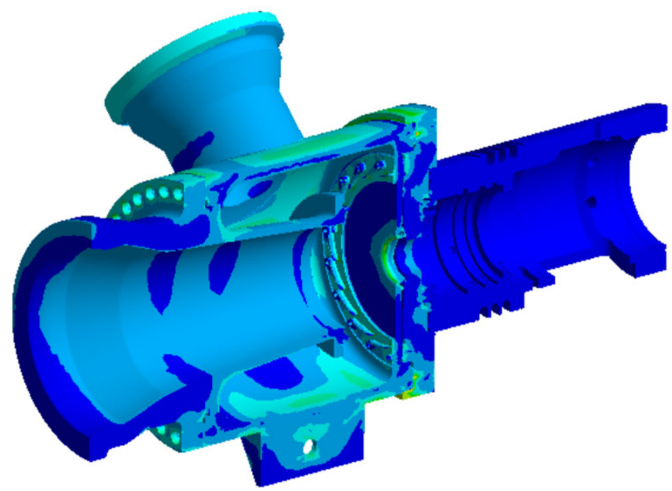
- Not Cognizant of Complicated Geometry Found in Turbomachinery (Can use Div 2 for FEA)
- Transient Thermal Stresses

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

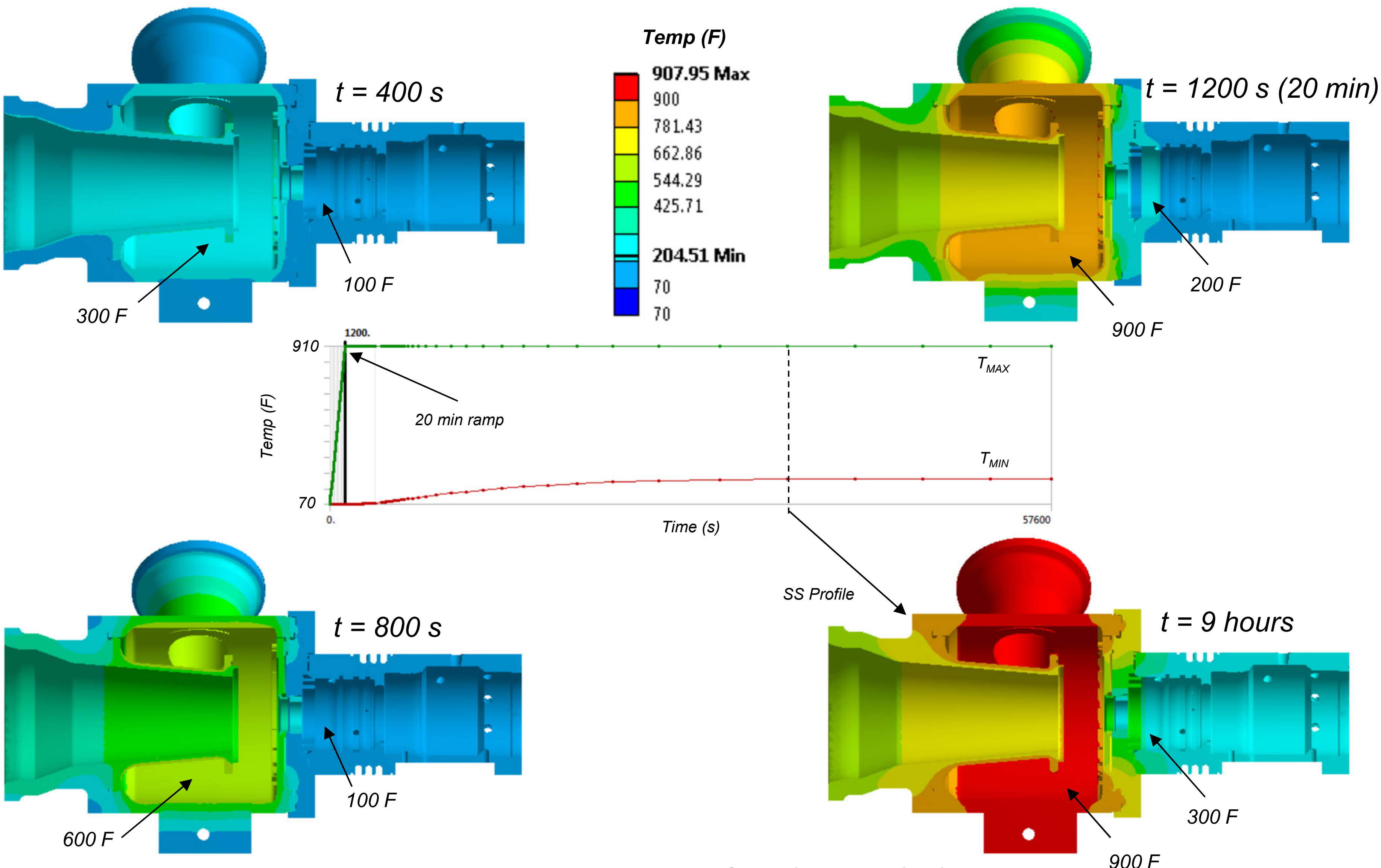






Courtesy of Barber-Nichols

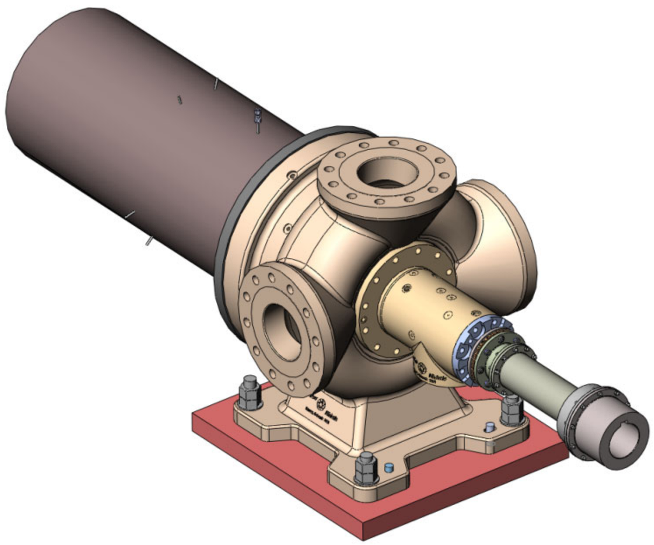
Radial Turbine Housing – Transient Thermal Profile

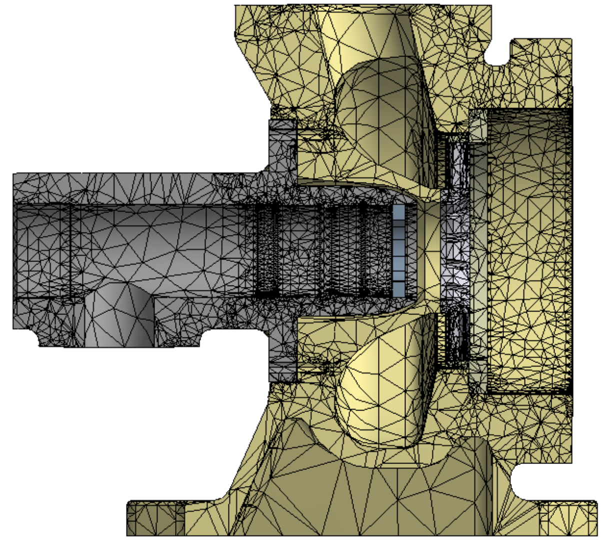


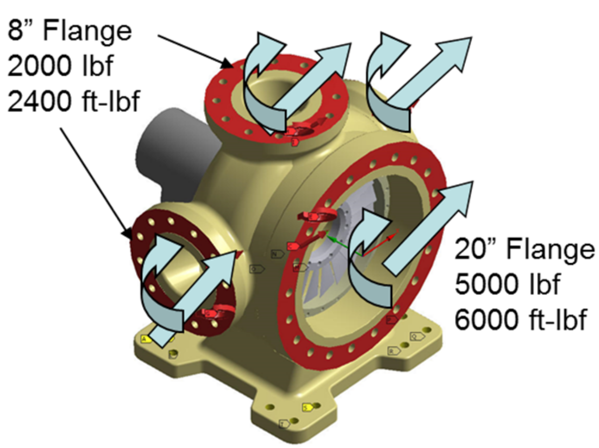
Courtesy of Barber-Nichols

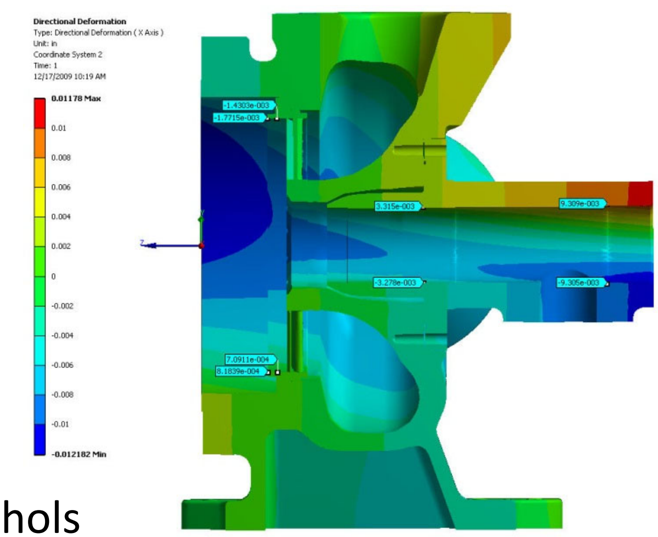
Thermally Induced Stresses Can Limit Startup Time

Single Stage Axial Turbine Example









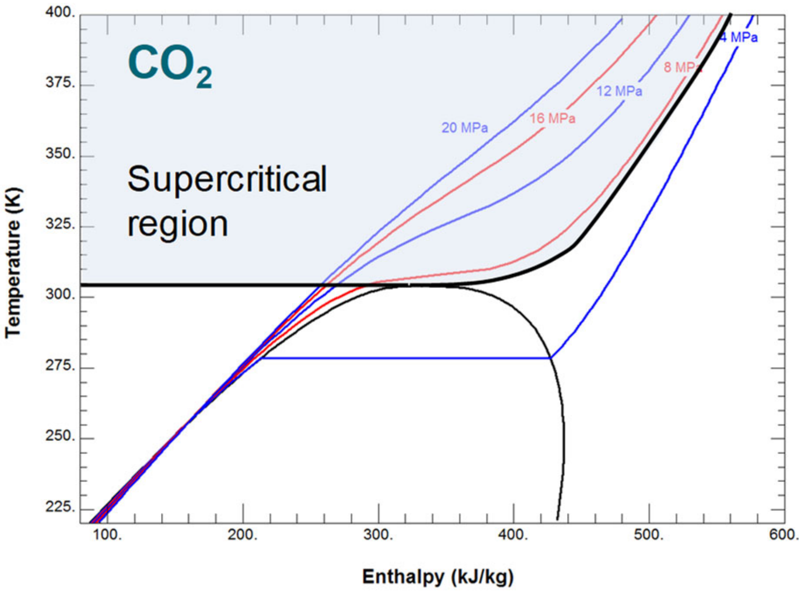
Courtesy of Barber-Nichols

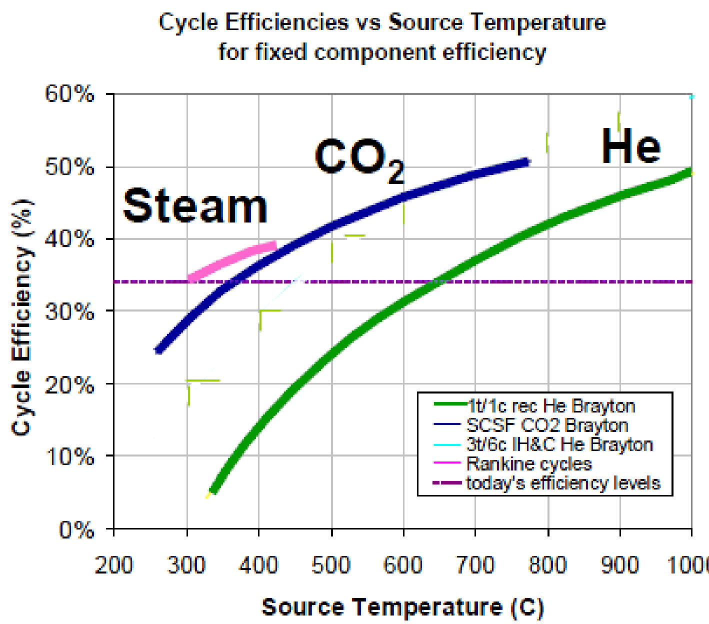
Combined Thermal, Pressure, Pipe Loading

Summary

Both supercritical power cycles and the use of S-CO_2 are not new concepts

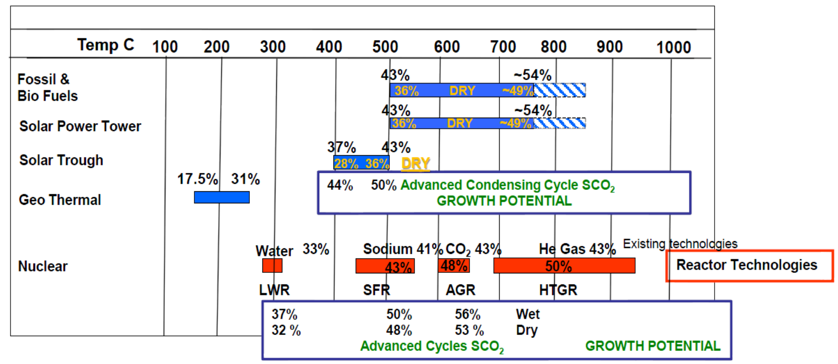
S-CO_2 is desirable for power cycles because of its near-critical fluid properties



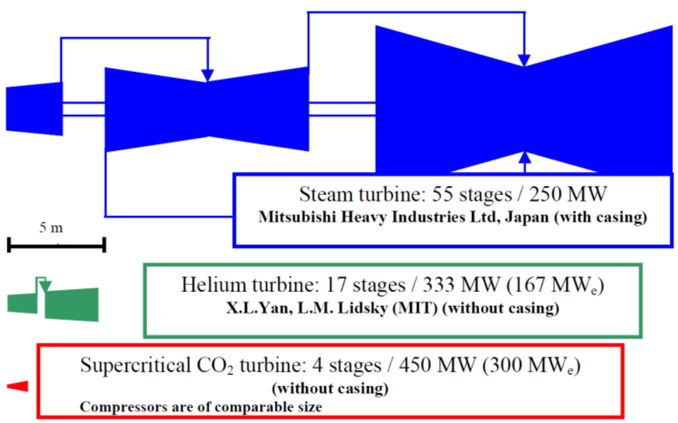


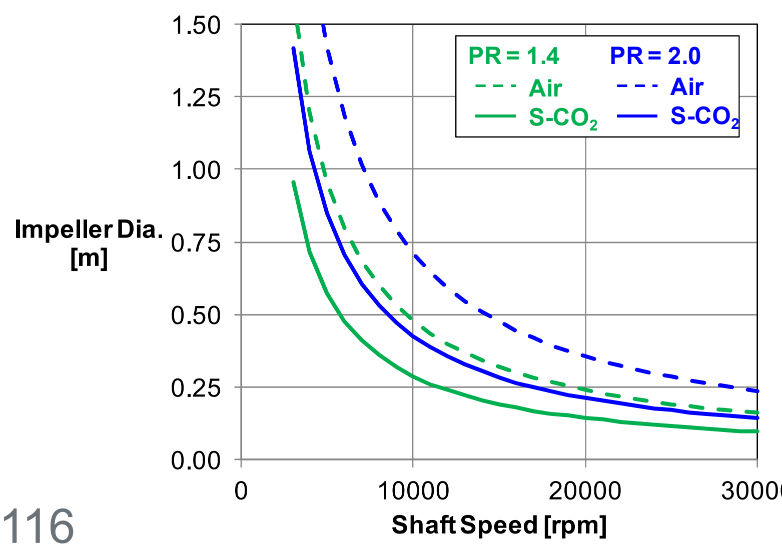
S-CO_2 power cycles can be applied to many heat sources and have a small footprint

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 S-CO_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 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.