Tutorial: Turbomachinery Design and Operation for Supercritical CO2 Applications

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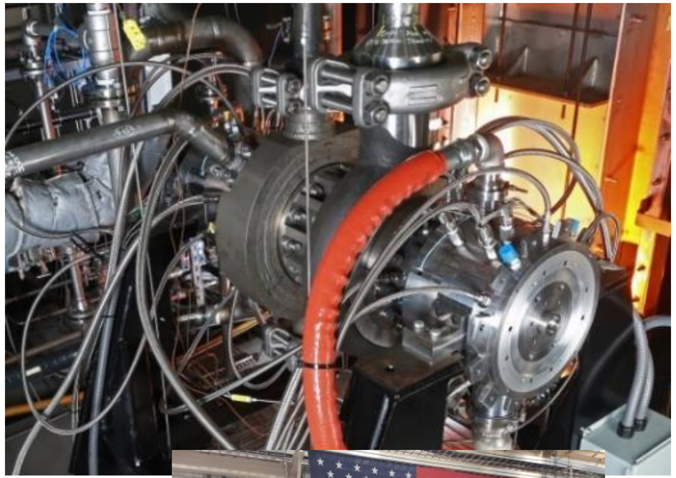
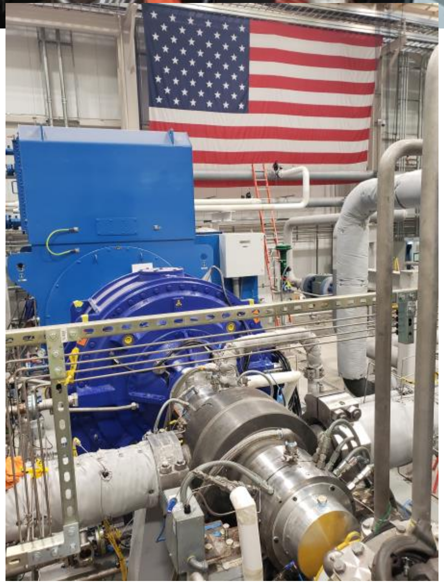


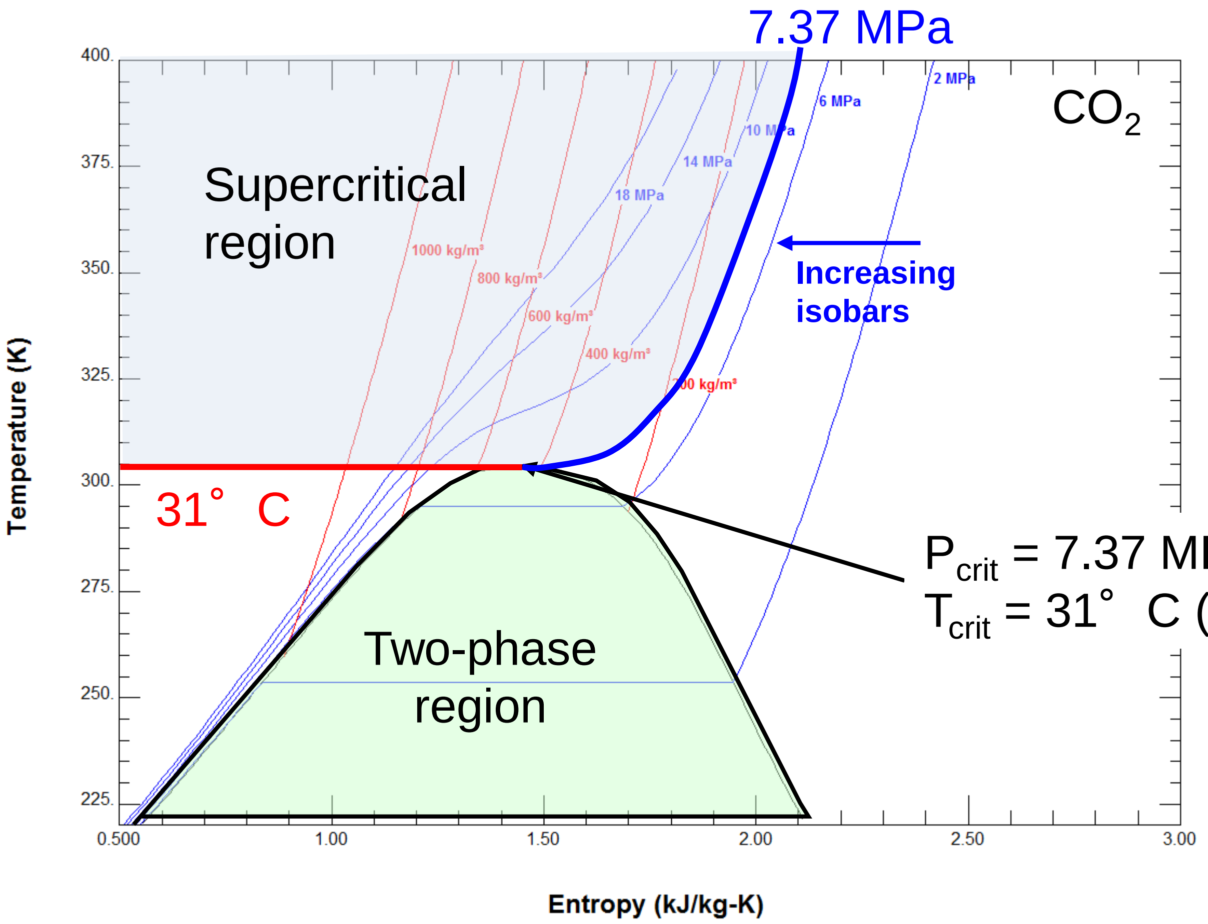


Outline

- Introduction to Supercritical CO_2
- Machinery Types and Layouts
- Existing Machinery
- Aerodynamic Requirements & Design
- Seals and Bearings
- Rotordynamics and Blade Dynamics
- Thermal Management
- Mechanical Design Process

Introduction to sCO_2

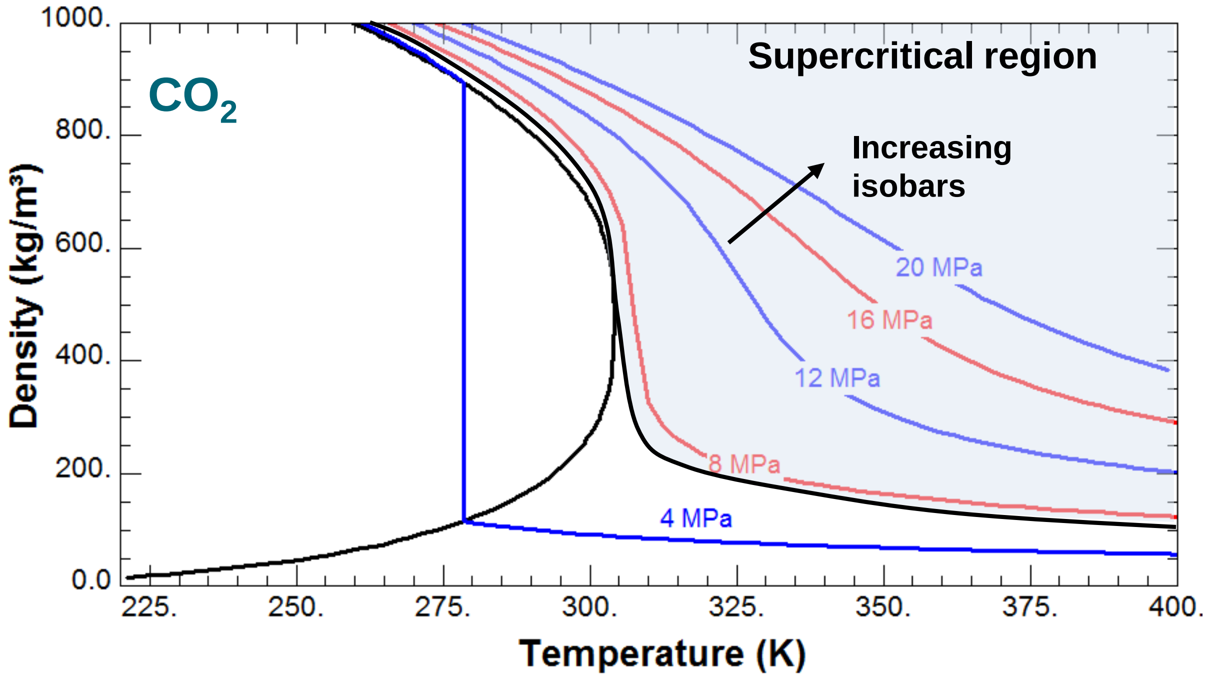
A fluid is supercritical if the pressure and temperature are greater than the critical values



P_{crit} = 7.37 MPa (1070 psi)
T_{crit} = 31° C (88° F)

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

Fluid density sharply decreases near the critical point



REFPROP (2007)

Layouts & Machinery Types

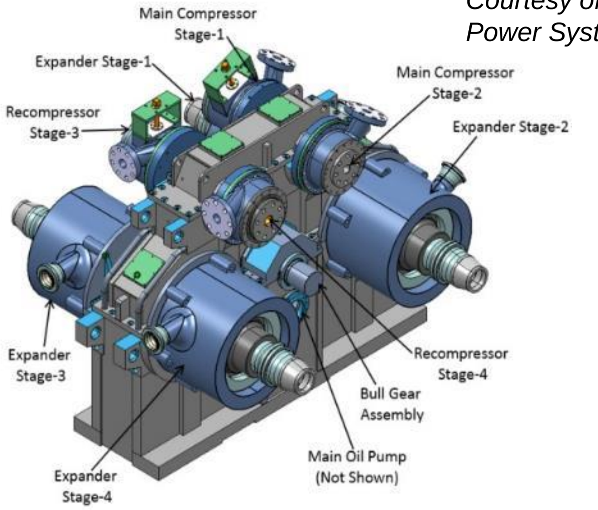
Turbomachines for Supercritical CO_2 Cycles

- 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

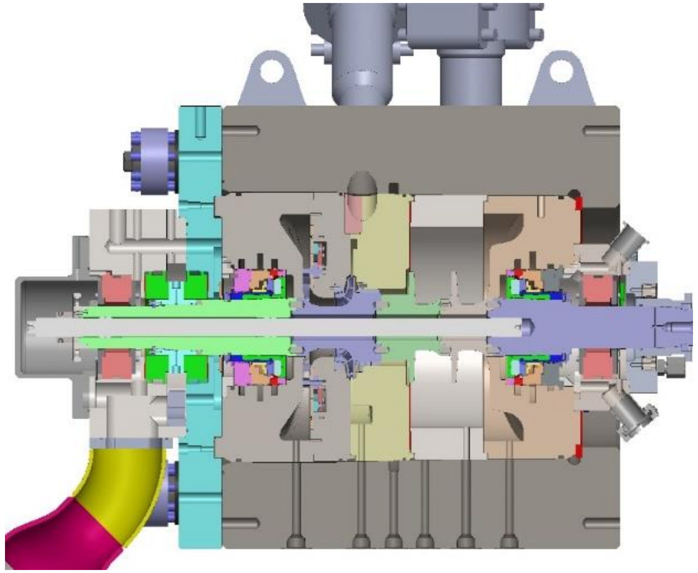
Courtesy of Hanwha Power Systems



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

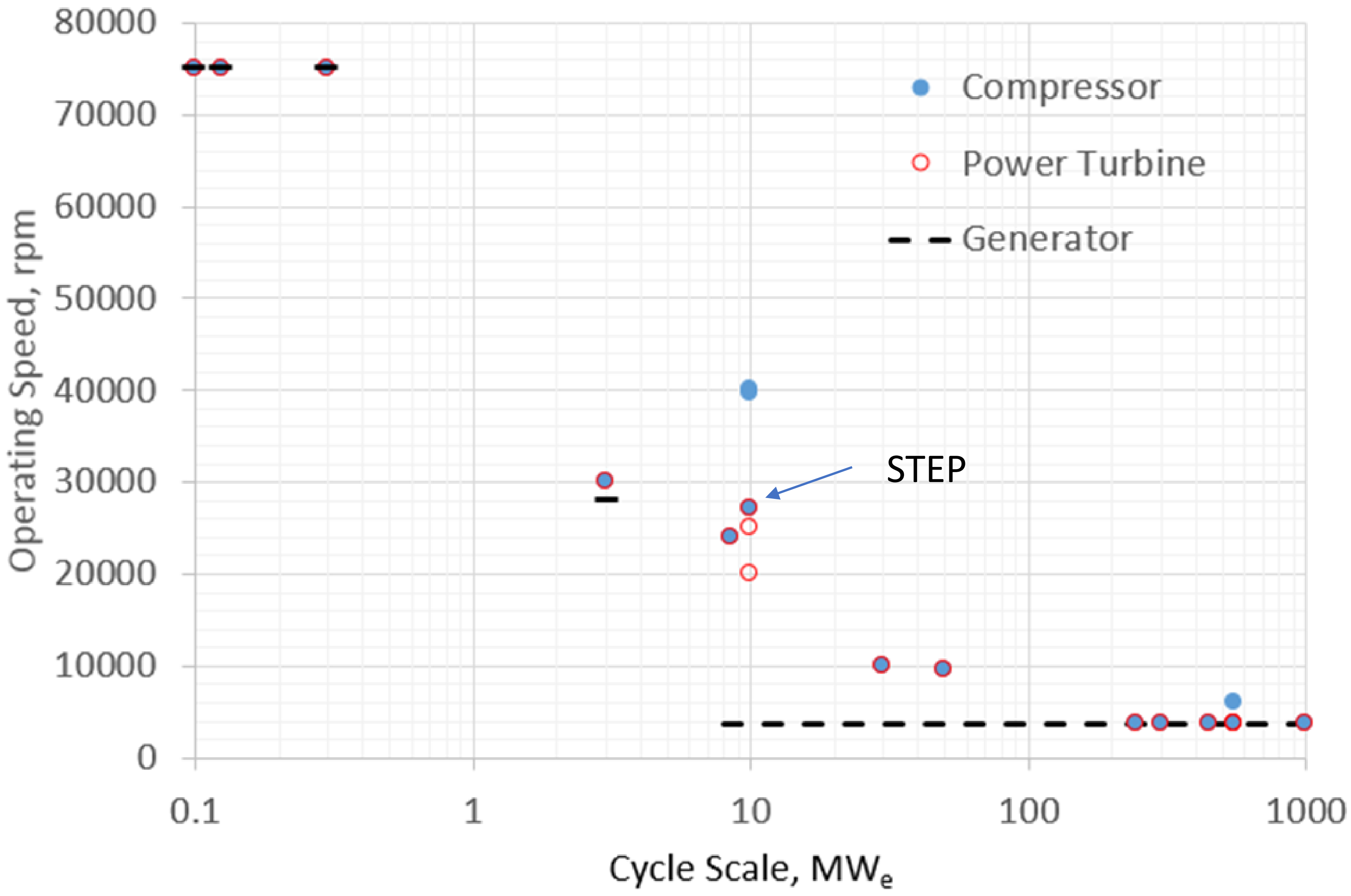
Single-Shaft Multi-stage Centrifugal Compressor



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

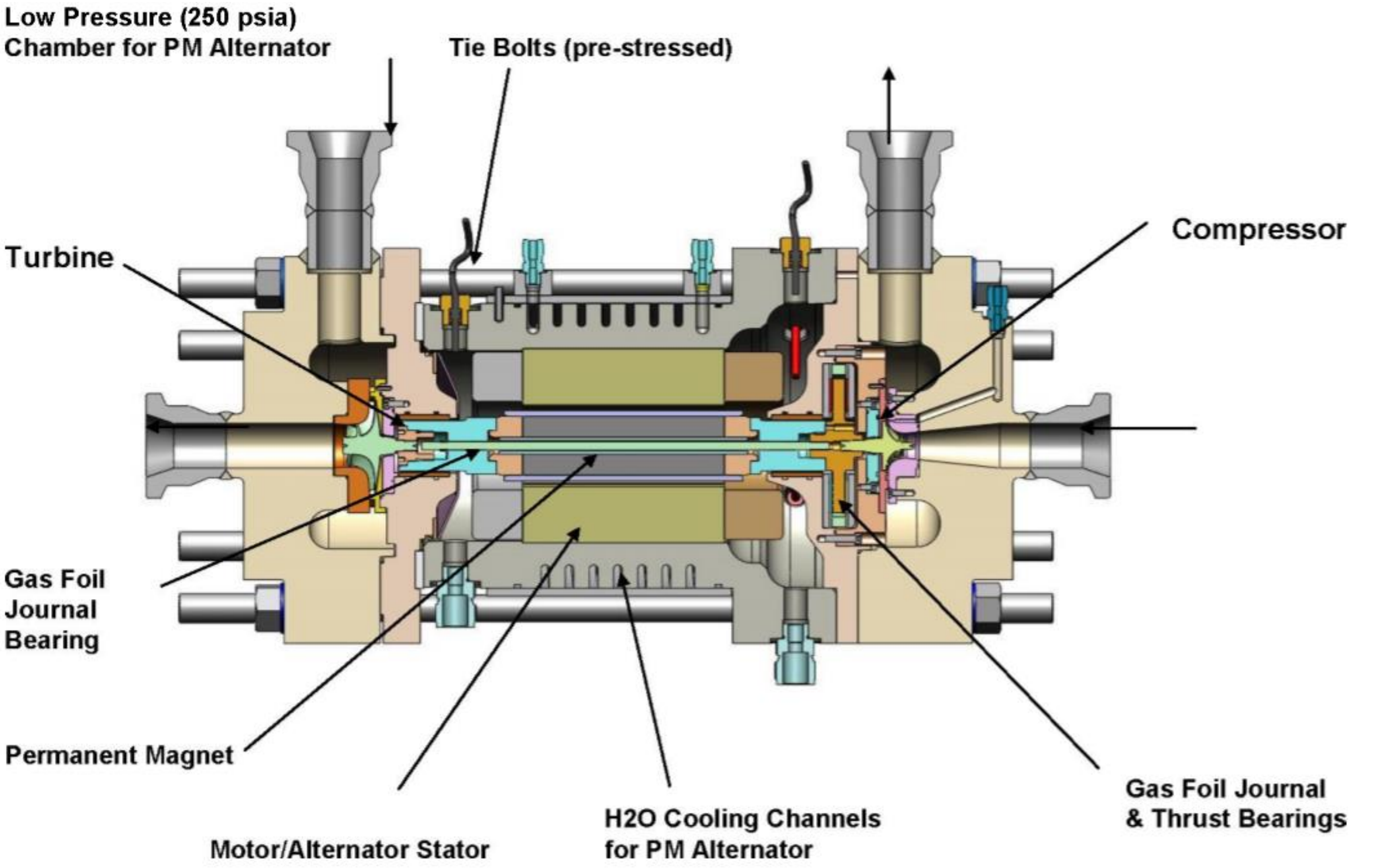
- Multi-stage centrifugal proven reliable and used in many critical service applications currently (oil refining, LNG production, etc.)
- Fewer bearings and seals
- Highest reported pressure of 550 bar in CO_2 and 900 bar in natural gas service

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)



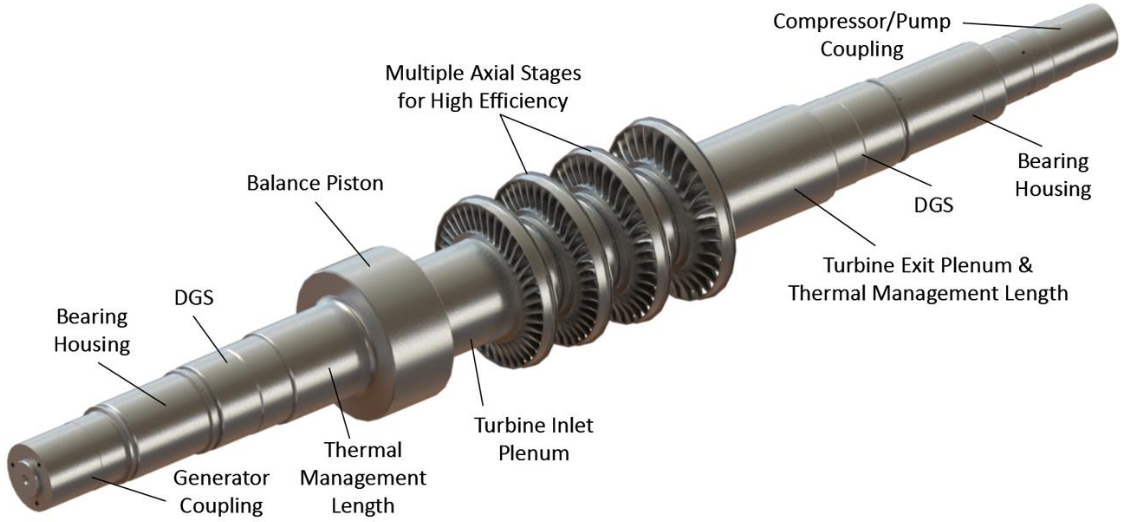
Wright, S.A., Conboy, T.M. and Rochau, G.E. (2011) "Break-Even Power Transients for Two Simple Recuperated s-CO_2 Brayton Cycle Test Configurations", in *Supercritical* CO_2 *Power Cycle Symposium*, Boulder, CO.

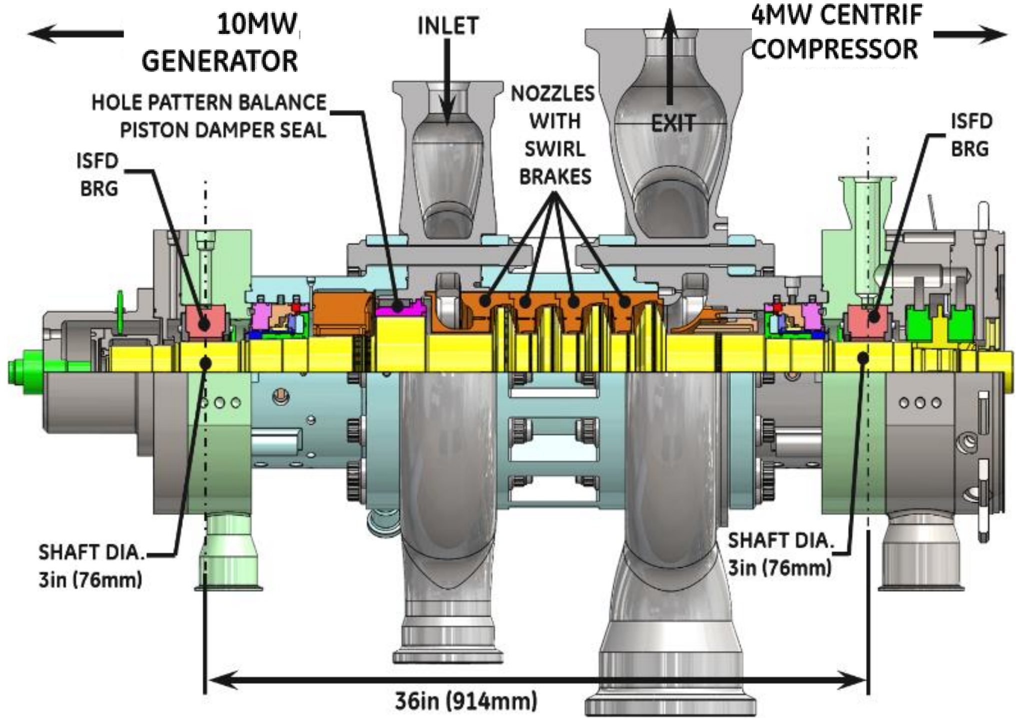
Experience with TAC Units

- 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



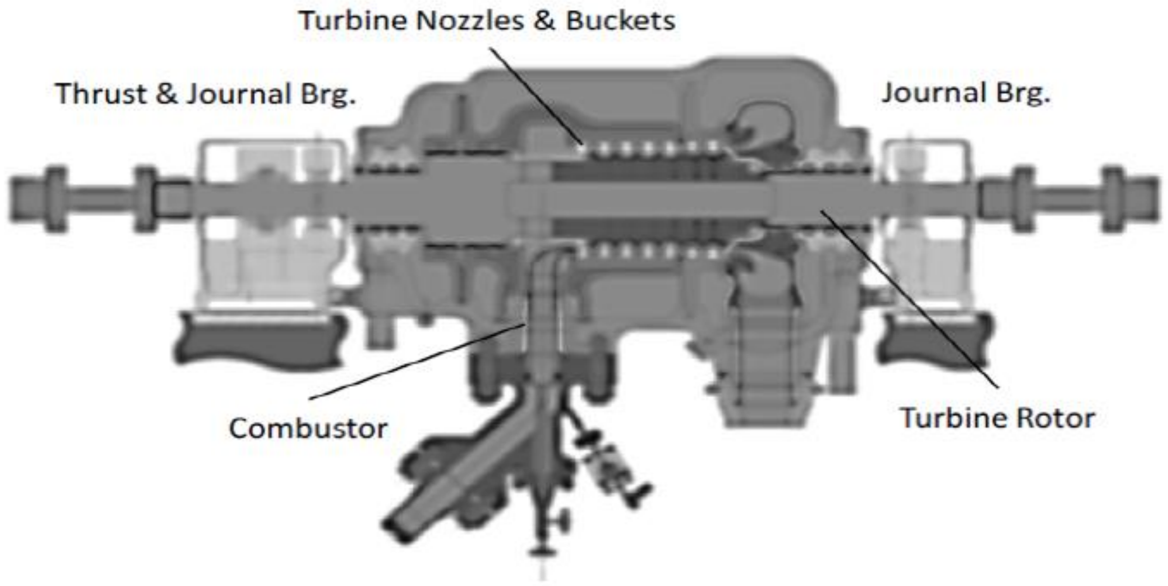


Ertas, B. Delgado, A., Moore, J., 2017, "Dynamic Characterization Of An Integral Squeeze Film Bearing Support Damper For A Supercritical Co2 Expander", Gt2017-63448, ASME Turbo Expo, Charlotte, NC

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

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

- Turbine Inlet: 300 bara, 950 C
- Cooled Rotor

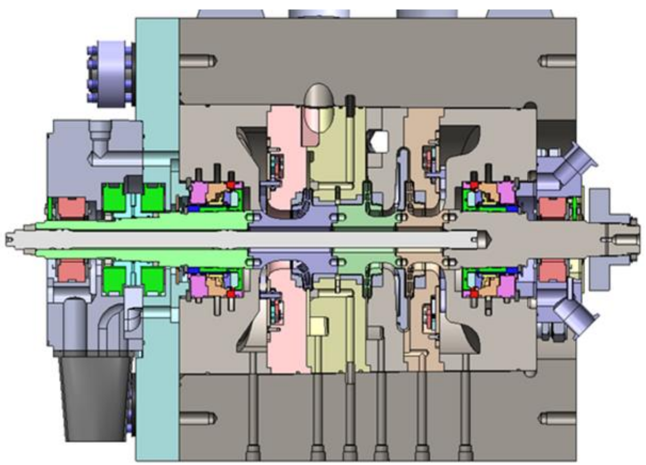


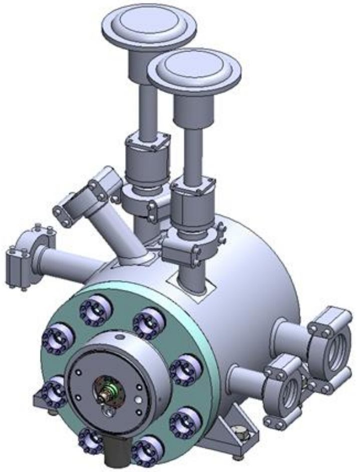
Iwai, Y., Ito, M., Morisawa, Y., Suzuki, S., Cusano, D., and Harris, M. (2015) "Development Approach to the Combustor of Gas Turbine for Oxy-Fuel, Supercritical CO_2 Cycle", *Proc. ASME Turbo Expo GT2015-43160*, Montreal, Canada.

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

PROJECT OBJECTIVES

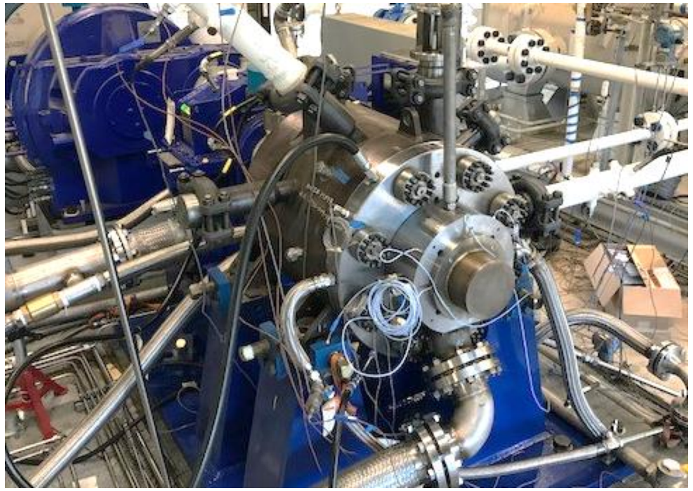
- Develop high-efficiency sCO_2 compression system
  - Main Compressor Efficiency of 80%
  - Preliminary Design completion June 2016
- High efficiency centrifugal impeller
- Variable inlet guide vanes (IGV)
- Advanced aerodynamic design provided by GE implemented into the detail compressor design provided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





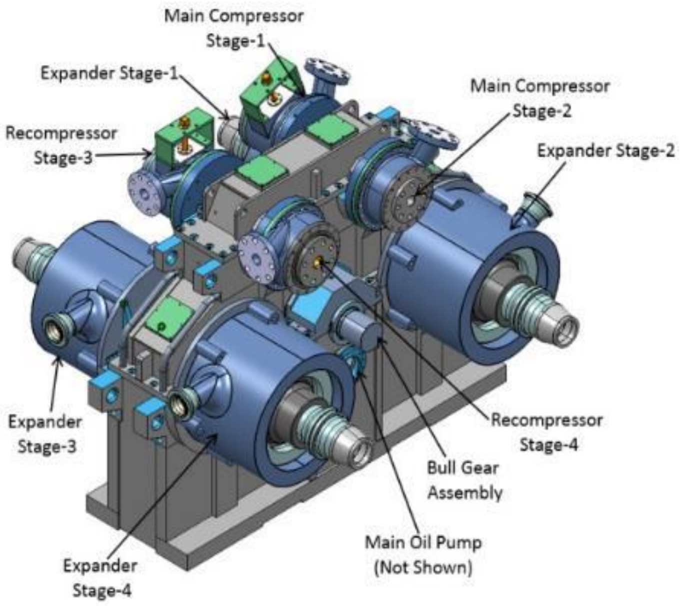
KEY RESULTS AND OUTCOMES

- 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

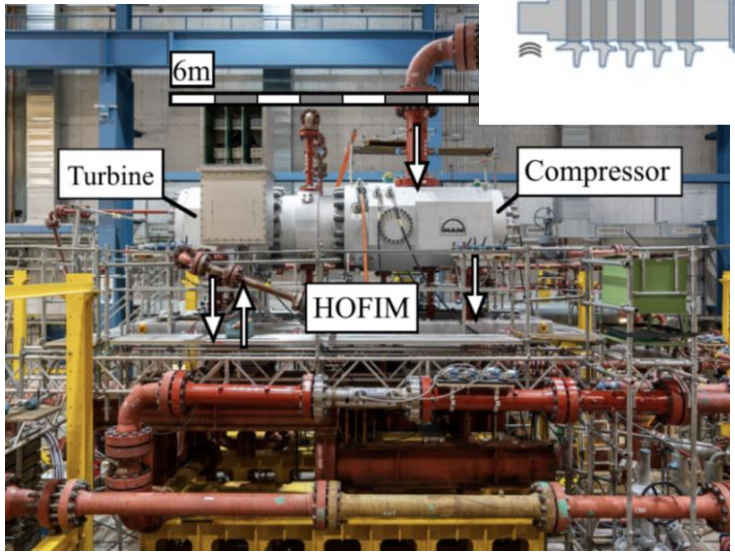


Wilkes, J., Allison, T., Schmitt, J., Bennett, J., Wygant, K., Pelton, R., Bosen, W. (2016) "Application of an Integrally Geared Compander to an sCO_2 Recompression Brayton Cycle," in *The 5th International Symposium - Supercritical* CO_2 *Power Cycles*, San Antonio, TX.

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



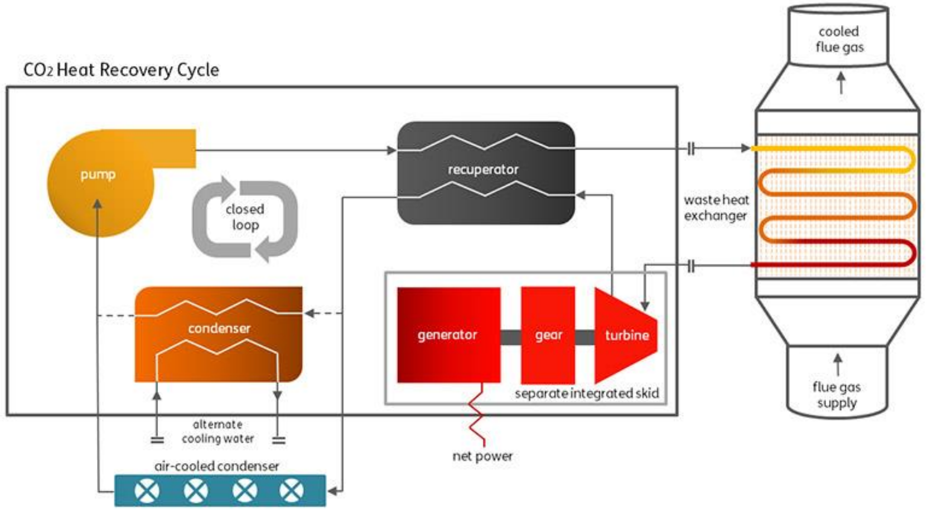
Wolscht, L., Somaini, R., Jacquemoud, E., Jenny, P., 2023, “Full Scale Demonstration and Validation of a 35 MW Transcritical CO2 Heat Pump,” GT2023-101356, ASME Turbo Expo, Boston, MA

Echogen Waste Heat Recovery

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

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

echogen.com



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

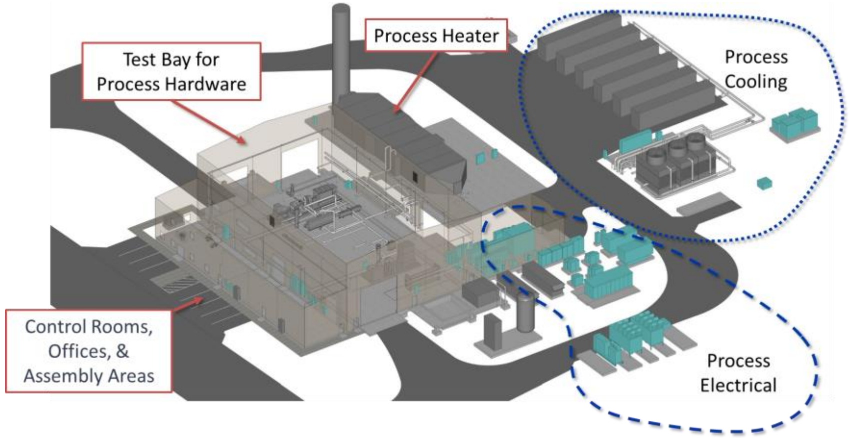
Quantify performance benefits:

- 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

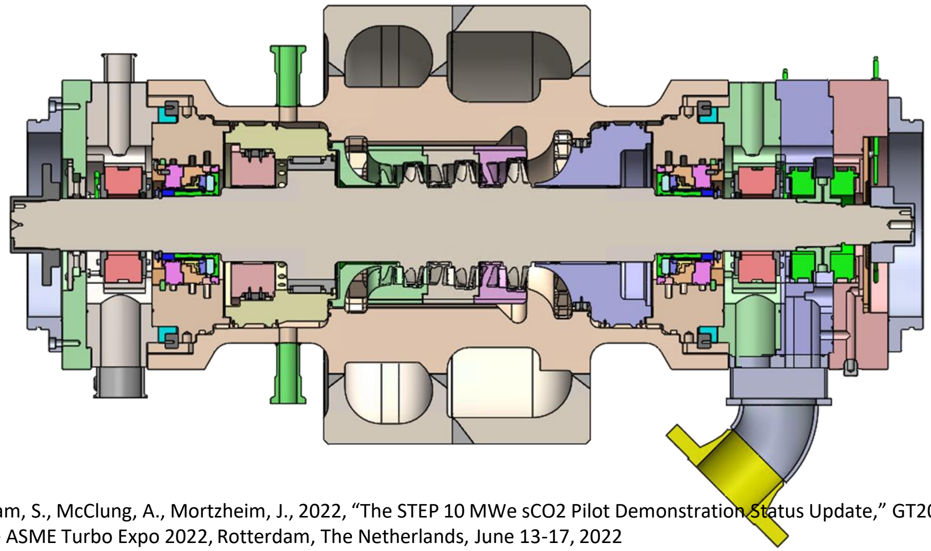
Achieved **mechanical completion** October 2023



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
- One of highest power density of any industrial turbine
- Inlet conditions: 265 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
- 18,000 rpm at 200C turbine inlet conditions achieved to date



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STEP DEMO

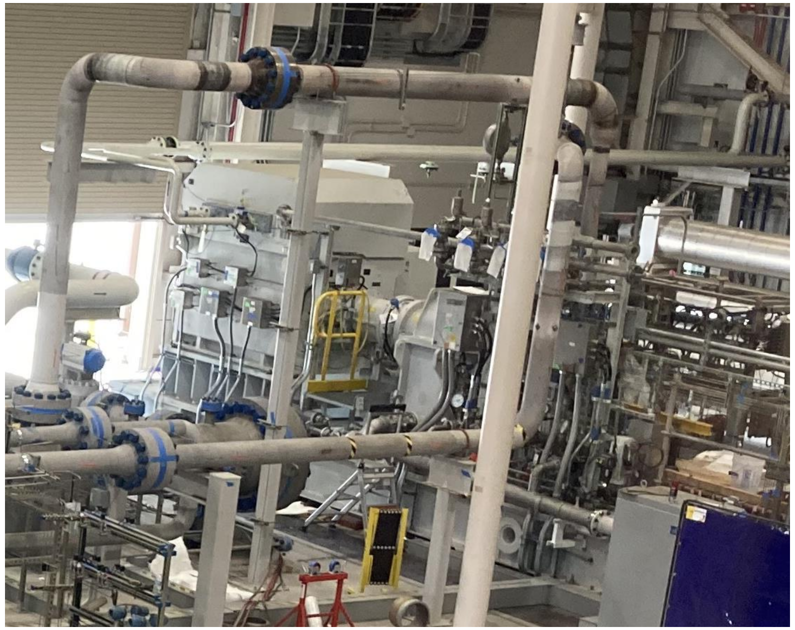
NATIONAL ENERGY TECHNOLOGY LABORATORY

gti

SwRI

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



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
- Volume reduction significant due to high density and real gas effects
  - Results in low flow coefficient impellers in final stages for in-line barrel compressors
- Potential for condensation/cavitation at compressor inlet

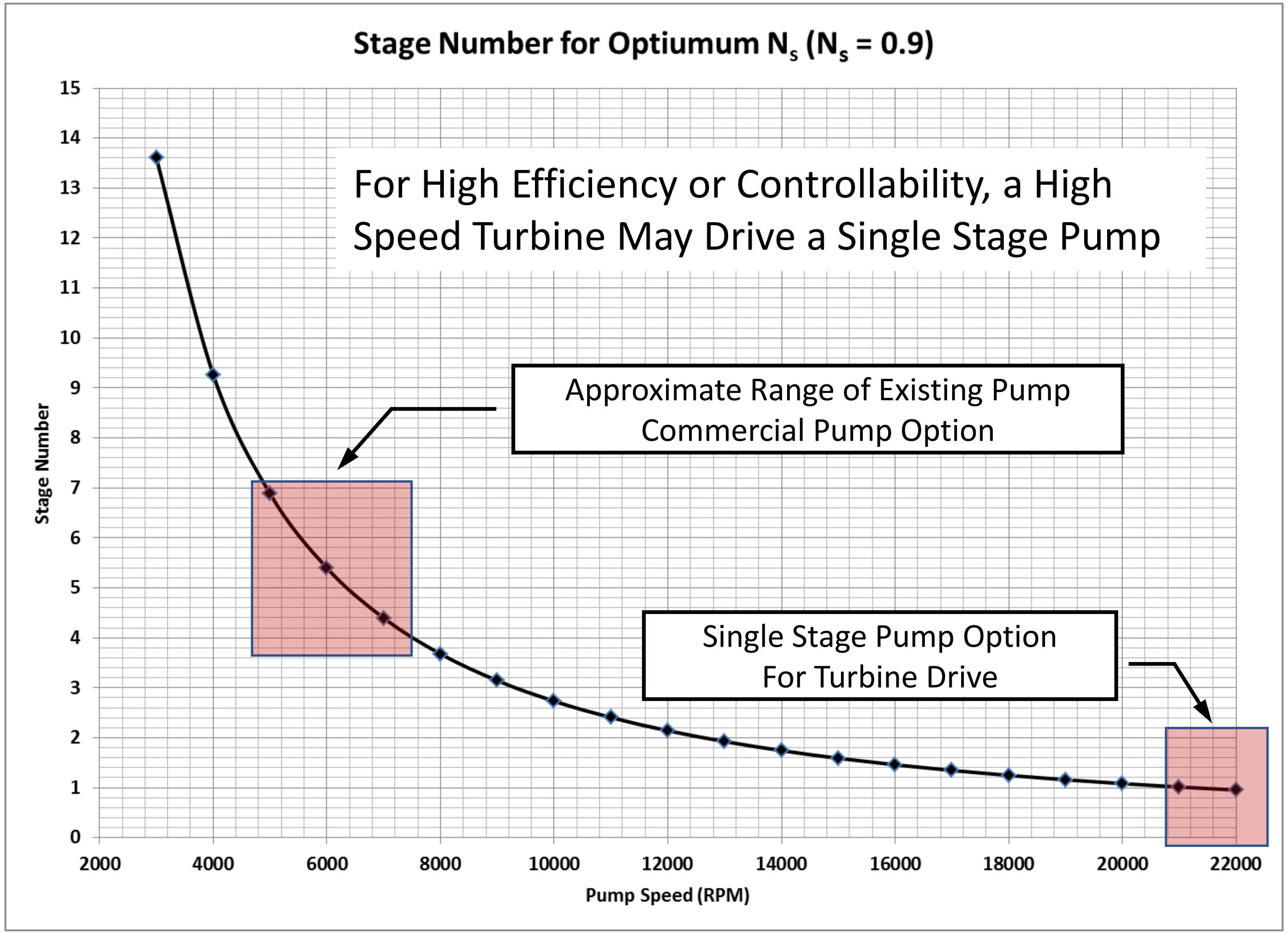
Axial/Radial Designs

- 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

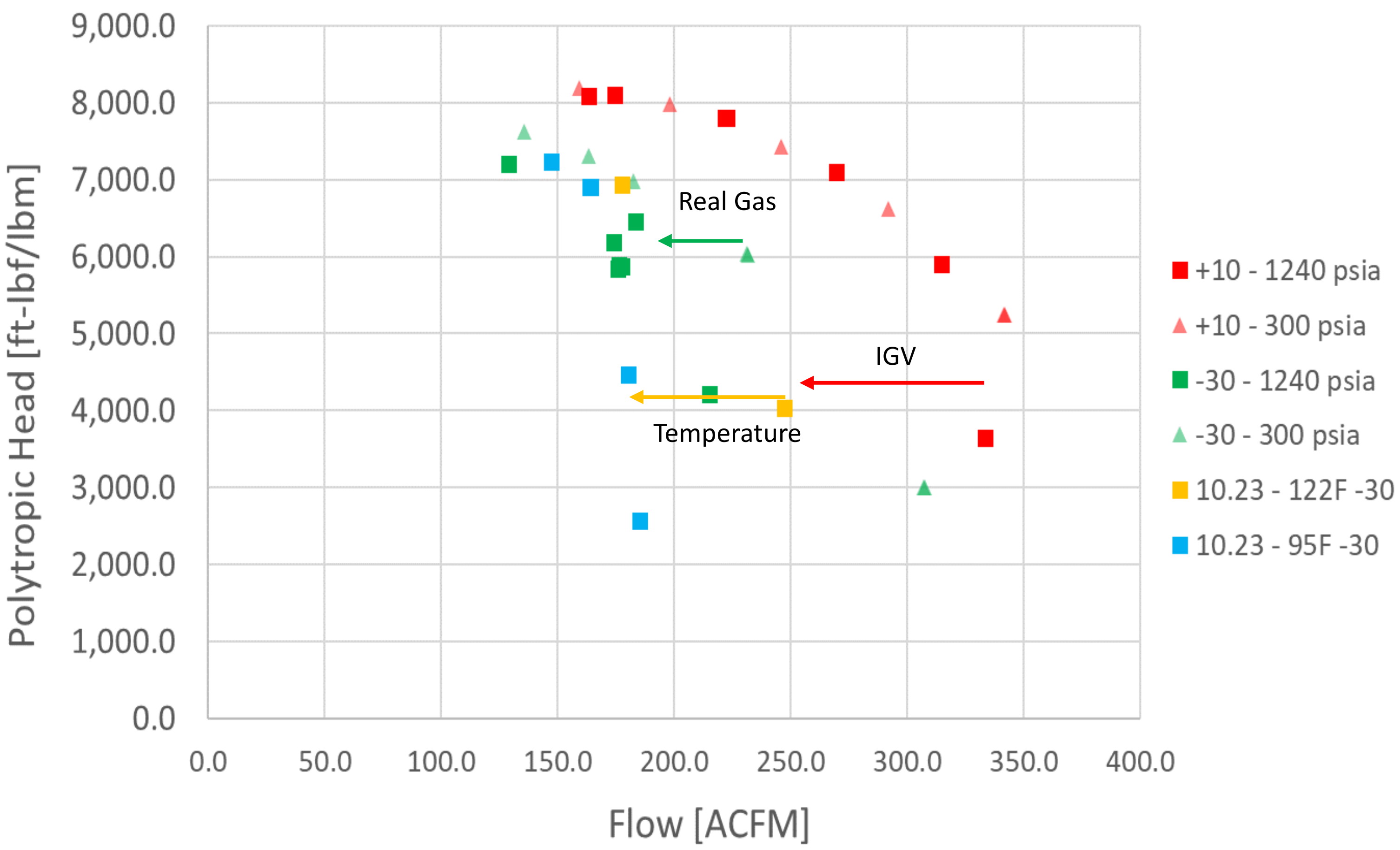
- Centrifugal compressor impellers typically closed
  - Mechanically robust
  - Insensitive to axial motion
  - Low head requirement

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



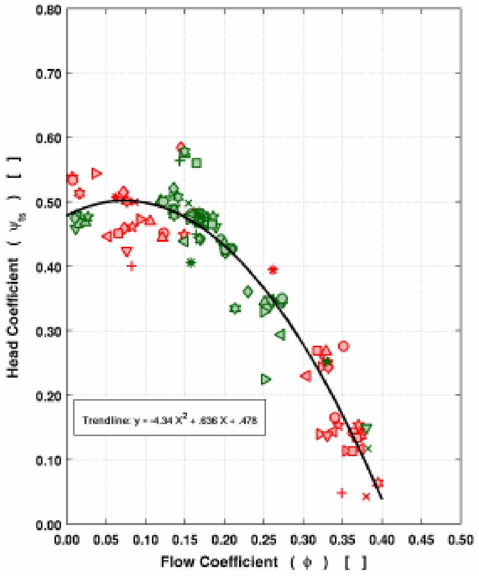
SCO2 Compressor Test Results

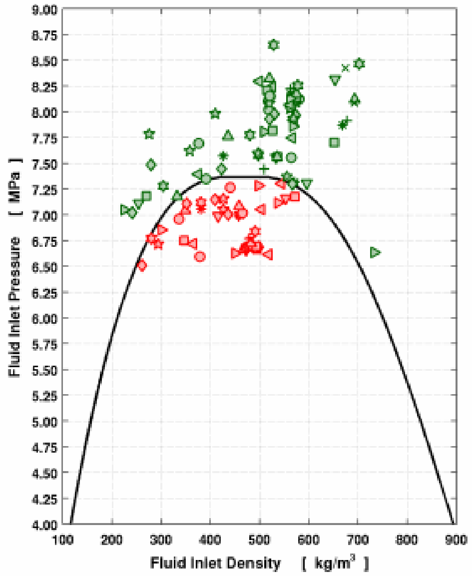
- After break-in, testing performed at medium and full pressure
  - 300 and 1240 psi suction pressure
  - +10° and -30° IGV setting angles
  - 95° and 122°F suction temperature
- Both speed of sound and IGV setting has strong effect on the flow capacity of the compressor



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





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

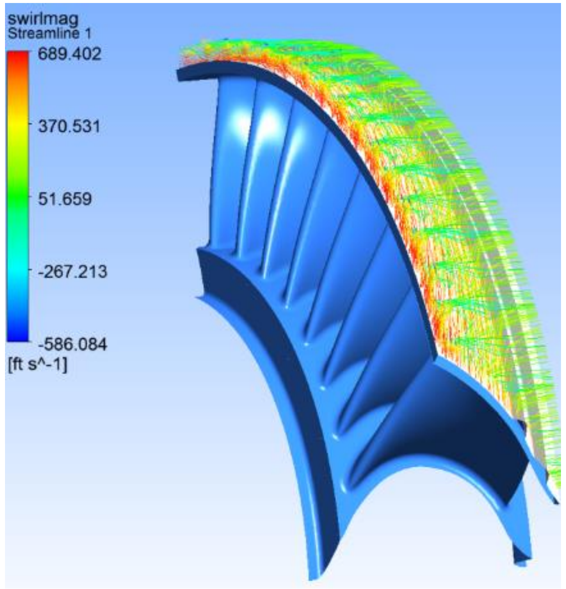
Seals

Labyrinth:

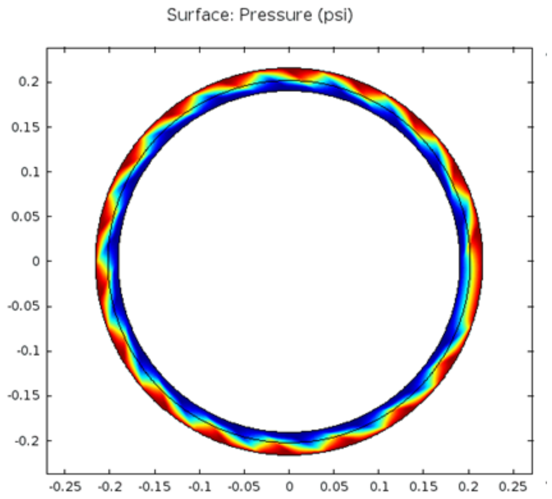
- Labyrinth seals used at blade tip and interstage locations
- Swirl brakes used to minimize swirl entering seal
  - CFD used to optimize and evaluate swirl brake performance

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

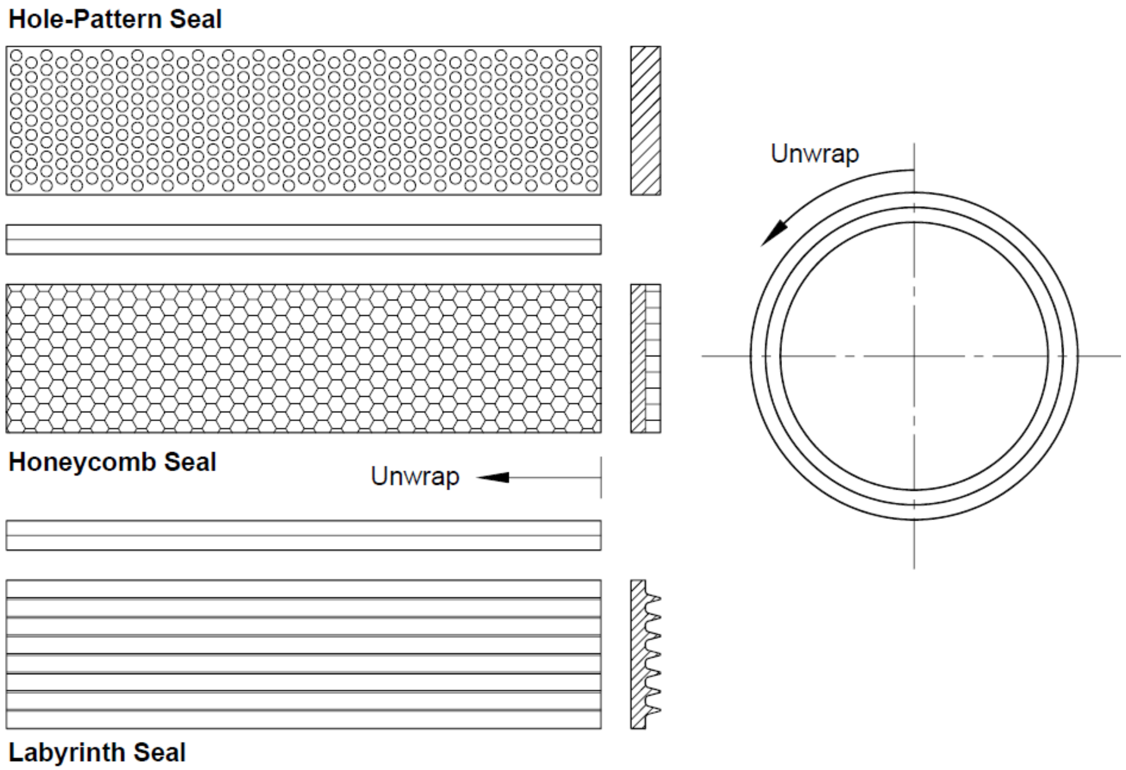


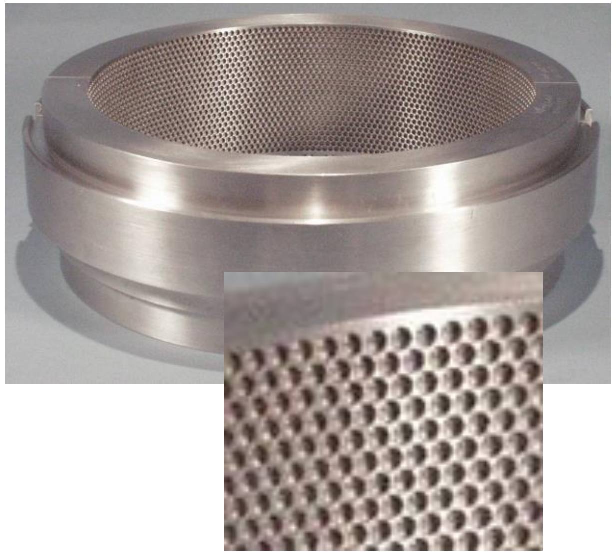
CFD analysis of interstage laby seal flow in CO_2



DGS Face Pressure Distribution from CFD

Different Seal Geometries





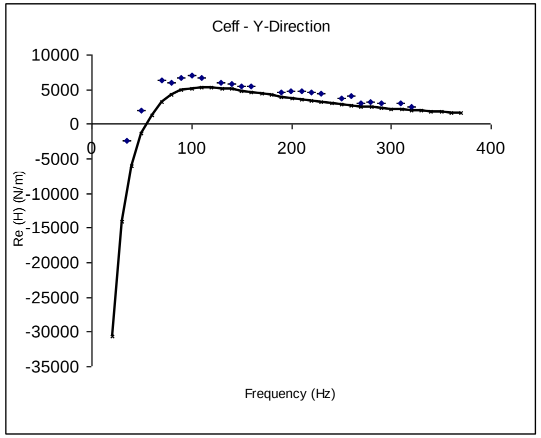
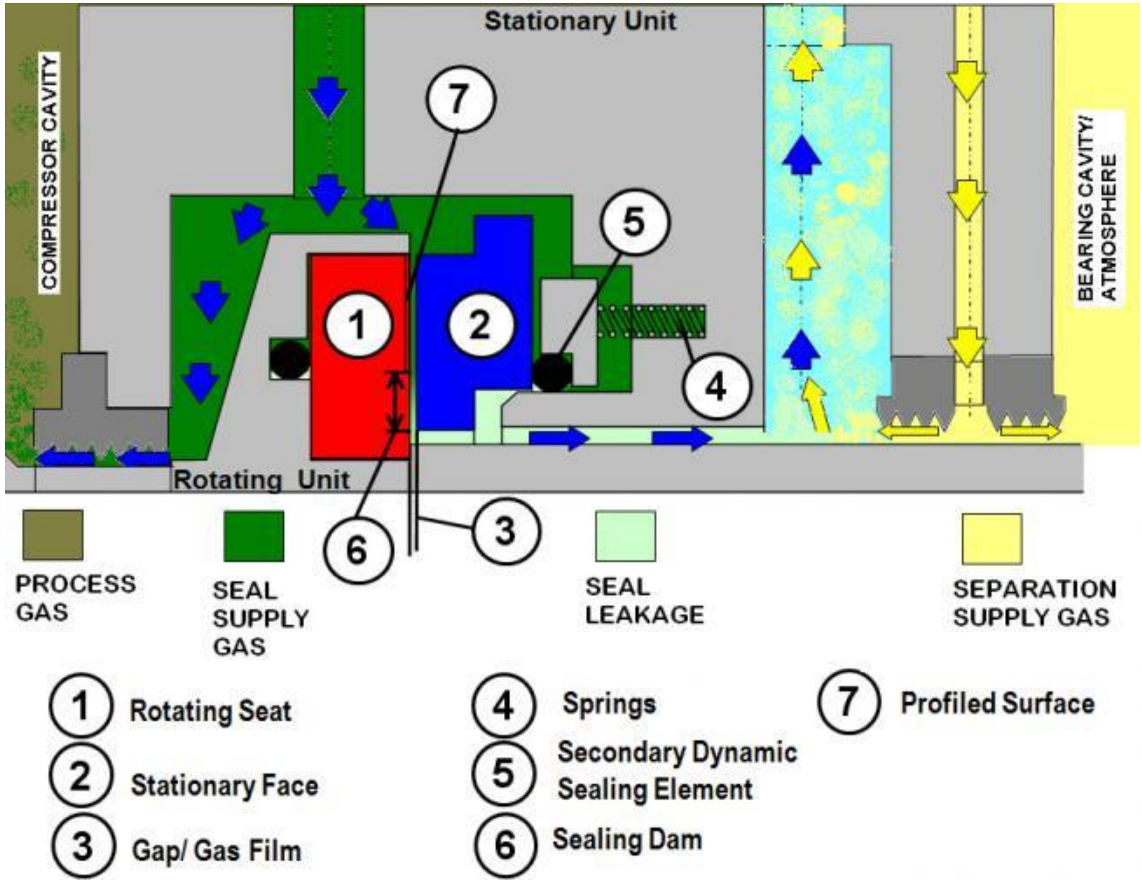


Image source [7-2]

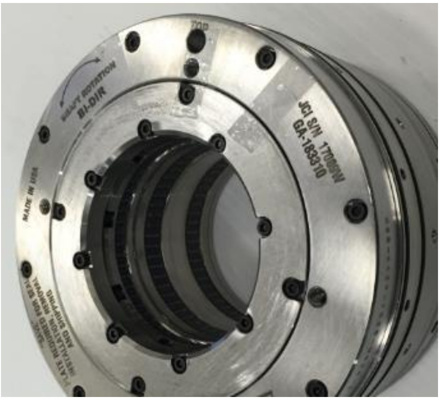
Reference: Camatti, M., Vannini, G., Fulton, J.W., Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," ***Proc. of the Thirty-Second Turbomachinery Symposium*****, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College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

Dry Gas Seals



Rotating seal surface…

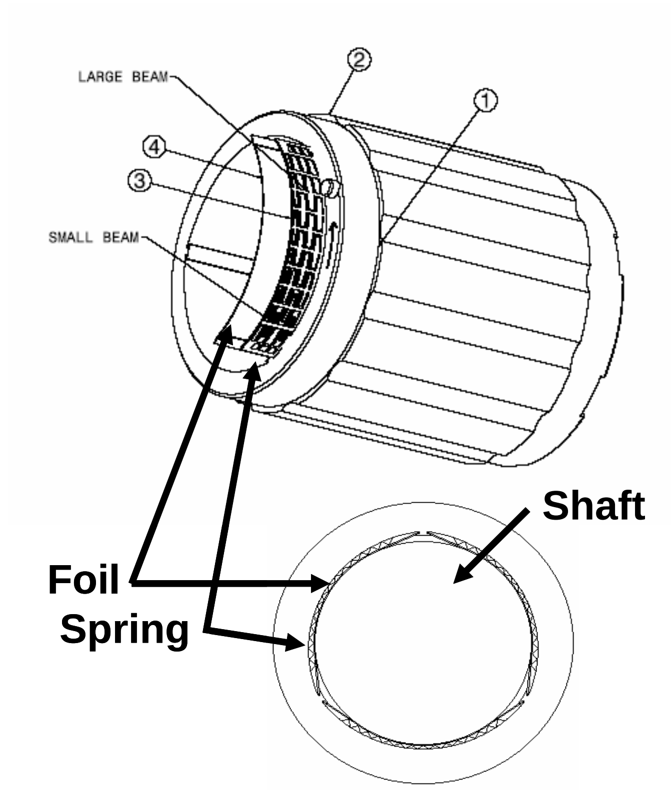
Image source [7-4]



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

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 due to low load capacity
- Thrust disk is larger than bladed wheels



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

Hydrodynamic Oil-Lubricated Bearings

- Thrust or radial bearing
- Most common bearing type
- 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)

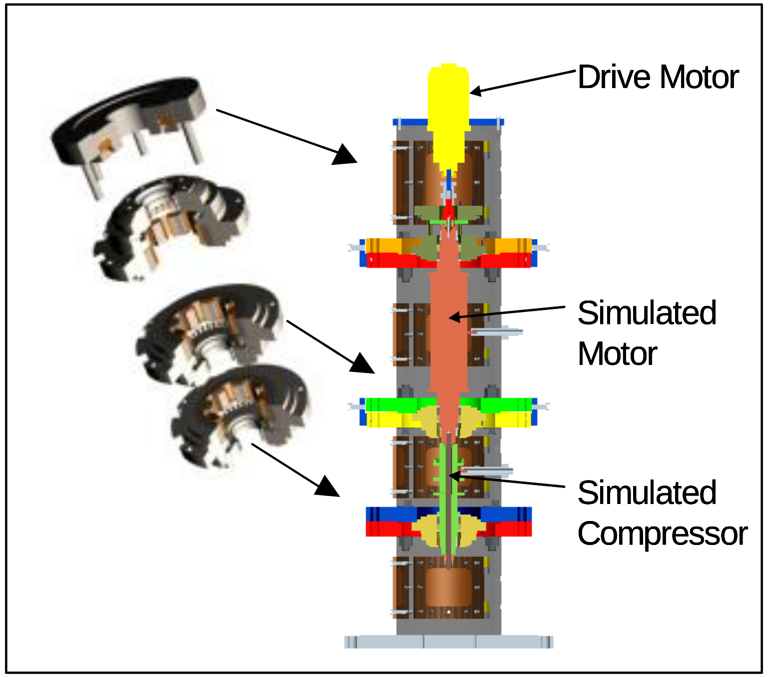
Thrust tilting pad

Image source [7-6]

Radial tilting pad

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

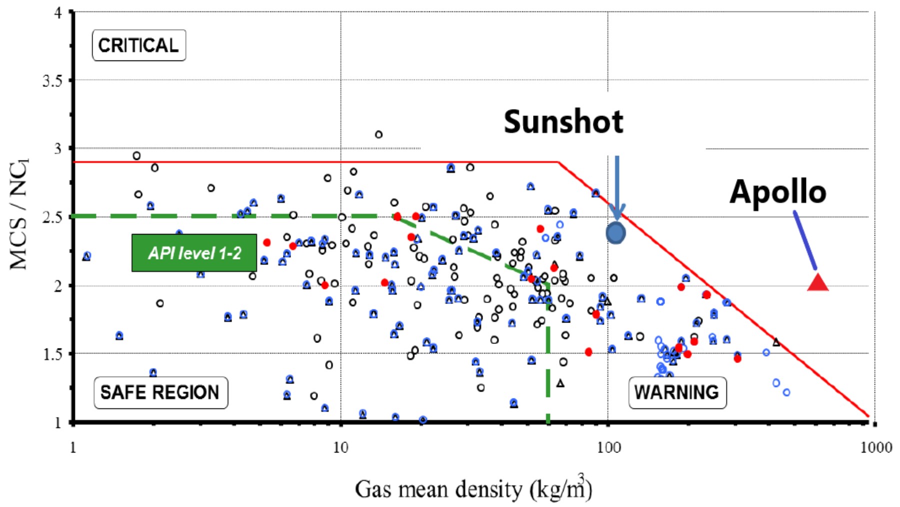


Ransom, D.L., Masala, A., Moore, J.J., Vannini, G., Camatti, M., 2009, **Development of a Vertical High Speed Motor-Compressor Simulator for Rotor Drop onto Auxiliary Bearings,** Presented at the 38th Turbomachinery Symposium, September 2009, Houston, TX.

Rotordynamics

Challenges

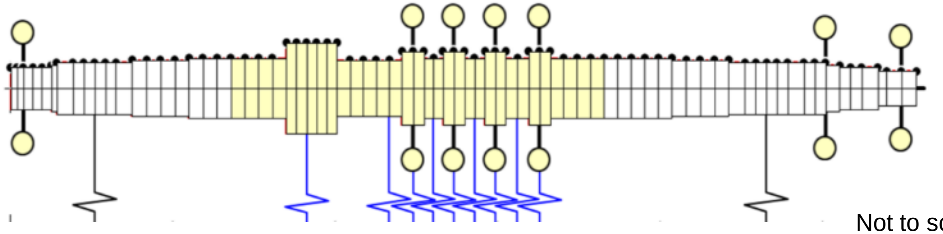
- High gas density
- High operating speed
- Low critical speed (large L/D)
- Similar design methodology as high pressure turbocompressors



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

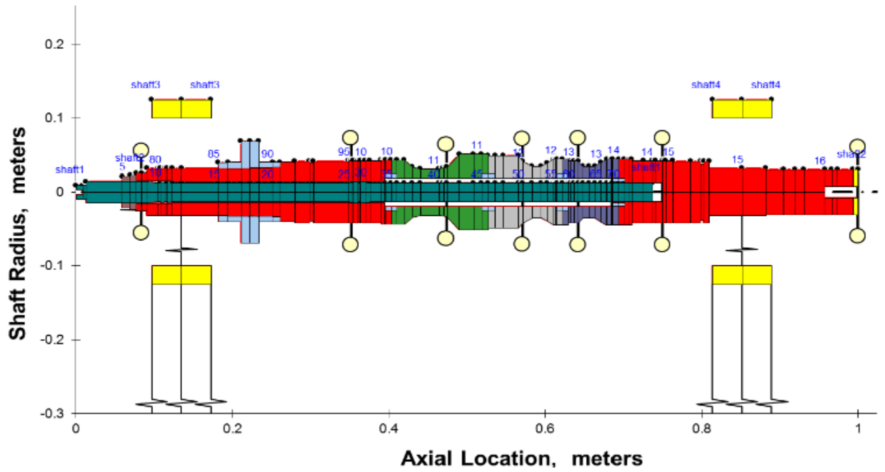
- Texas A&M code
- Perfect gas properties

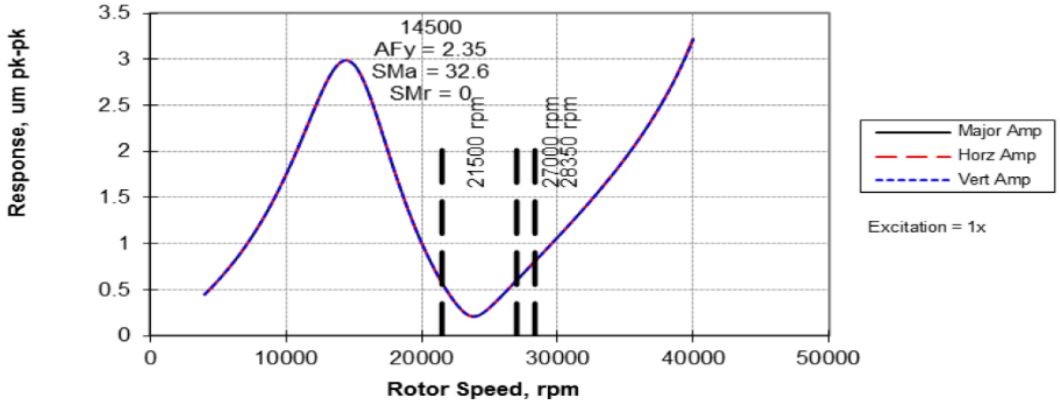
Result

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

SCO2 Compressor Rotordynamic Analysis

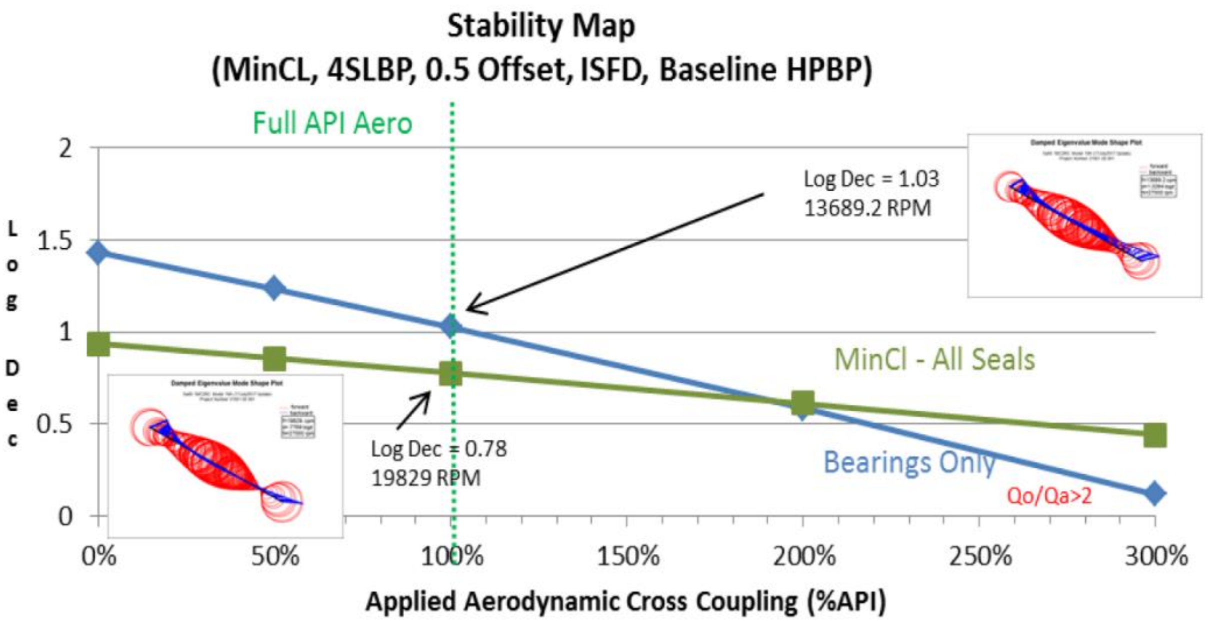
- Final rotor configuration used inboard thrust bearing to provide better control of the bending mode
- Critical speed is well damped with good separation margin from the running speed
- Bending critical well above running speed

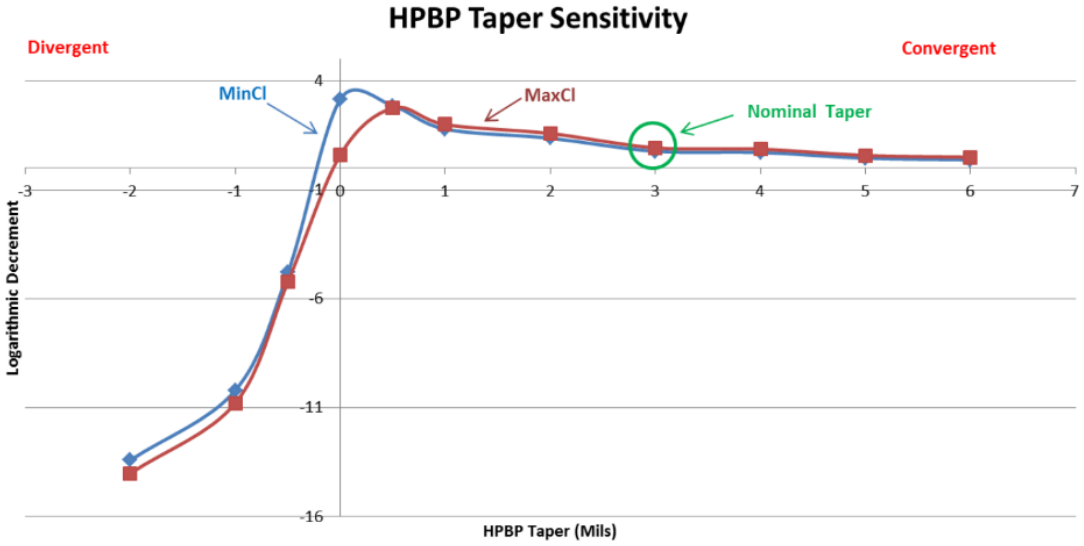


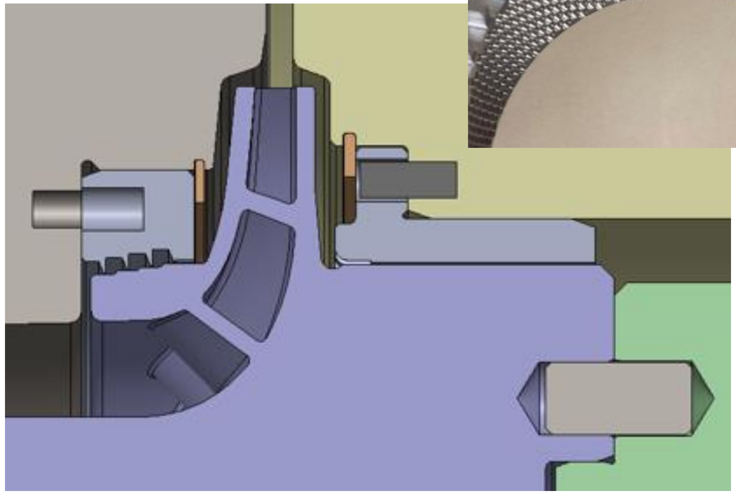


Rotordynamic Analysis

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



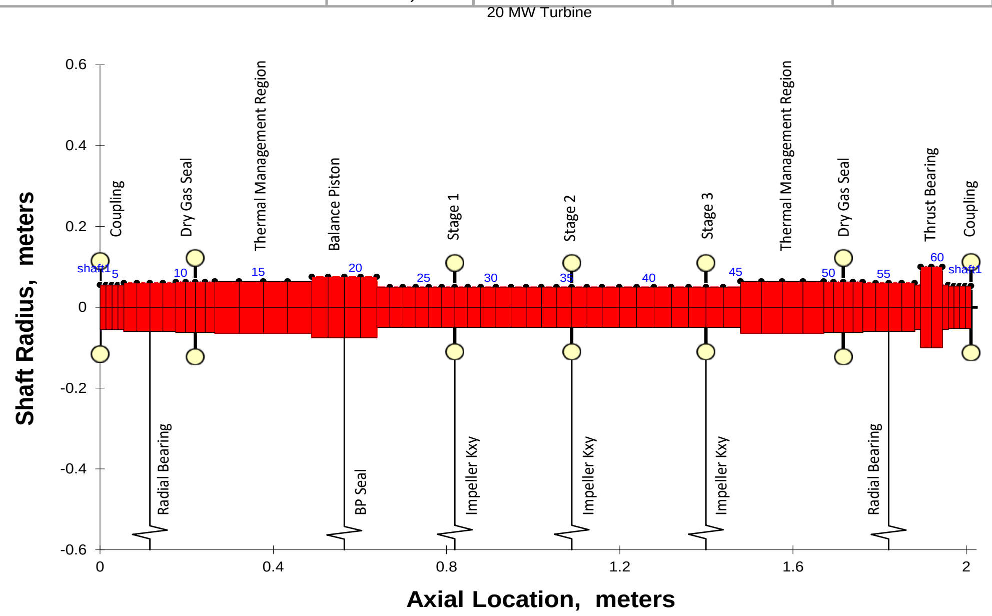




$$K_{xy} = \frac{C_{mr}\,\rho_{dis}\,U^2\,L_{shr}}{Q/Q_{design}}$$

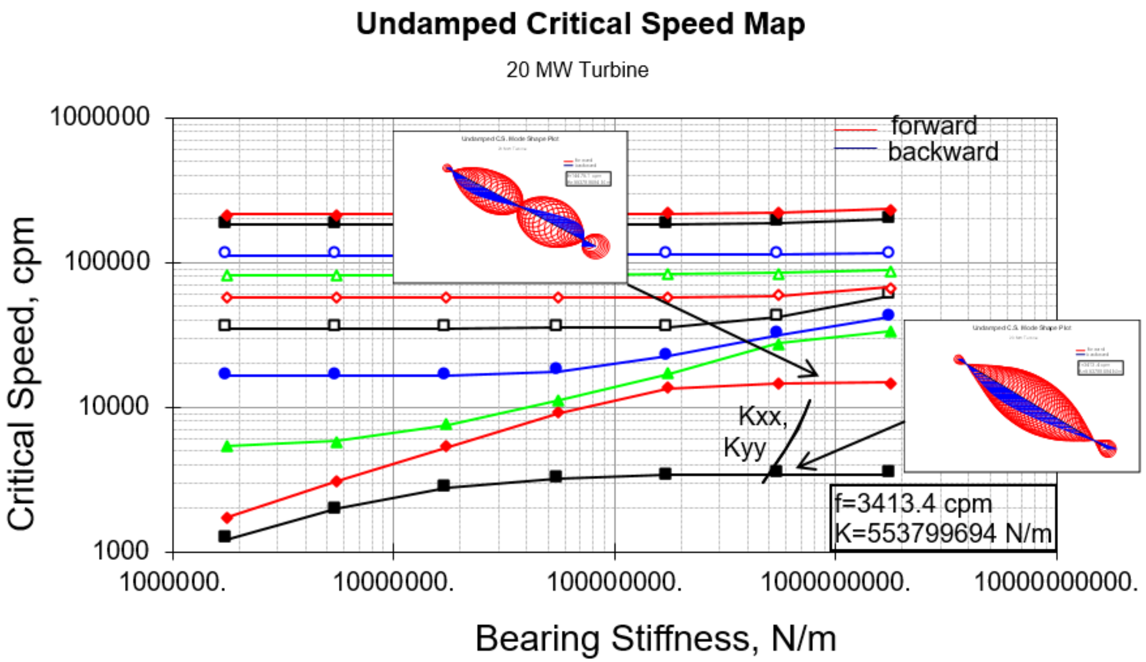
Turbine Design Details for 20 MWe Case Study

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%



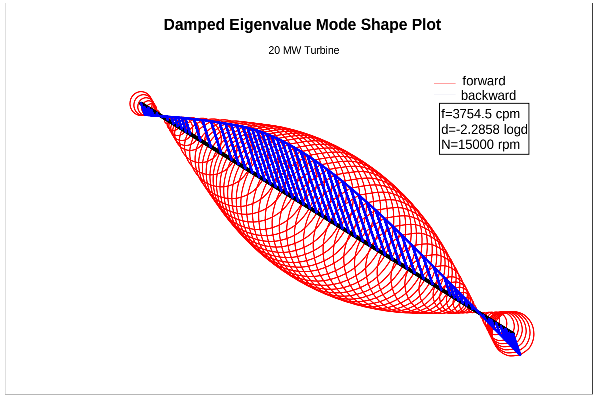
Rotordynamic Results for 20 MWe Case Study

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



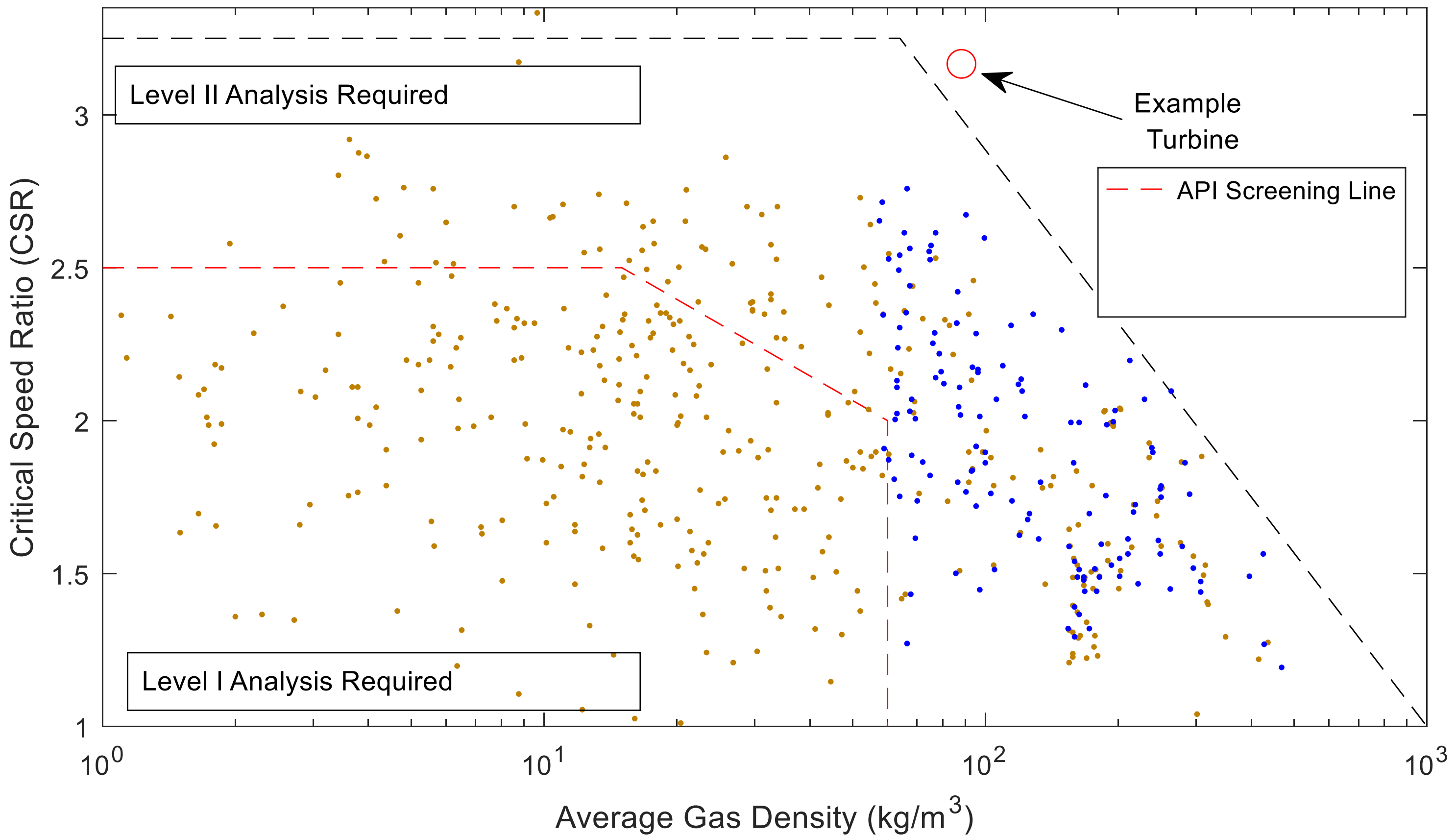
	K_{XY} 1e6 [N/m]
Stage 1	4.02
Stage 2	3.38
Stage 3	2.77

Log Dec = -2.2 without damping devices



Fulton Experience Chart

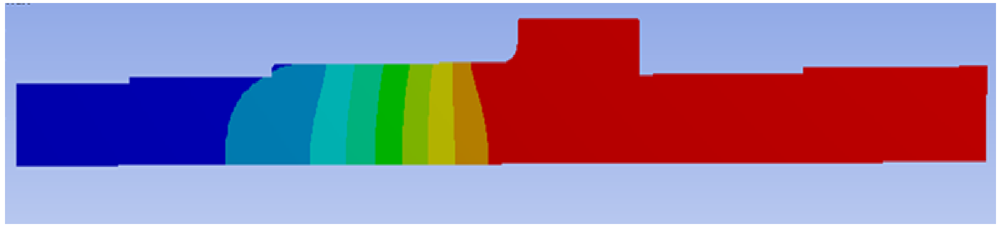
Example Turbine = 20 MW sample turbine



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

Temperature profile

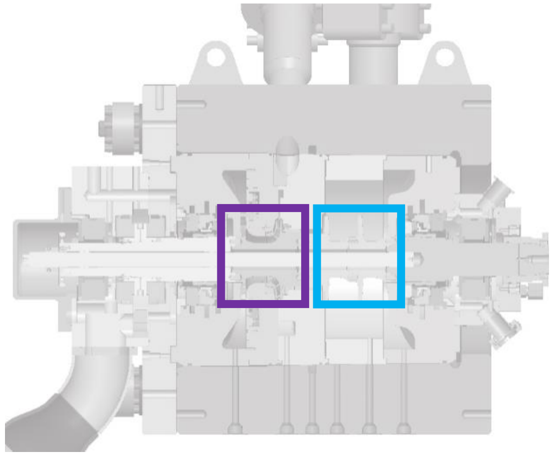


Kalra, C., Hofer, D., Sevincer, E., Moore, J., Brun, K., 2014, "Development of High Efficiency Hot Gas Turbo-Expander for Optimized CSP Supercritical CO_2 Power Block Operation," 4th International Symposium – Supercritical CO_2 Power Cycles, Sept. 9-10, 2014, Pittsburgh, PA

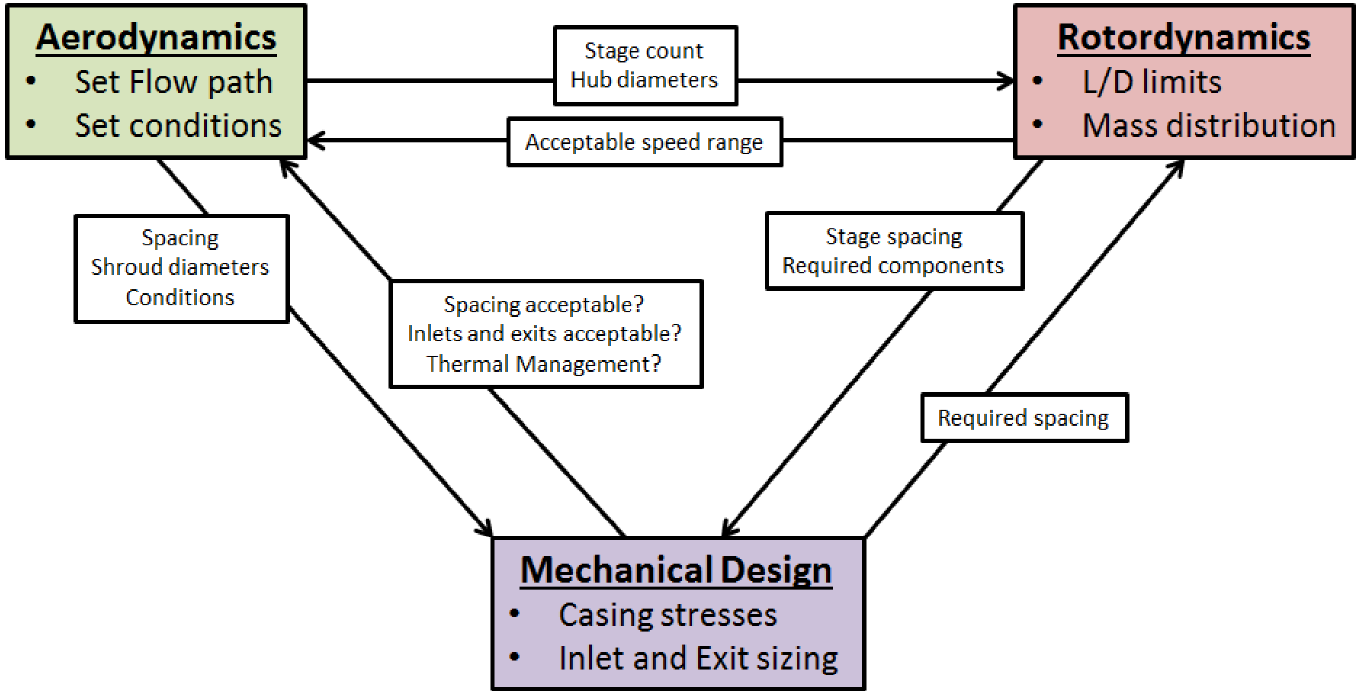
- Large thermal gradient coupled to pressure containment including transients is challenging
- May result in life limited designs due to LCF and creep

Example SCO2 Compressor

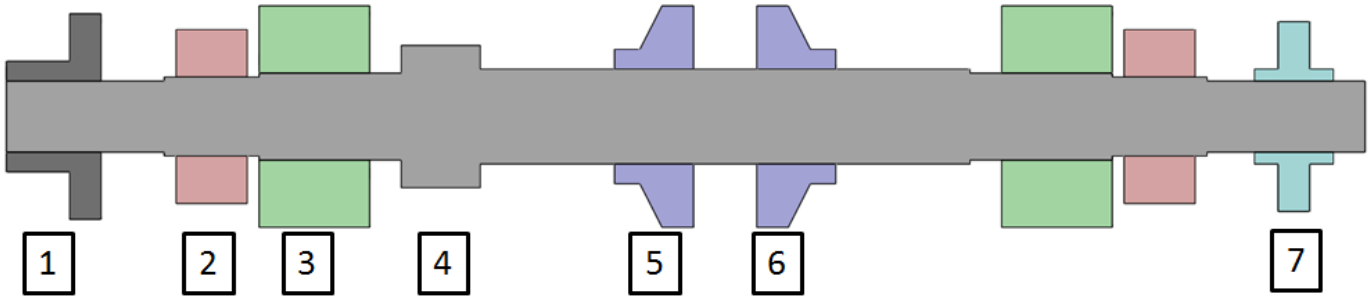
- Design a sCO_2 compressor for use in a closed loop Recompression Brayton Cycle (RCBC) to achieve thermal efficiencies > 50% with both **MAIN** and **BYPASS** compressor
- Compressor must handle suction temperatures from 68 to 122°C and pressures from 1,090 to 1,540 psia to maintain near steady mass-flow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



Example SCO2 Compressor



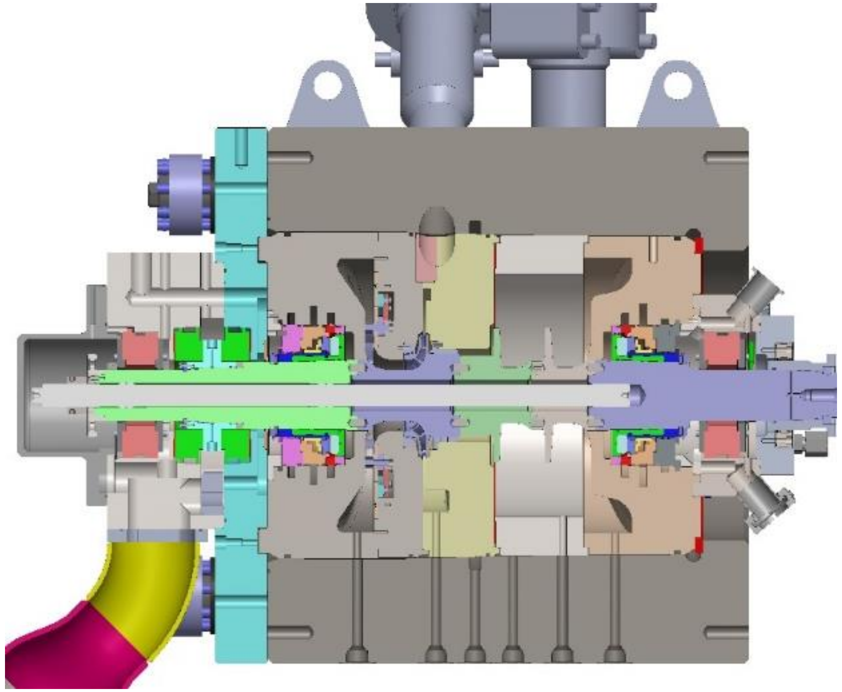
Example SCO2 Compressor



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 (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)

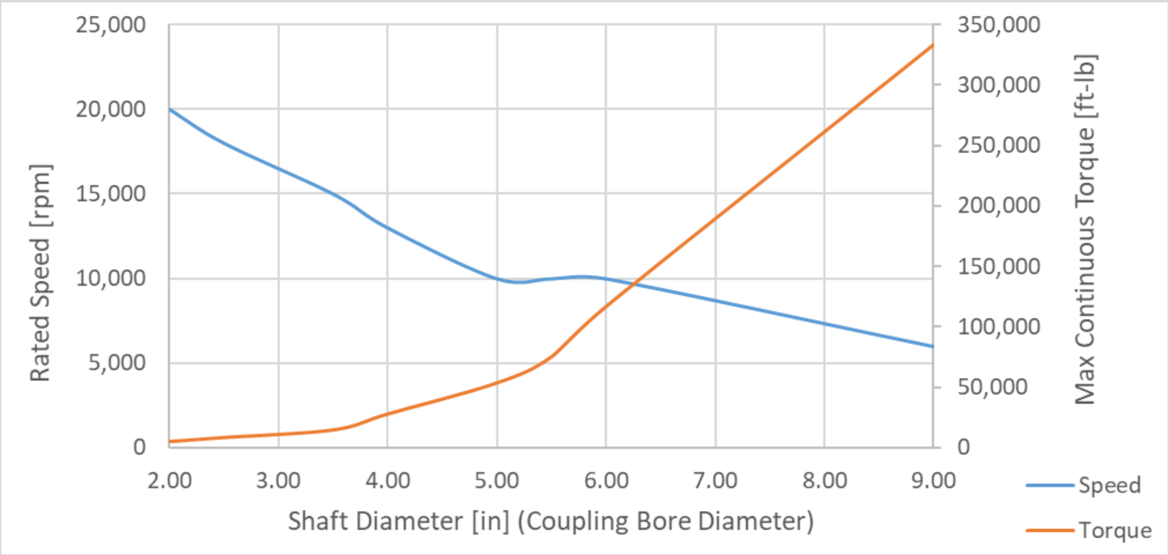


Cich, S., Moore, J., Kulhanek, C., Mortzheim, J., 2020, "Development and Testing of a Supercritical CO2 Compressor Operating Near the Dome, Proceedings of 49th Turbomachinery Symposium, Houston, TX, September 2020, Turbomachinery Laboratory, Texas A & M University

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

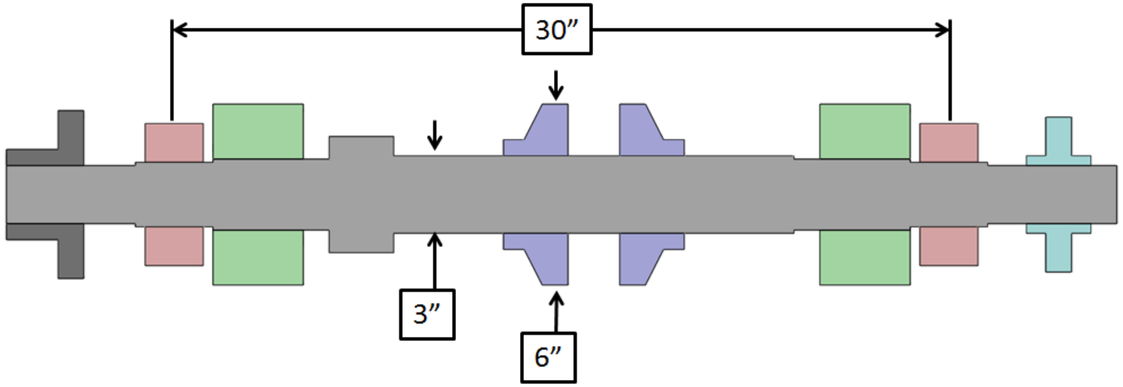
[3]



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

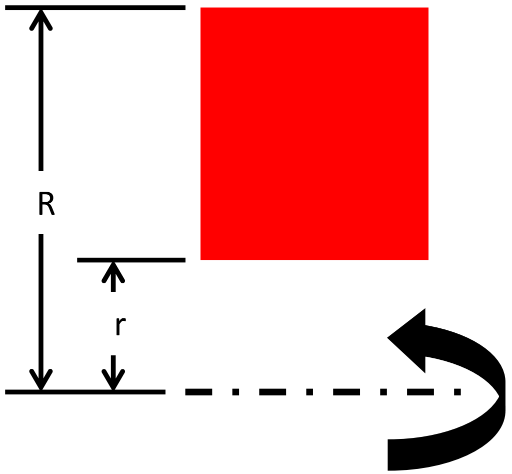
Mechanical Design

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



Mechanical Design

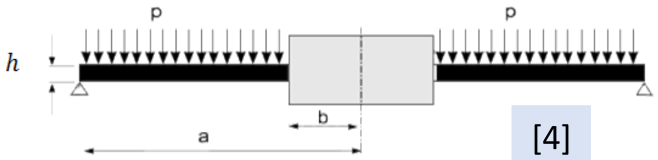
- Larger diameters → More head, more rotating weight, higher hub stress
- Smaller Inner Diameters → reduced tie bolt diameter, more rotating weight, reduced stresses
- Solid shafts have the lowest stresses but required monolithic shafts or more complicated joints
- Aero hub diameter may be smaller than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

- 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

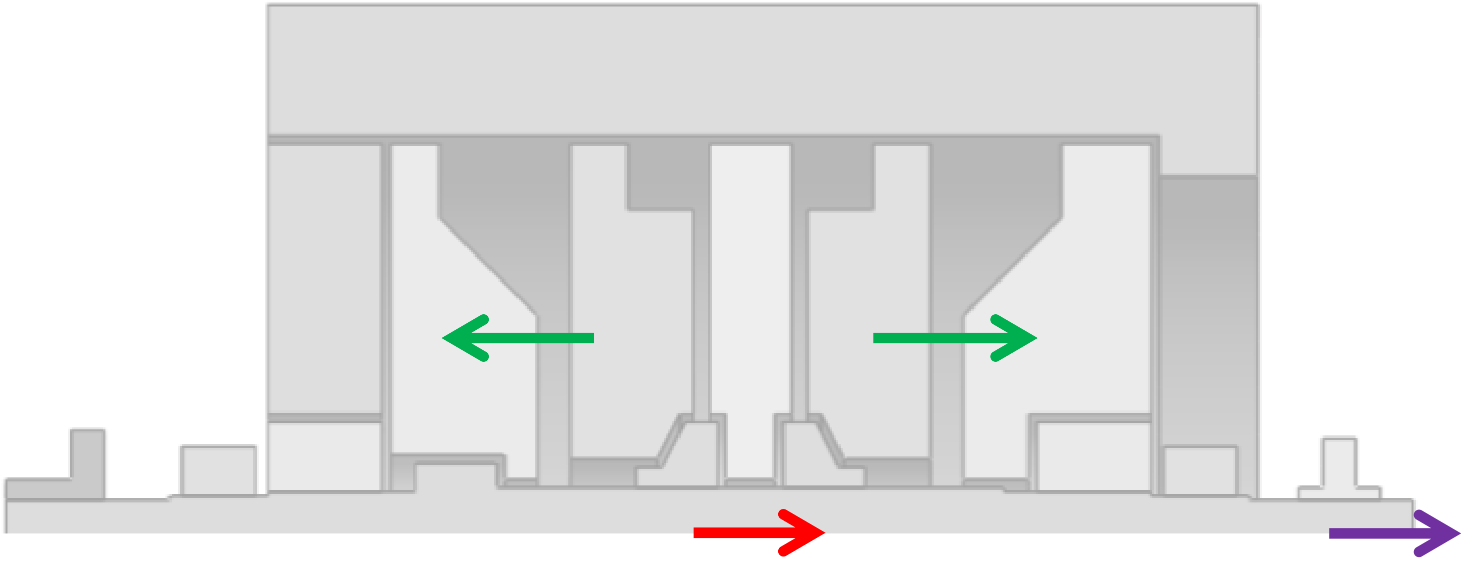


[4]

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 diaph 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

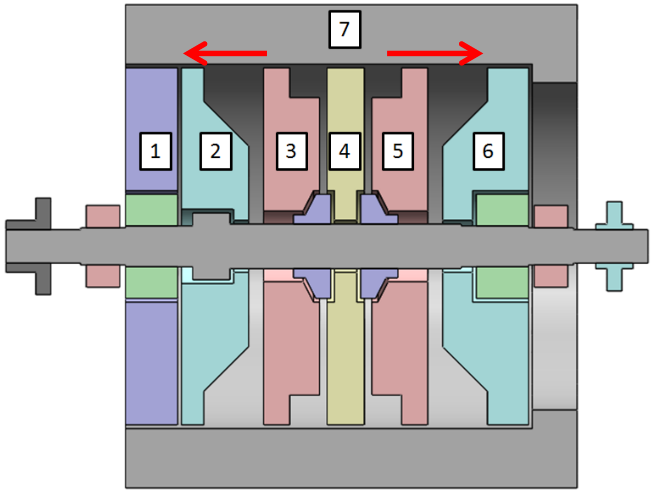
Housings and Heads Deflect Away from High Pressure – Opening of Diffuser

Housing Deflection Pulls Rotor at Thrust Bearing

Impeller Move to the Right

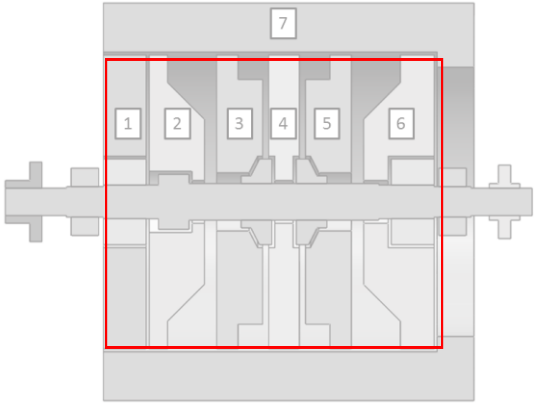
Mechanical Design

- 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 seals)
    - Pilot fits



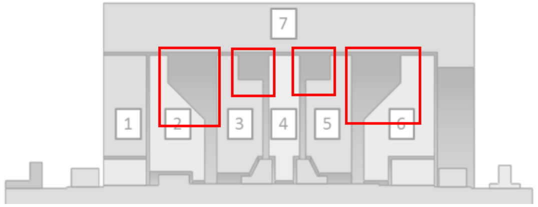
Mechanical Design

- 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



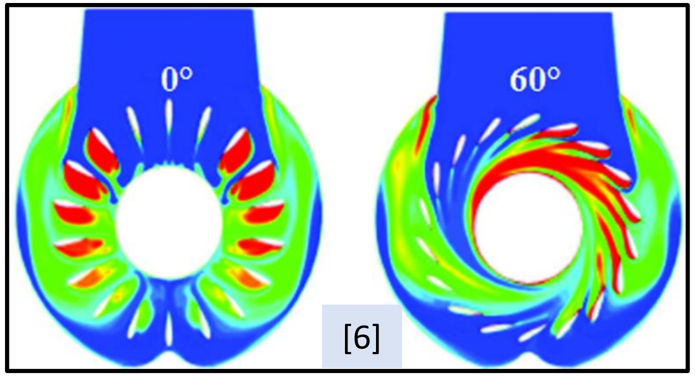
- Flow path Sizing
- Lower velocity limits → larger flow areas → Increase bundle diameter (or case complexity) → reduced pressure losses
- Higher velocity limits → smaller flow areas → Reduced bundle diameter → erosion concerns & increased pressure loss

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

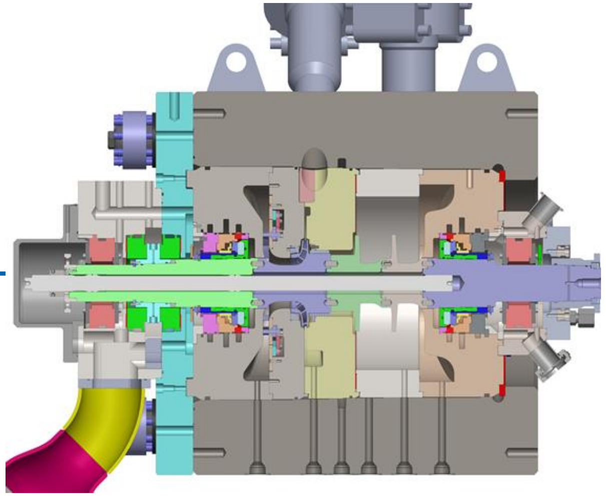
- How to manage large swings in suction density?
- Case Treatment (Passive)
    - Internal recirculation to maintain discharge flow
- Chilling / Heating (Active with Increased Power)
    - Reduce or Increase fluid density to match design
- Variable Inlet Guide Vanes (Active / Control Logic)
    - Increasing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

[5]



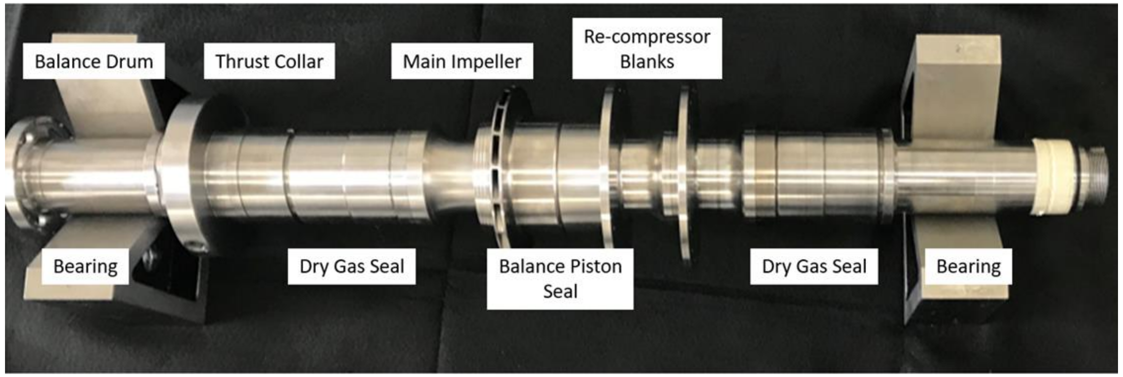
[6]

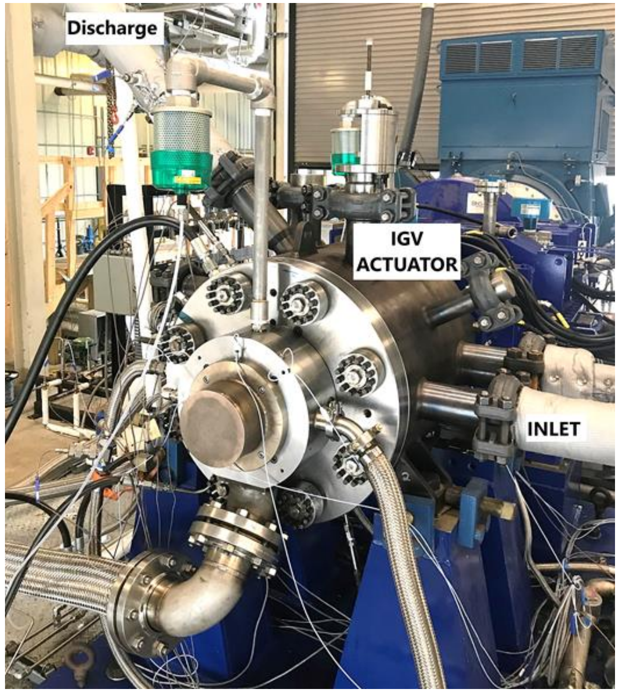
Final Embodiment – SwRI/GE Apollo Compressor



- 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)





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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