Tutorial: Turbo Machinery Design for Supercritical CO2 Applications

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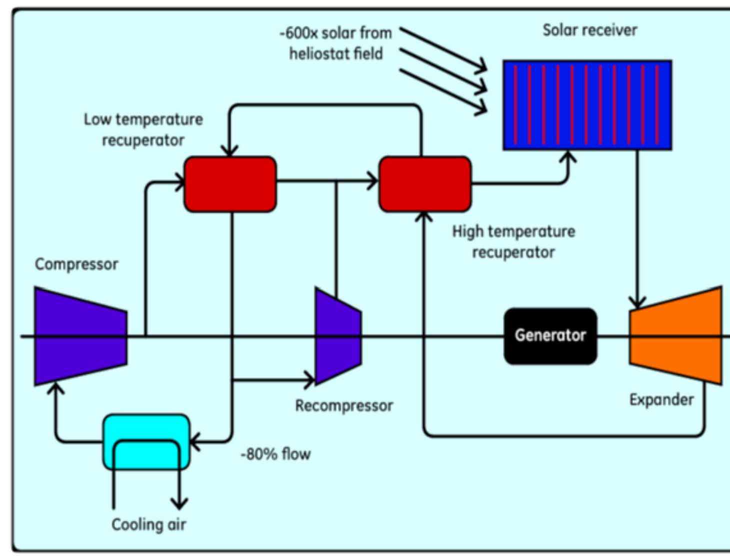
Robert Fuller

Barber Nichols Inc.



The 4th International Symposium
Supercritical CO_2 Power Cycles
September 9-10, 2014,
Pittsburgh, Pennsylvania

Outline

- Pumps/Compressor/Turbine Aero Designs
- Seals and Seal design – damper seals, dry gas seals
- Bearings: Gas and Oil Hydrodynamics
- Rotordynamics
- Blade Loading and Dynamics
- Materials For CO2

Outline Cont.

- Pressure containment
- Thermal management – Blade cooling, rotor, casing, dry gas seals
- Test Loop Design

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Supercritical CO2 Cycles Pumps/Compressors/Turbines Aero Design

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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

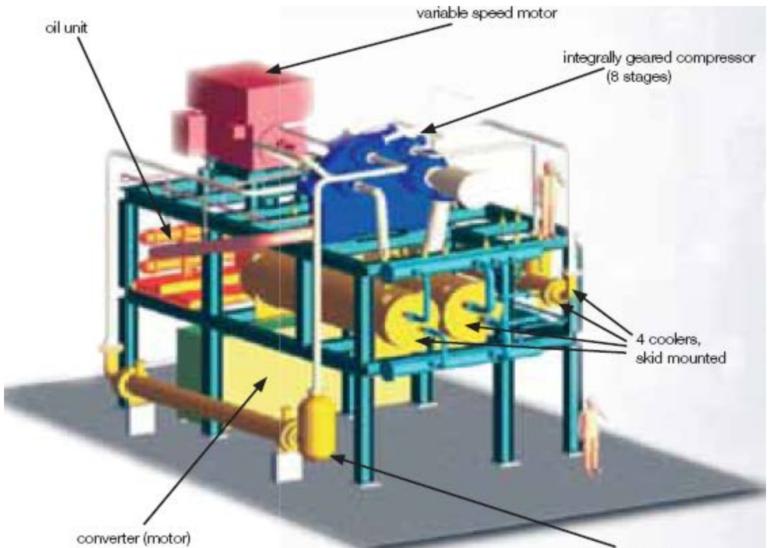
Centrifugal Pumps

Multi-Stage Typical, Motor Driven

- Low Speed CO2 Pumps, Standard Manufacture
  - Ruhrpumpen
  - Flowserve
  - Sulzer
  - Wood Group (GE)
  - Schlumberger

Centrifugal Compressor Options

Integrally Geared Isothermal Compressor

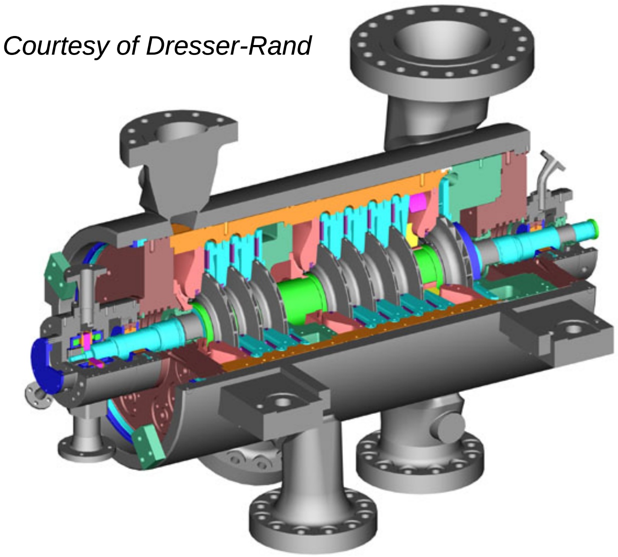


Courtesy of MAN

- Integrally geared can achieve near isothermal compression
- Can contain up to 12 bearings, 10 gas seals plus gearbox
- Impellers spin at different rates
  - Maintain optimum flow coef.

Single-Shaft Multi-stage Centrifugal Compressor

Courtesy of Dresser-Rand



- Multi-stage centrifugal proven reliable and used in many critical service applications currently (oil refining, high pressure CO2 re-injection, etc.)
- Fewer bearings and seals
  - (4 brgs & seals for 2 body train)
- Can be direct driven by sCO2 turbine

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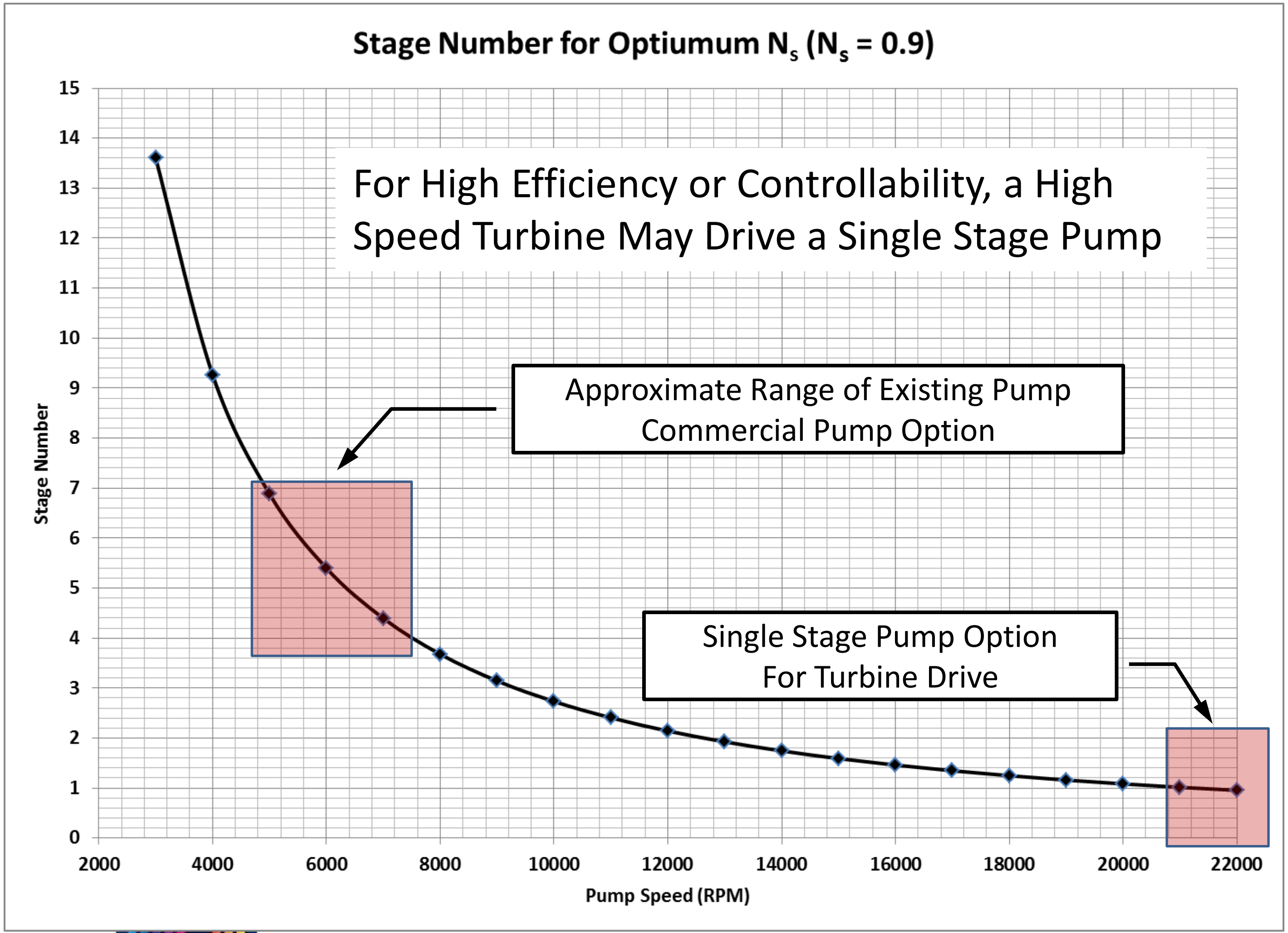
Optimum Single Stage Pump Requires N = 22000 rpm, 2.4MW CO2 Pump



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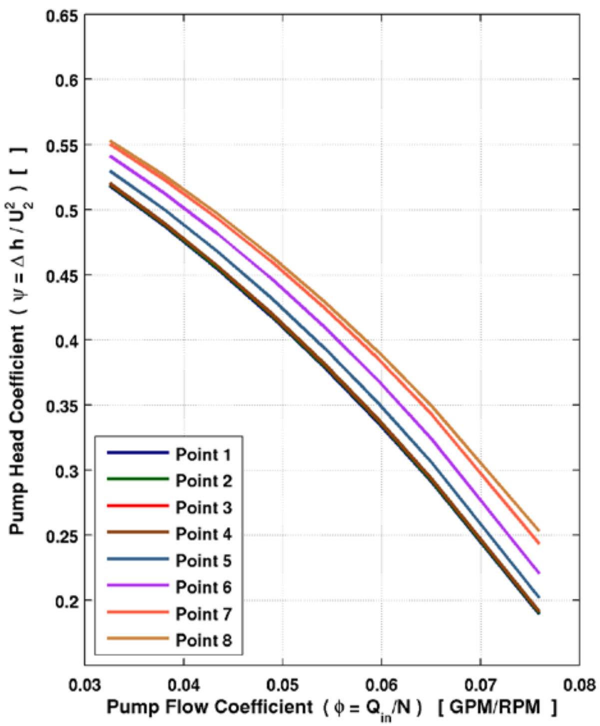
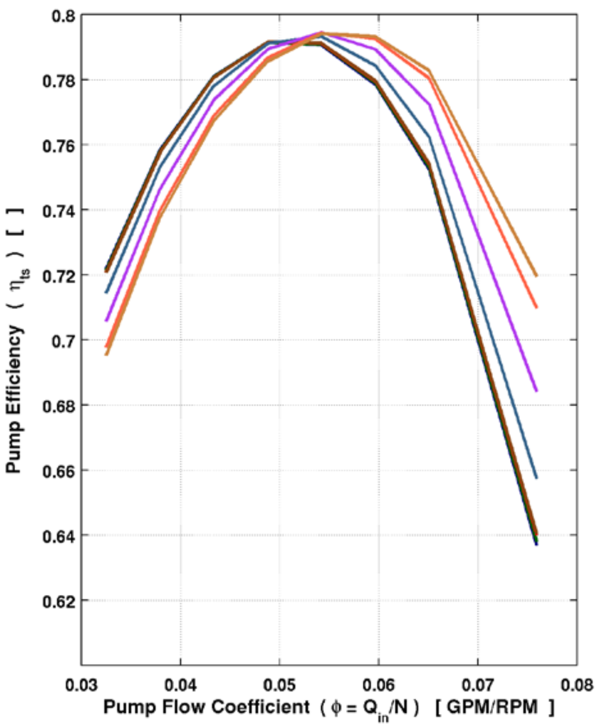
High Speed Pump Head and Flow Coefficients for CO_2
Over Wide Inlet Condition Range
Changes Due to Density Range





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Turbomachinery Elements-Main Compressor

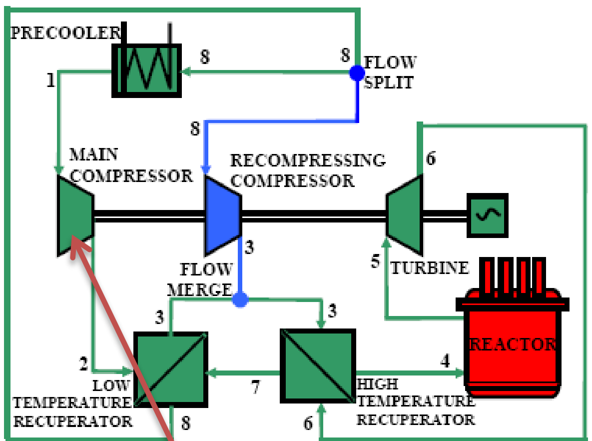
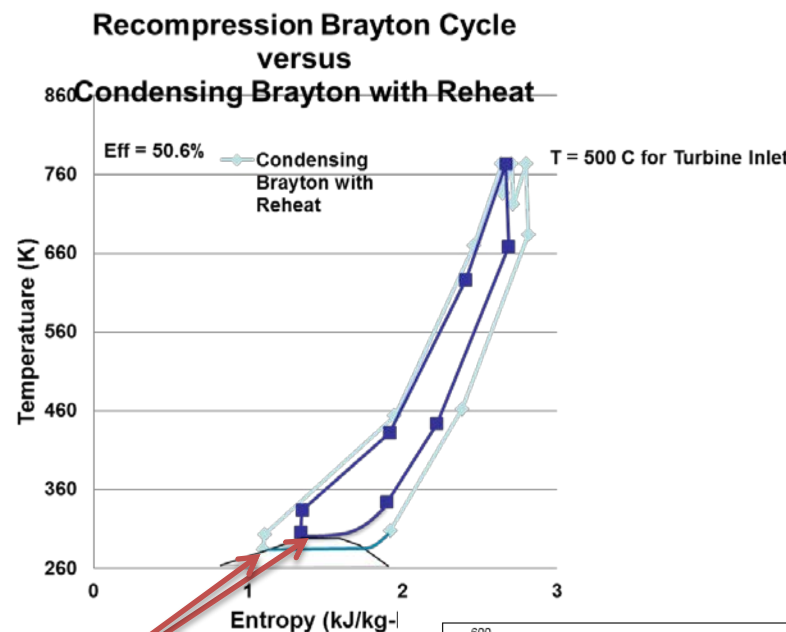


Figure 6.1 Recompression Brayton cycle layout



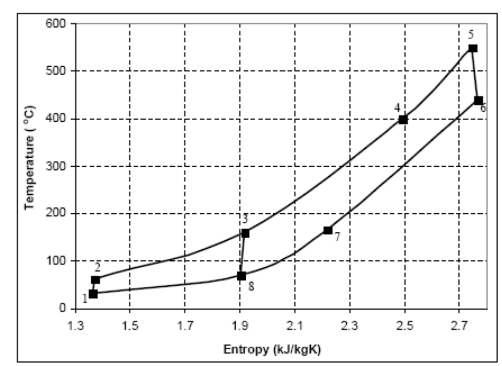


Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle

Main Compressor/Pump-Saturated Liquid or Vapor Inlet

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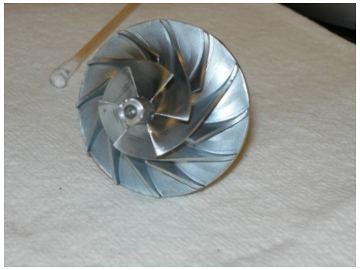
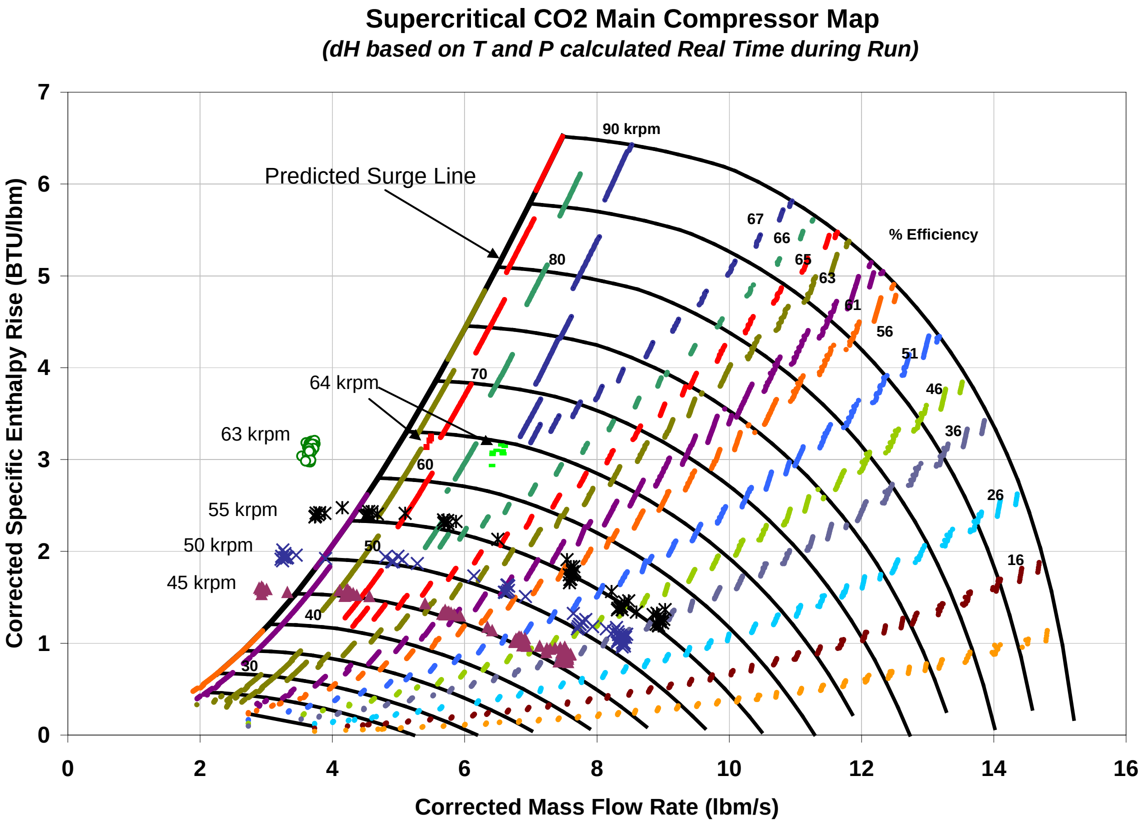
Main Compressor Example

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



37mm
Wheel
Diameter
30-50 BAR Pressure Rise



Turbomachinery Elements-Re-Compressor (Available Commercially at 1 MW and Larger

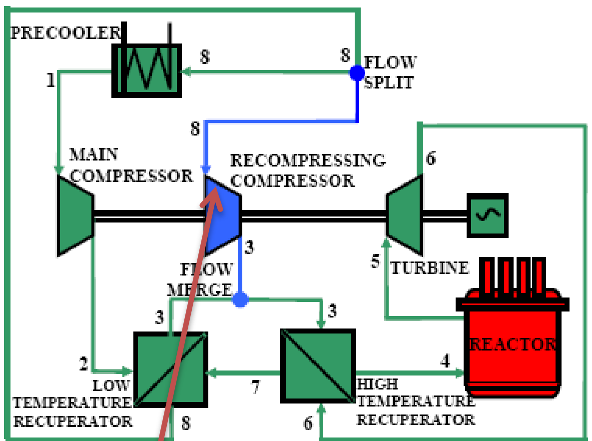


Figure 6.1 Recompression Brayton cycle layout

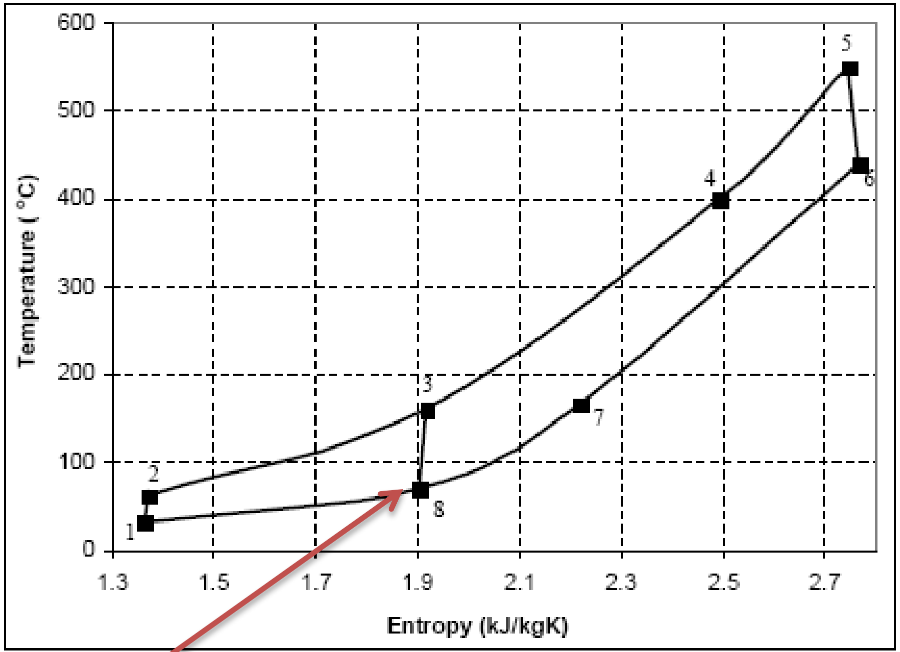


Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle

Re-compressor- Warm Gas Inlet

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Turbomachinery Elements-Turbine

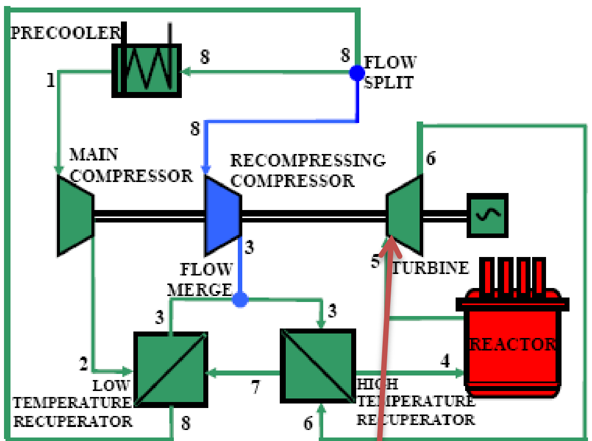


Figure 6.1 Recompression Brayton cycle layout

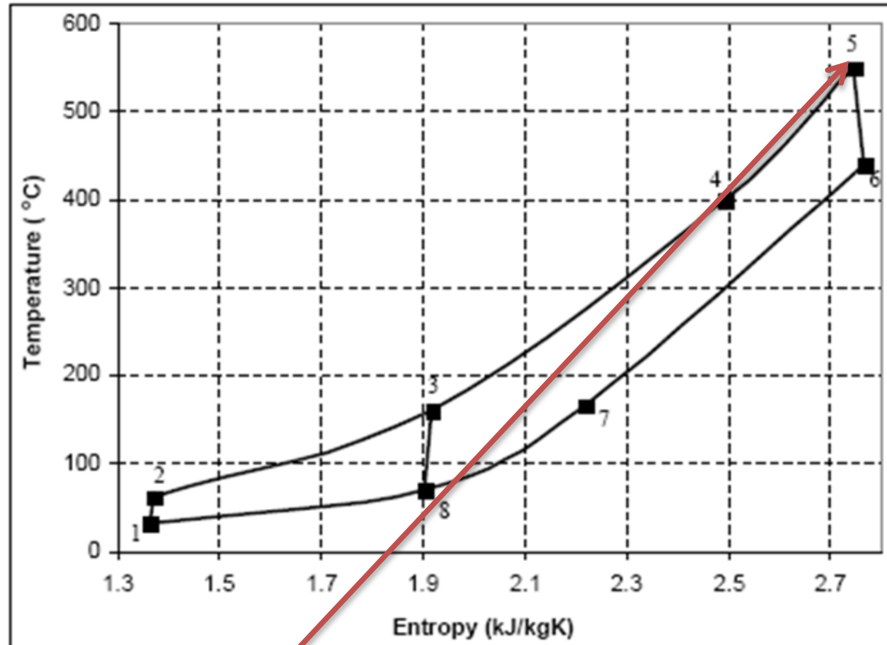
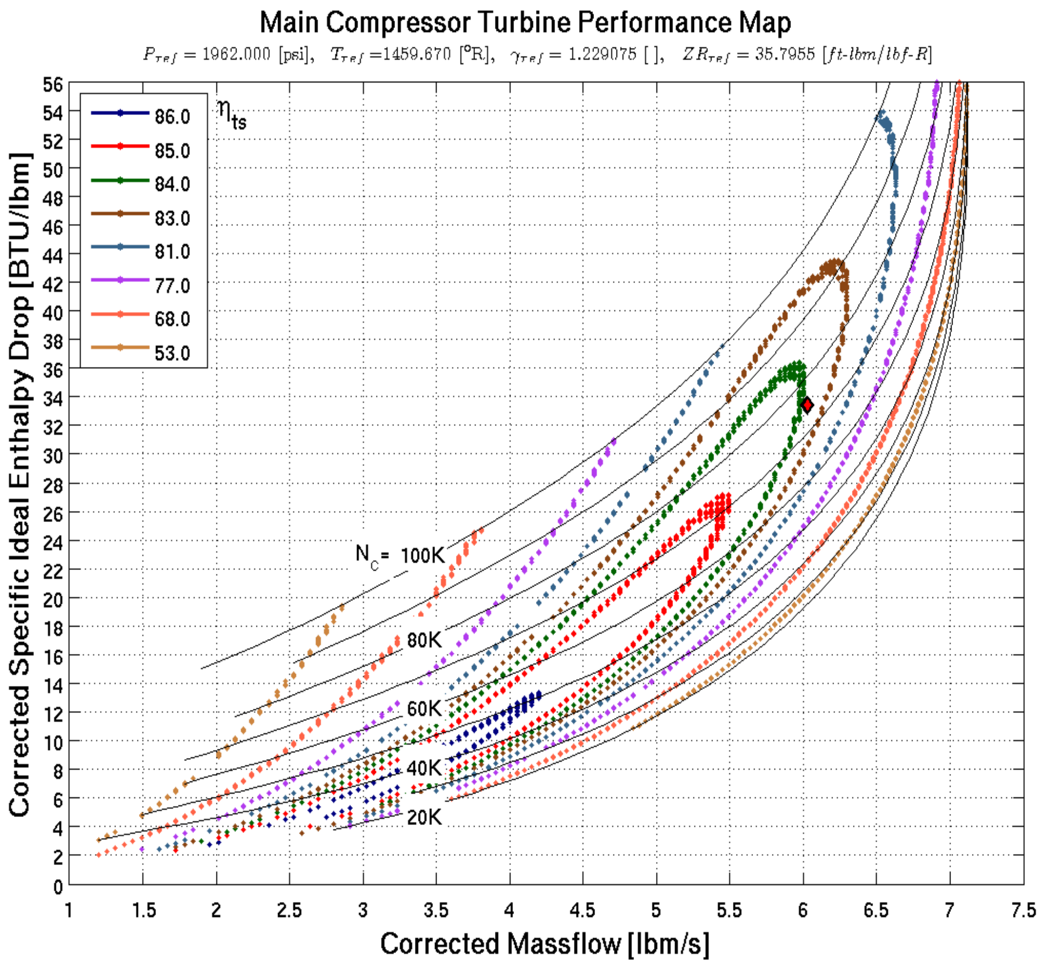


Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle

Turbine-Hot Gas Inlet

Turbine Performance Map



Seals: Internal and Shaft End

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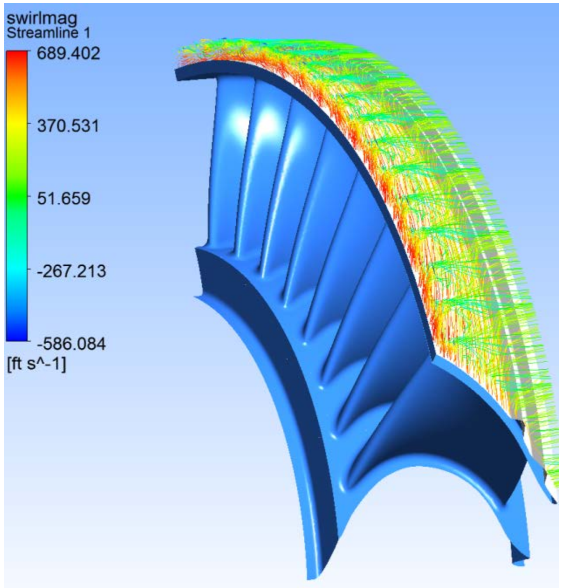
Seals

Labyrinth:

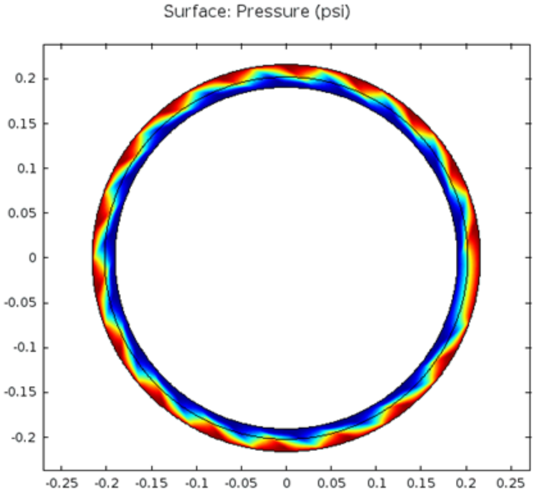
- Labyrinth seals used at blade tip and interstage locations
- Swirl brakes used to minimize swirl entering seal
    - CFD used to optimize and evaluate swirl brake performance



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

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Annular Gas Seals in Compressor

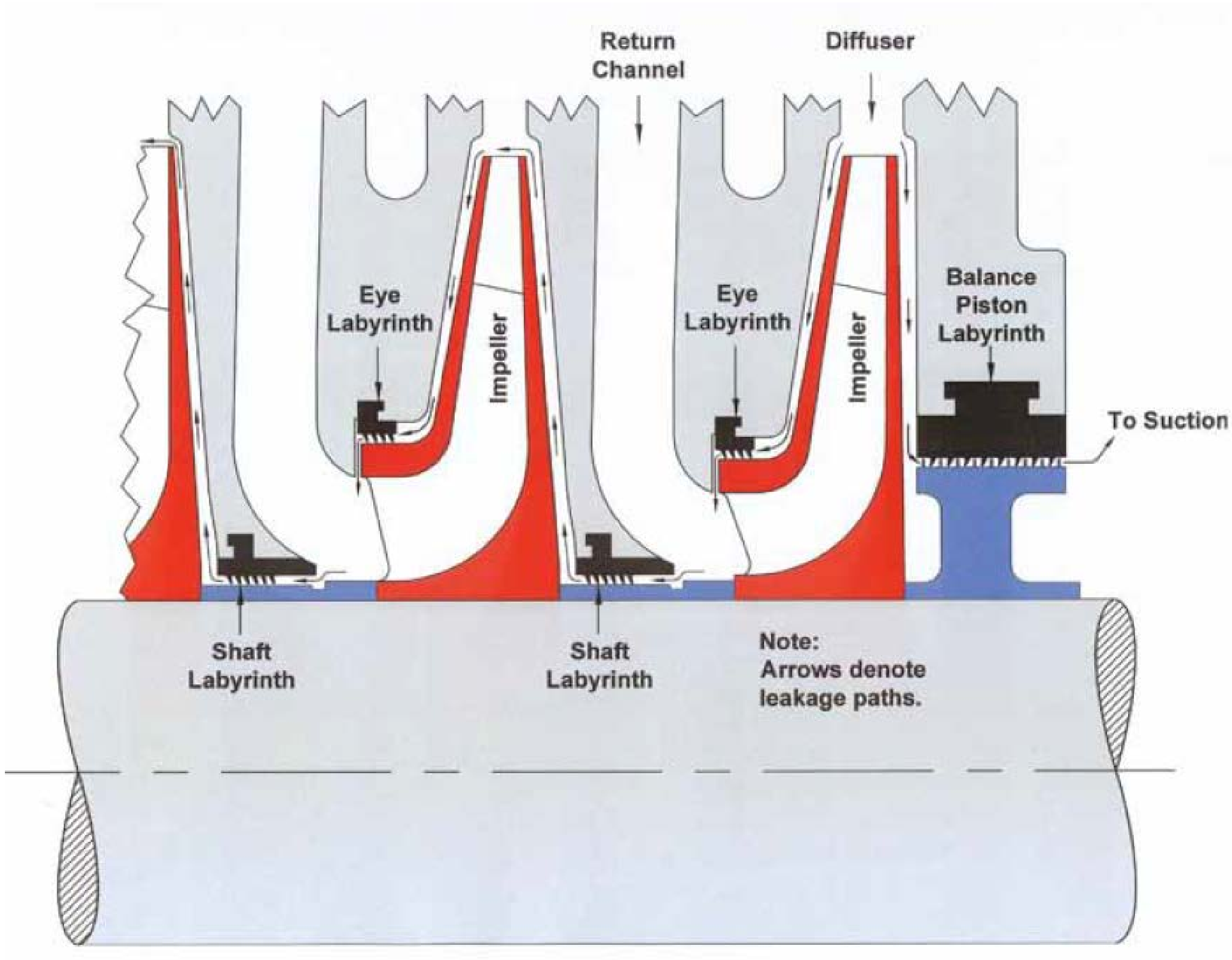
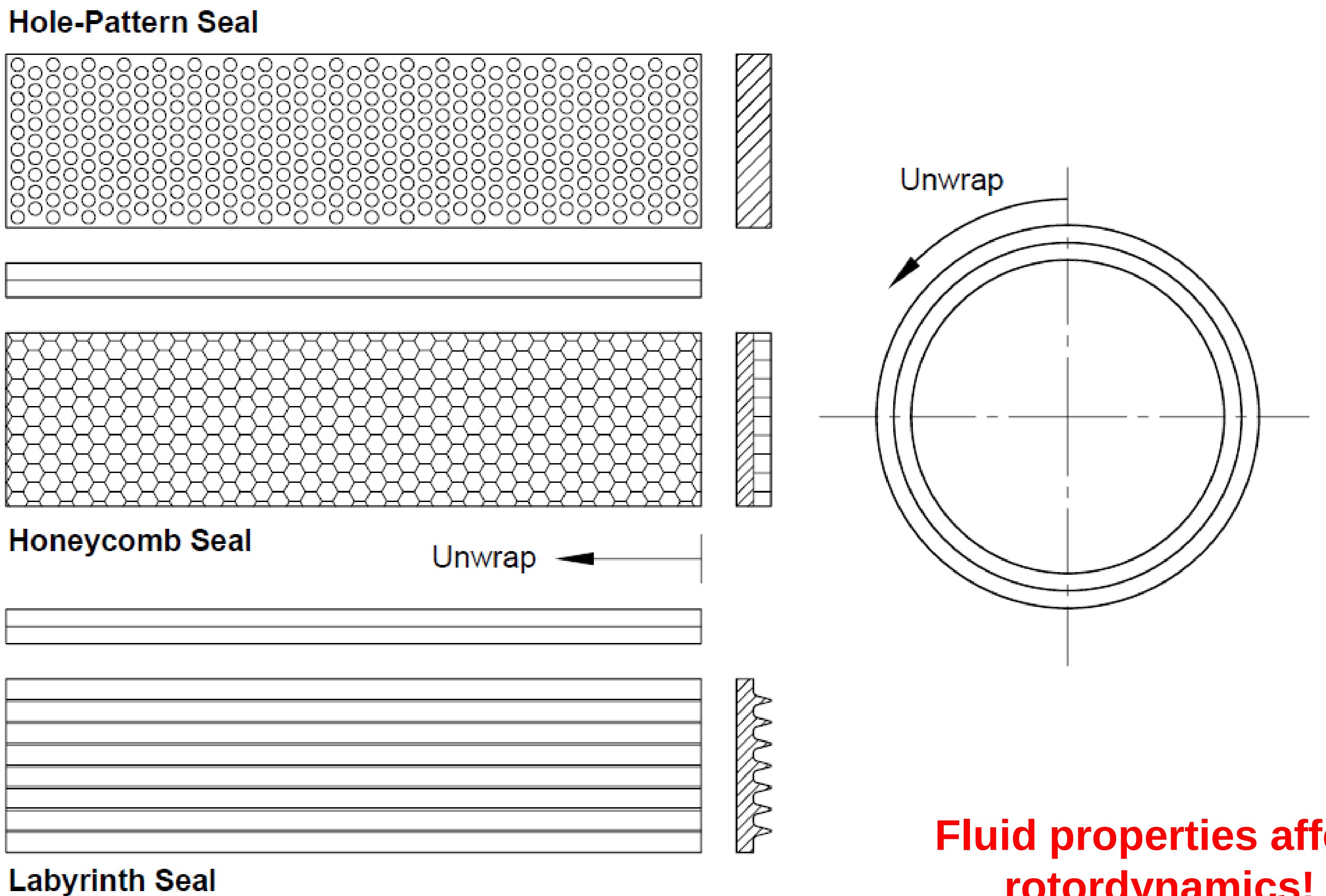


Image source [7-1]

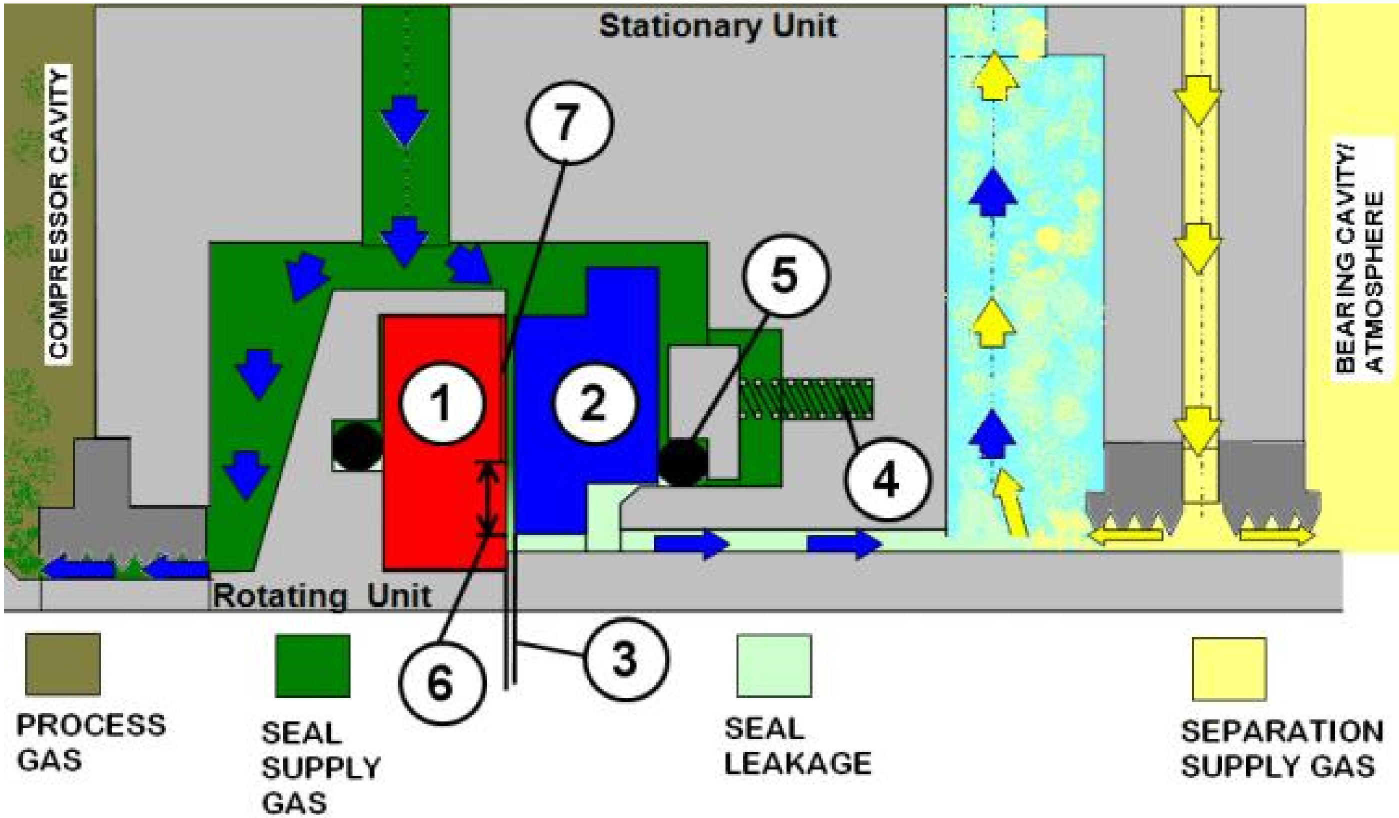
Different Seal Geometries



Fluid properties affect rotordynamics!

Image source [7-2]

Dry Gas Seals



1 Rotating Seat

2 Stationary Face

3 Gap/ Gas Film

4 Springs

5 Secondary Dynamic Sealing Element

6 Sealing Dam

7 Profiled Surface

Rotating seal surface…

Image source [7-4]

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Bearings

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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

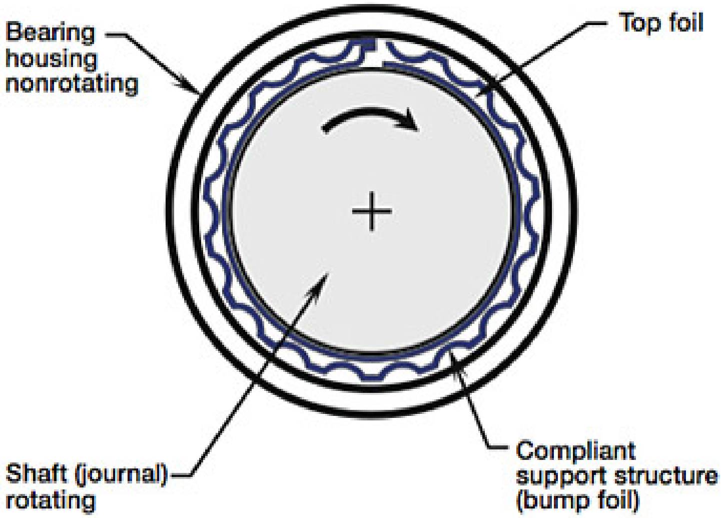
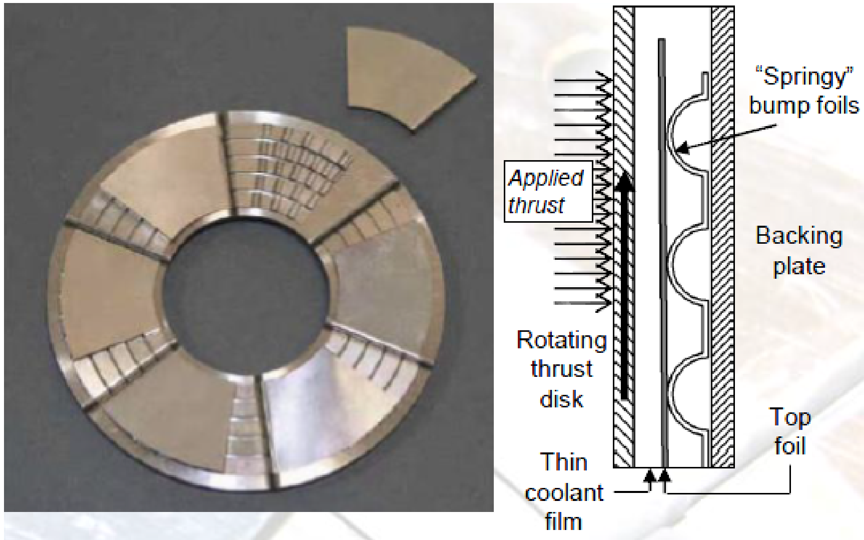


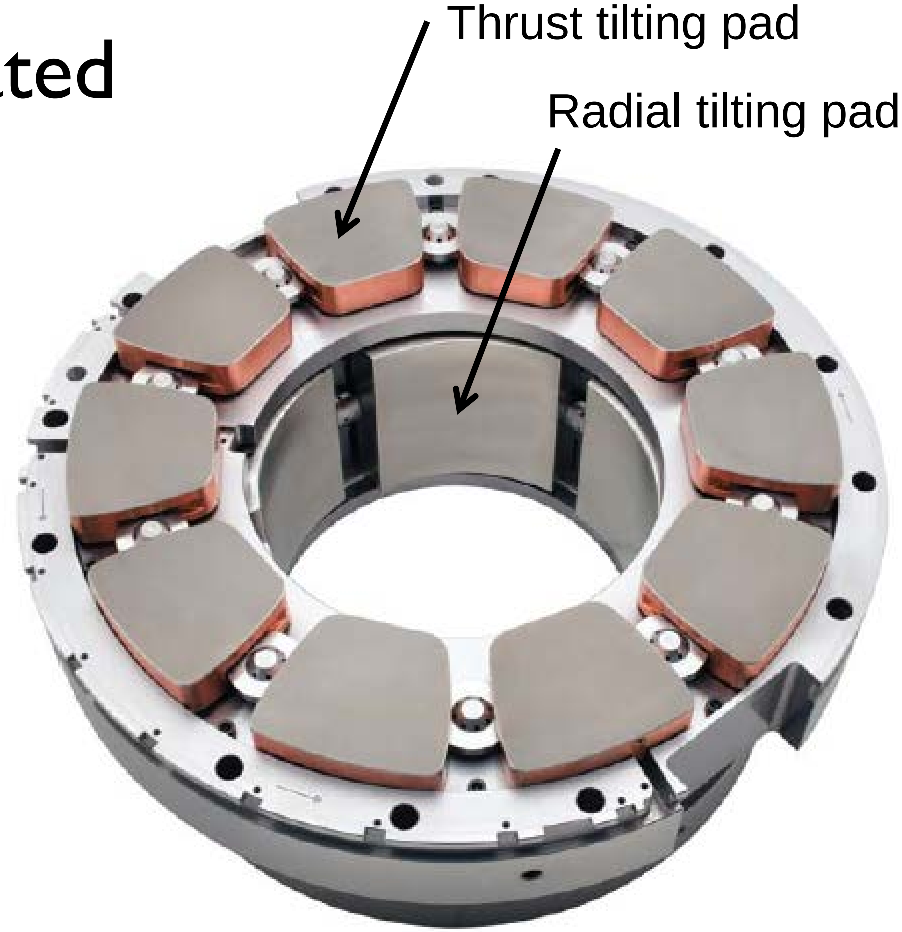
Image source [7-5]



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)



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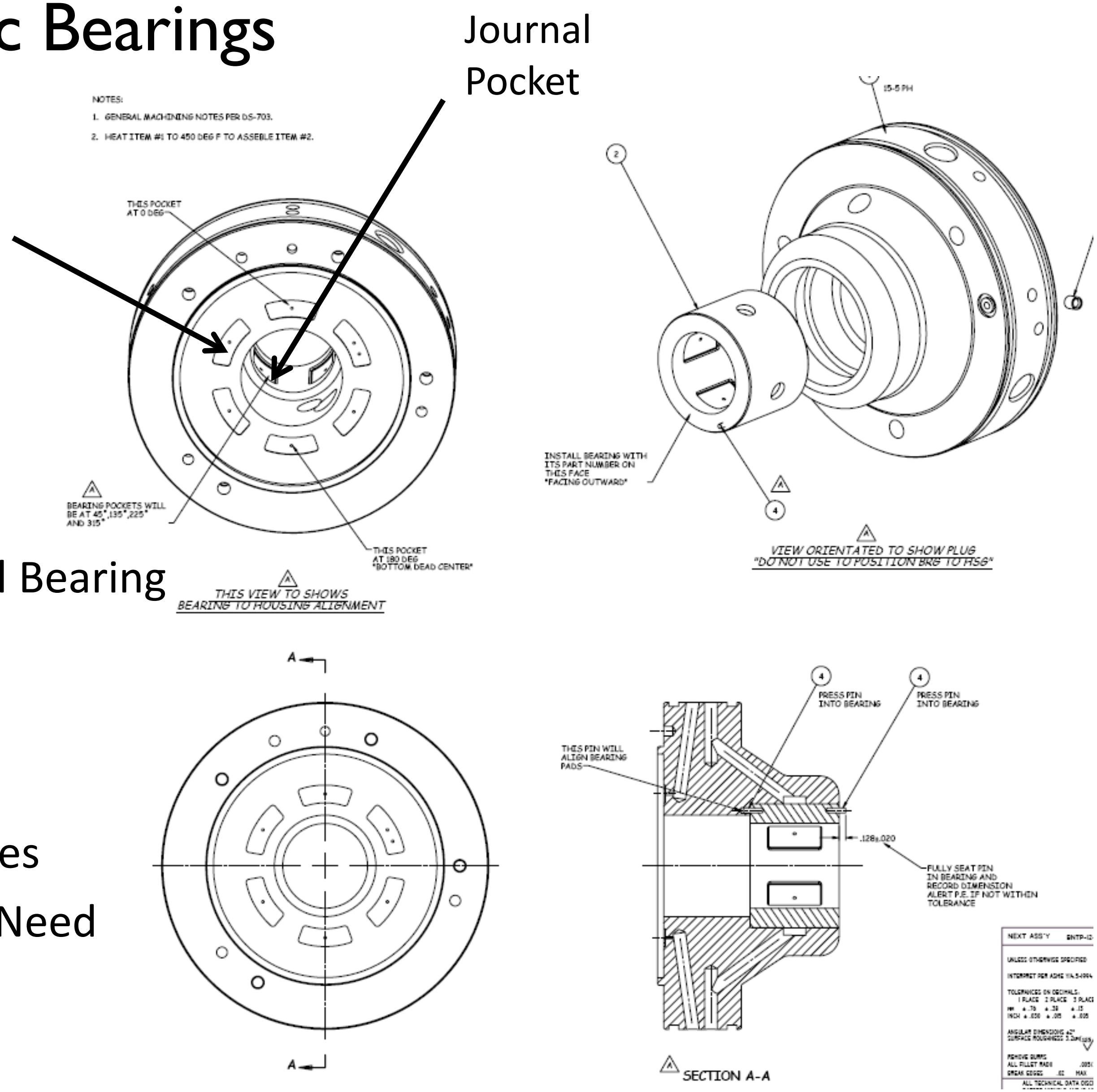
CO2 Hydrostatic Bearings



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

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Rotordynamics

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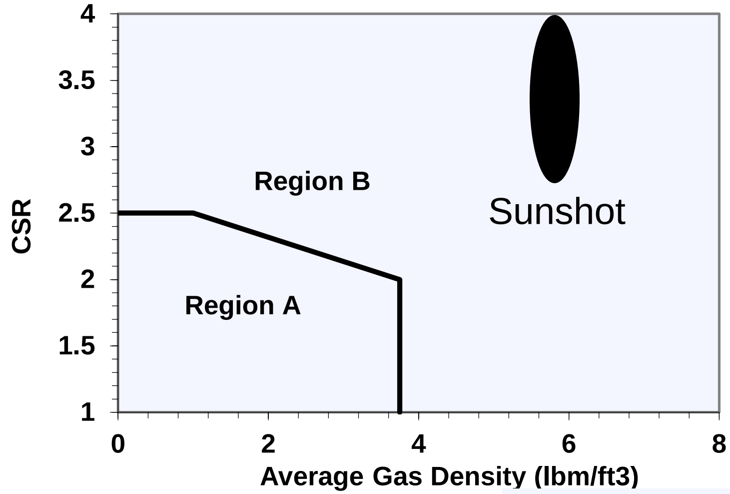
Rotordynamics

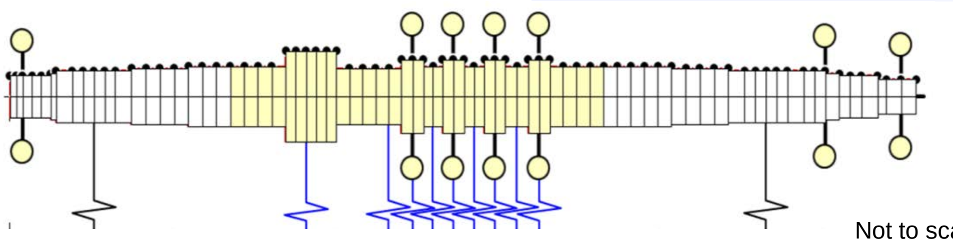
Challenges
- High gas density
- High operating speed
- Low critical speed (large L/D)
- Similar design methodology as high pressure turbocompressors

Interstage laby seals
- Texas A&M XLTRC code
- Real gas CO2 properties

Balance piston seal
- Texas A&M code
- Perfect gas properties



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

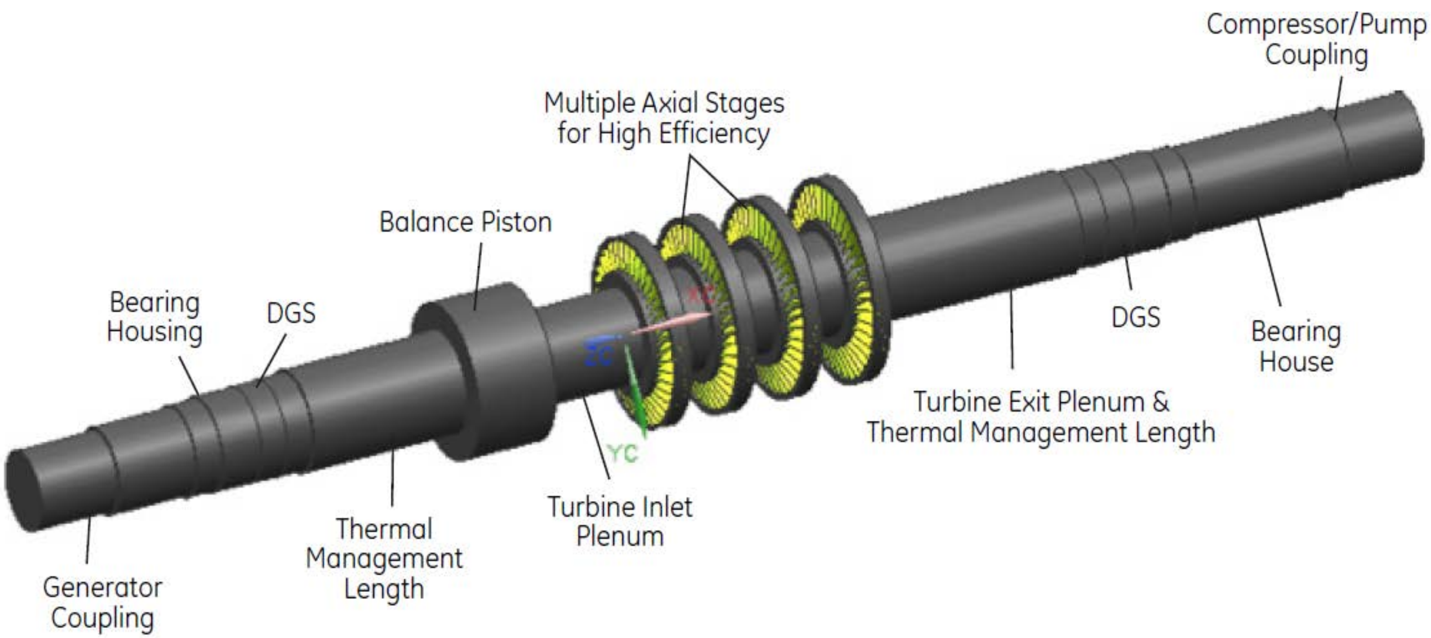


Not to scale

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

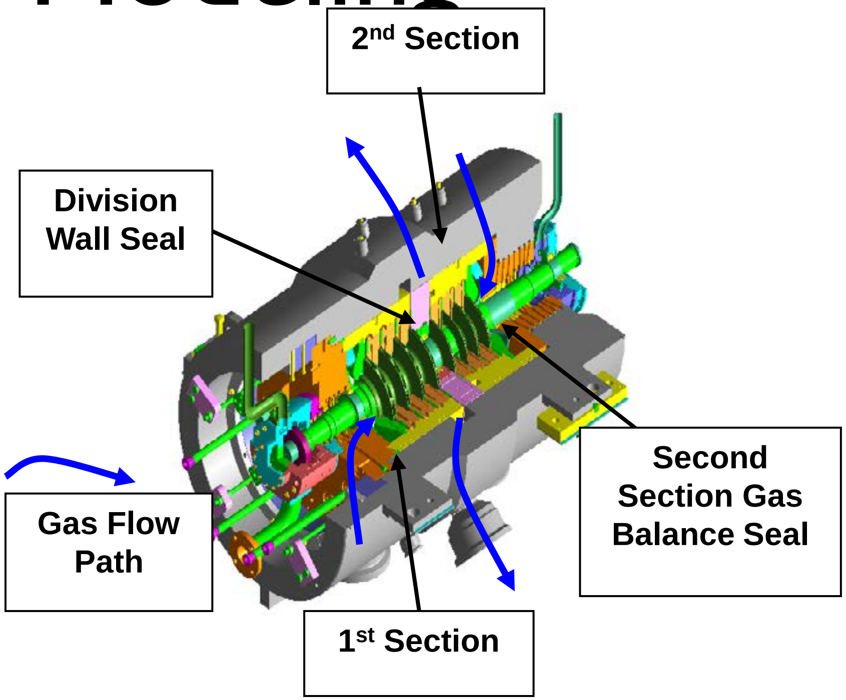


Kalra , C., Hofer, D., Sevincer, E., Moore, J., 2014, "Development Of High Efficiency Hot Gas Turbo-expander For Optimized CSP Supercritical Co_2 Power Block Operation," 4th International Symposium – Supercritical CO2 Power Cycles, Sept 9-10, 2014, Pittsburgh, PA

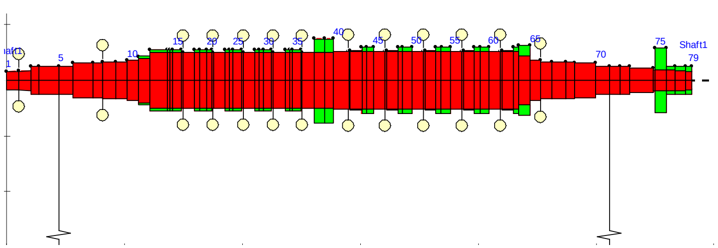
Rotordynamic Modeling

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

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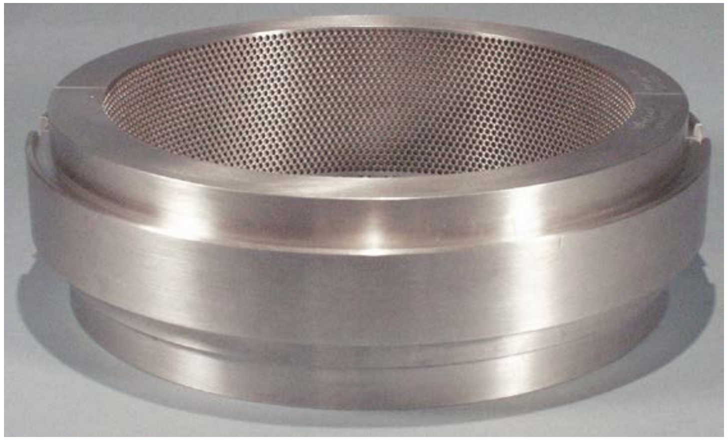
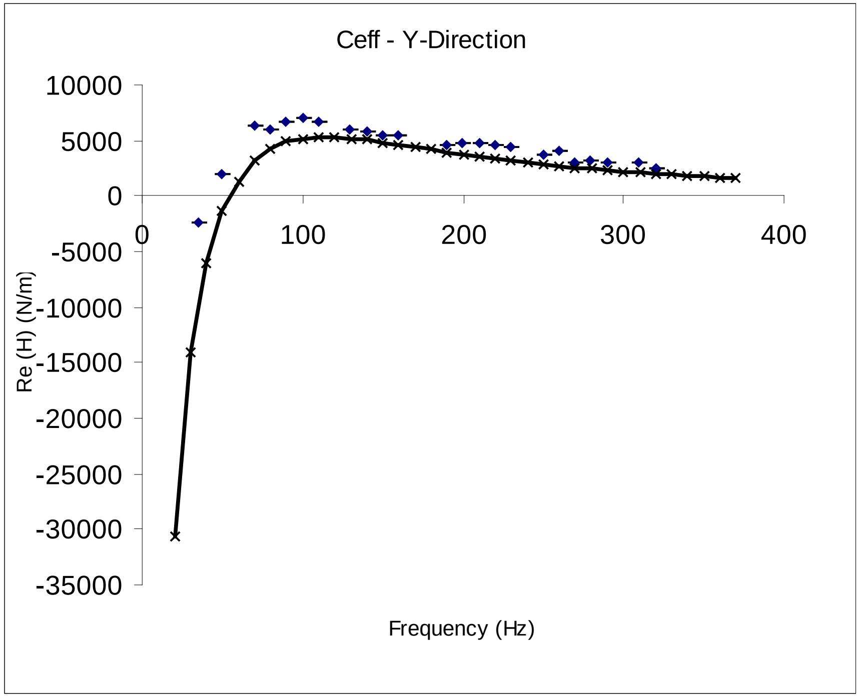
Rotordynamic Modeling

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.

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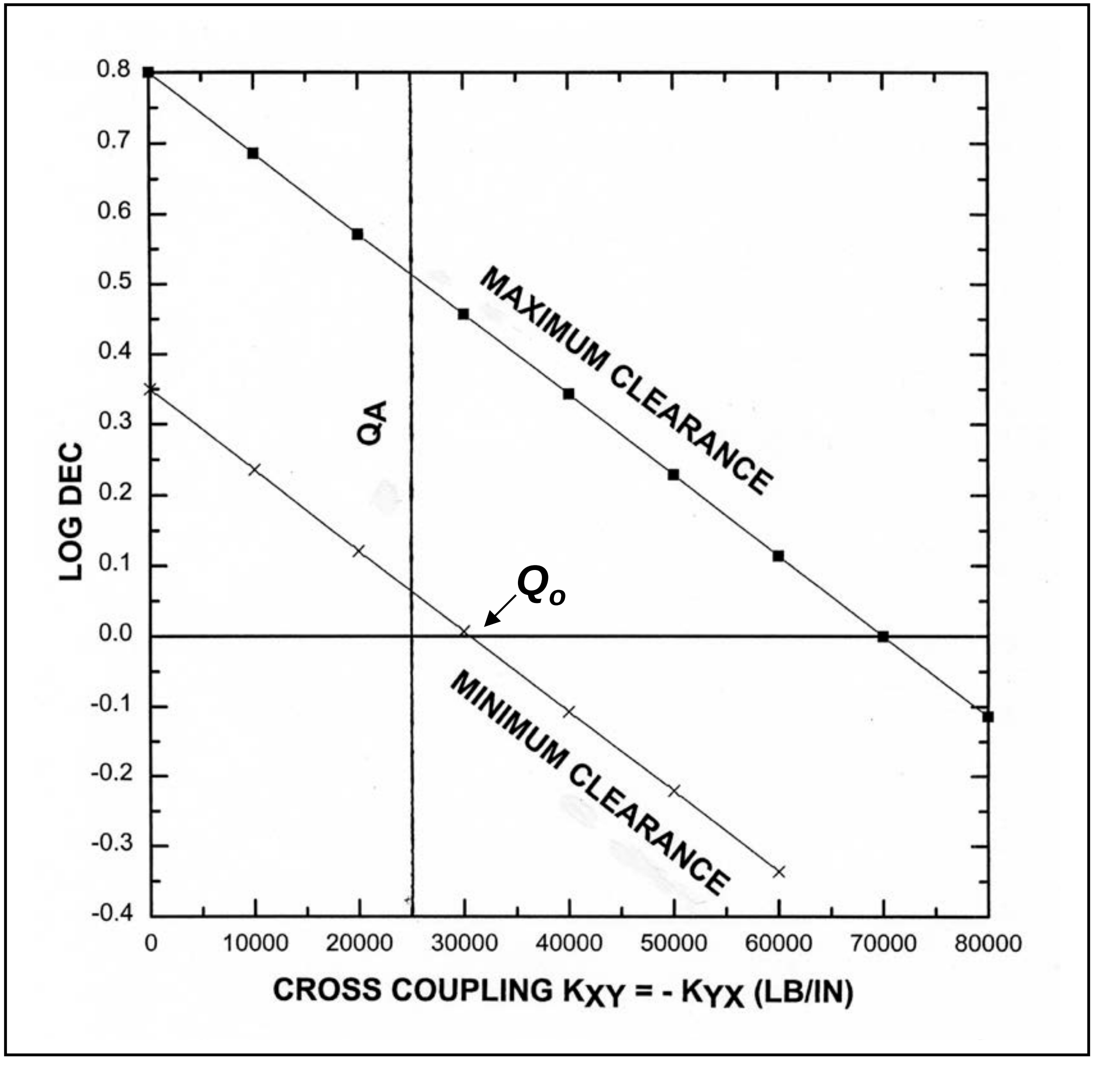
API 617 Requirements

- 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

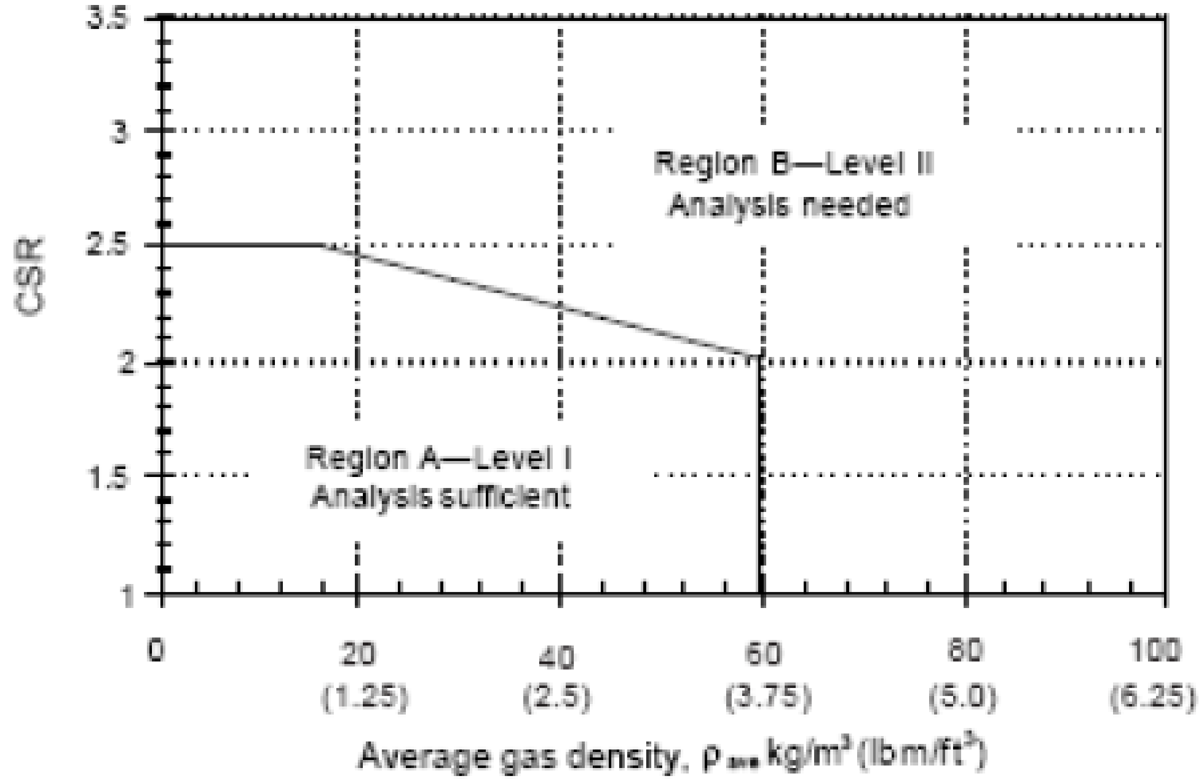


Figure 1.2-5—Level I Screening Criteria

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

Blade Dynamics

Jeff Moore

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

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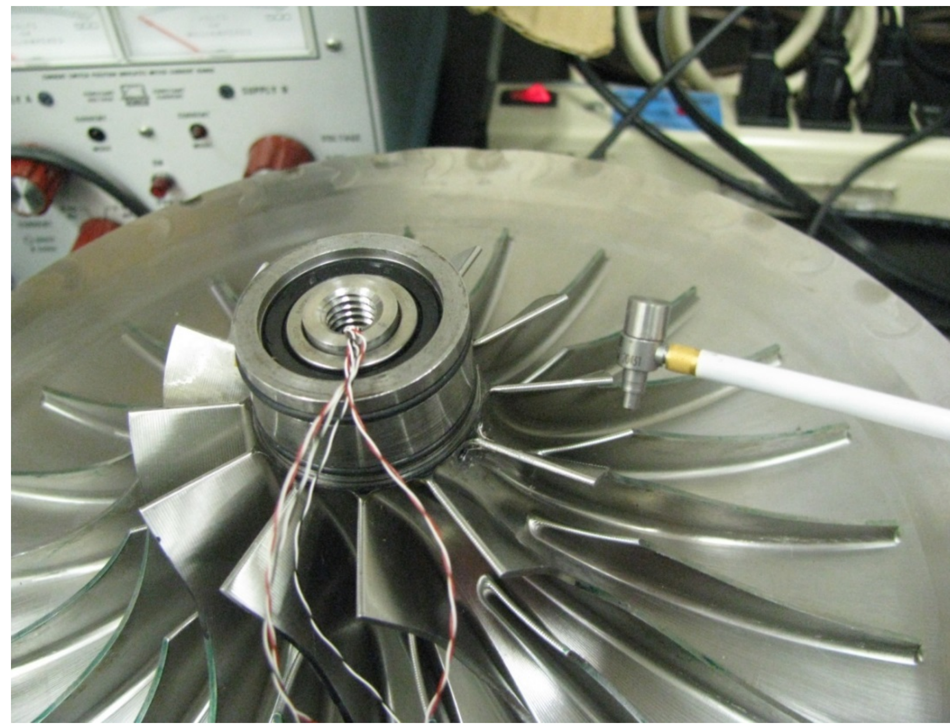
Modal Test Validation

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



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Supercritical CO2 Cycles Materials

Robert Fuller

Supercritical CO_2 Cycles Material Selection

- CO_2 Metal Compatibility/Corrosion
  - Low Temperature -40C to 150C
    - Medium Chrome Steels
  - Medium Temperature 150C to 300C
  - High Temperature 300C+
- CO_2 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

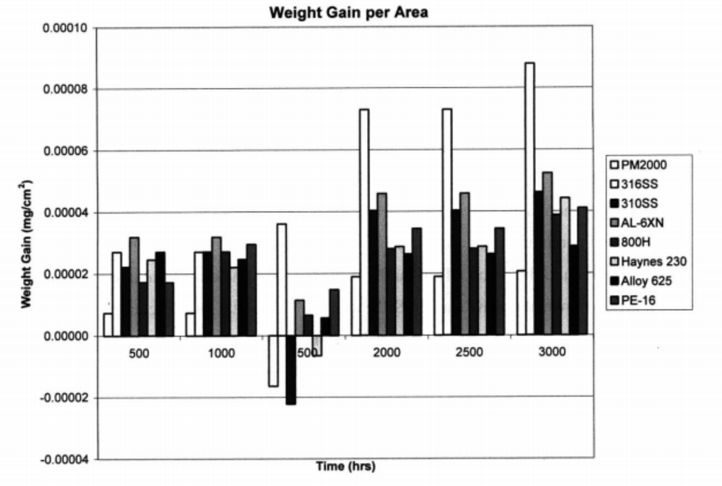


Figure 3.8: Weight gain per area, Alloys F91 and HCM12A not present

Table 3-2: 3000 hr weight gain

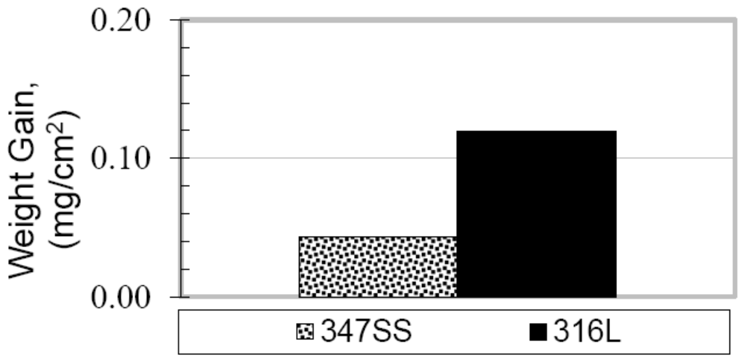
Alloy	Weight gain at 3000 hours (mg/cm^2)
F91	$4.1x10^{-3}$
HCM12A	$5.5x10^{-3}$
PM2000	$2.1x10^{-5}$
316SS	$8.7x10^{-5}$
310SS	$4.6x10^{-5}$
AL-6XN	$5.2x10^{-5}$
800H	$3.9x10^{-5}$
Haynes 230	$4.4x10^{-5}$
Alloy 625	$2.9x10^{-5}$
PE-16	$4.1x10^{-5}$

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 Selection, 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
  - 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

Supercritical CO2 Cycles Pressure Containment

Robert Fuller

Barber Nichols

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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

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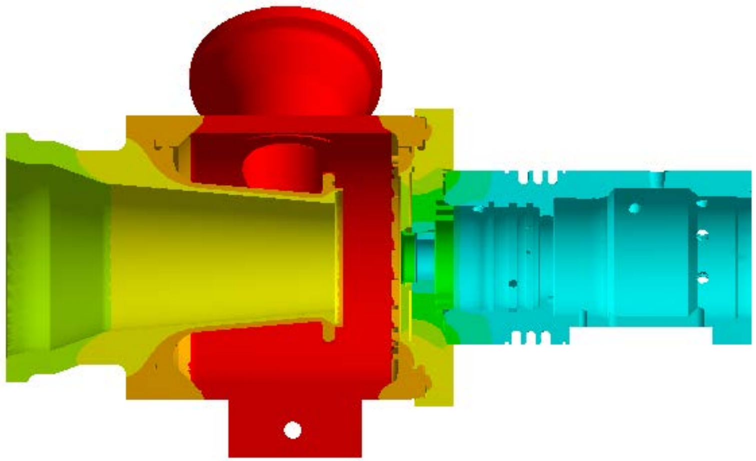
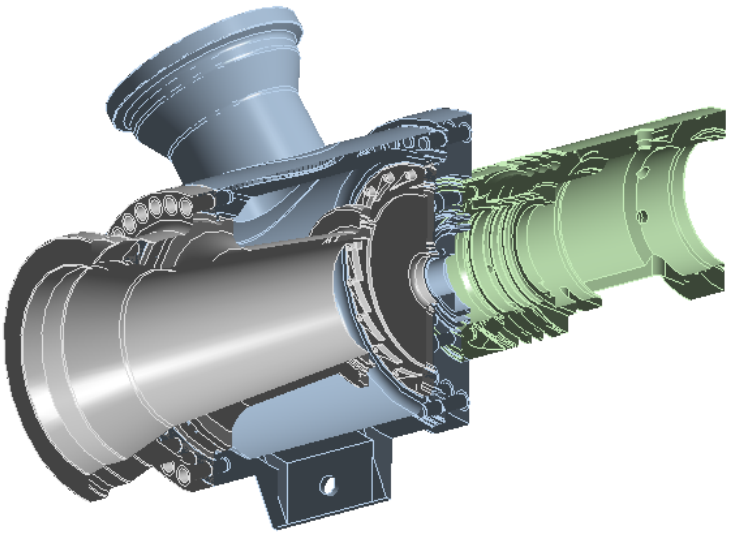
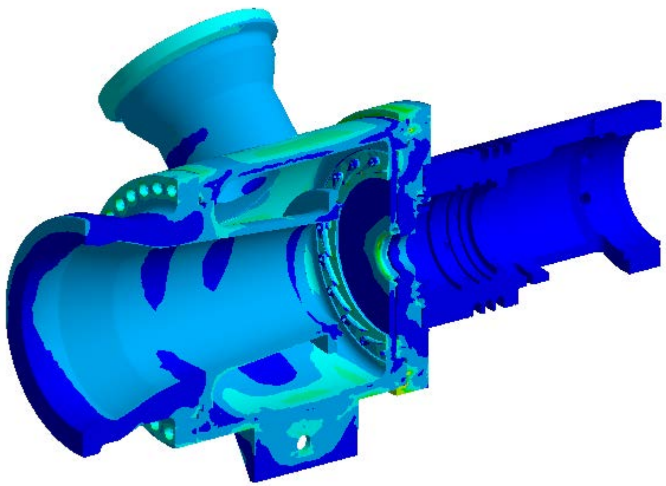
Radial Turbine Housing – Operating Stress Example

- 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







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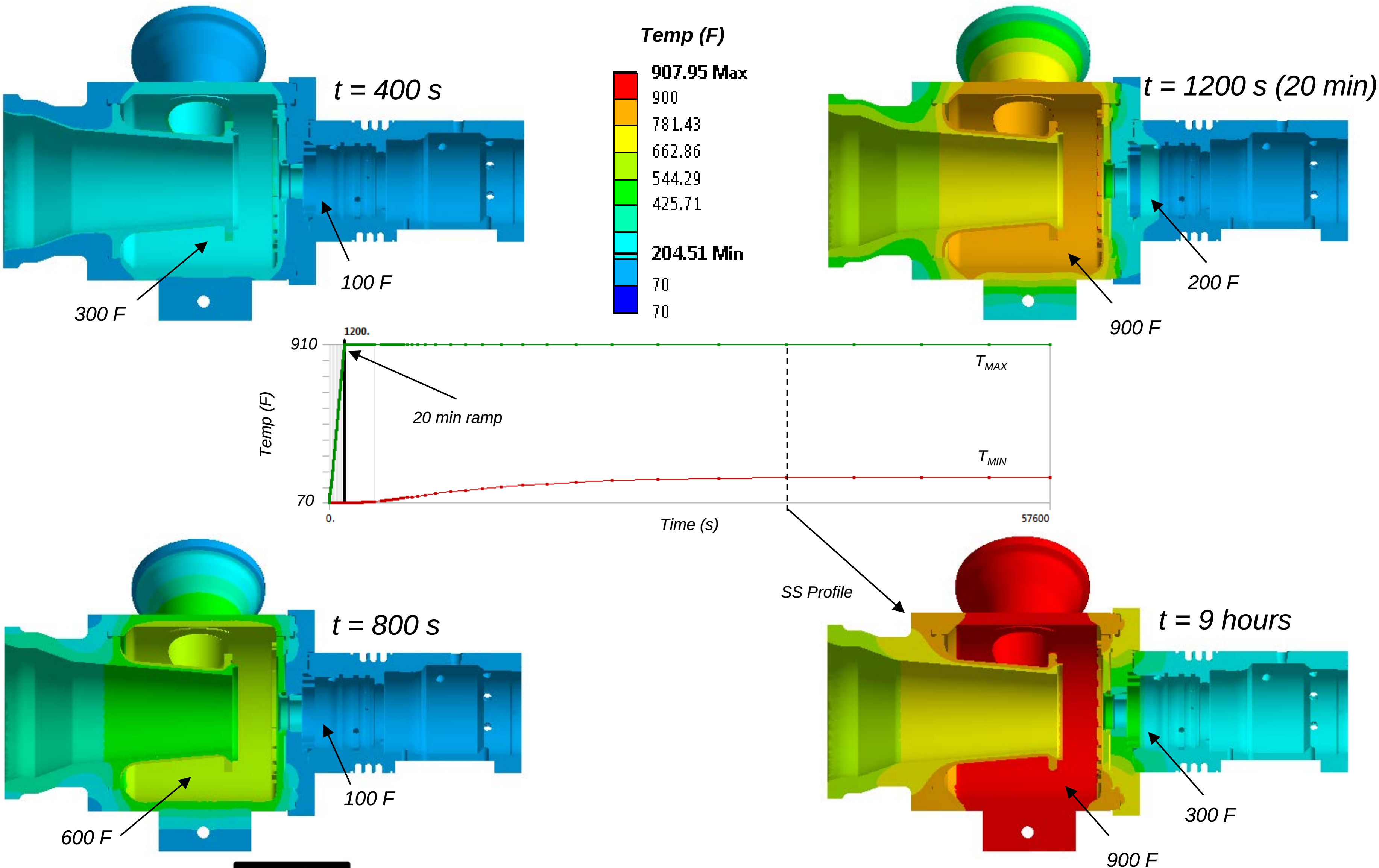
Radial Turbine Housing – Transient Thermal Profile



Thermally Induced Stresses Can Limit Startup Time

Thermal Management

Jeff Moore

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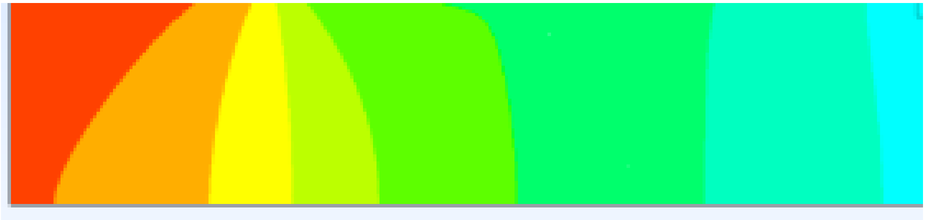
Thermal Management

- 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

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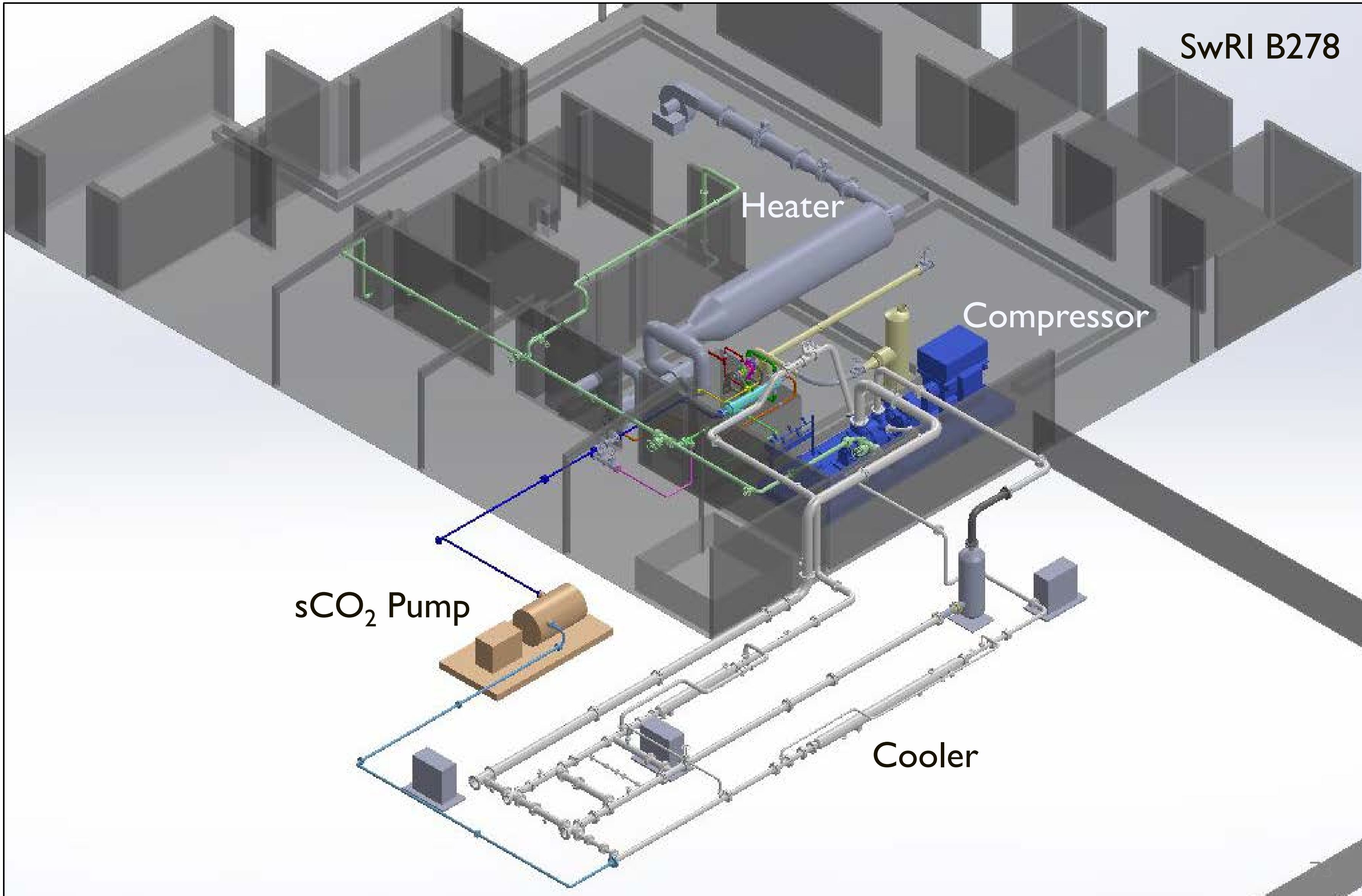
Test Loop Design

Jeff Moore

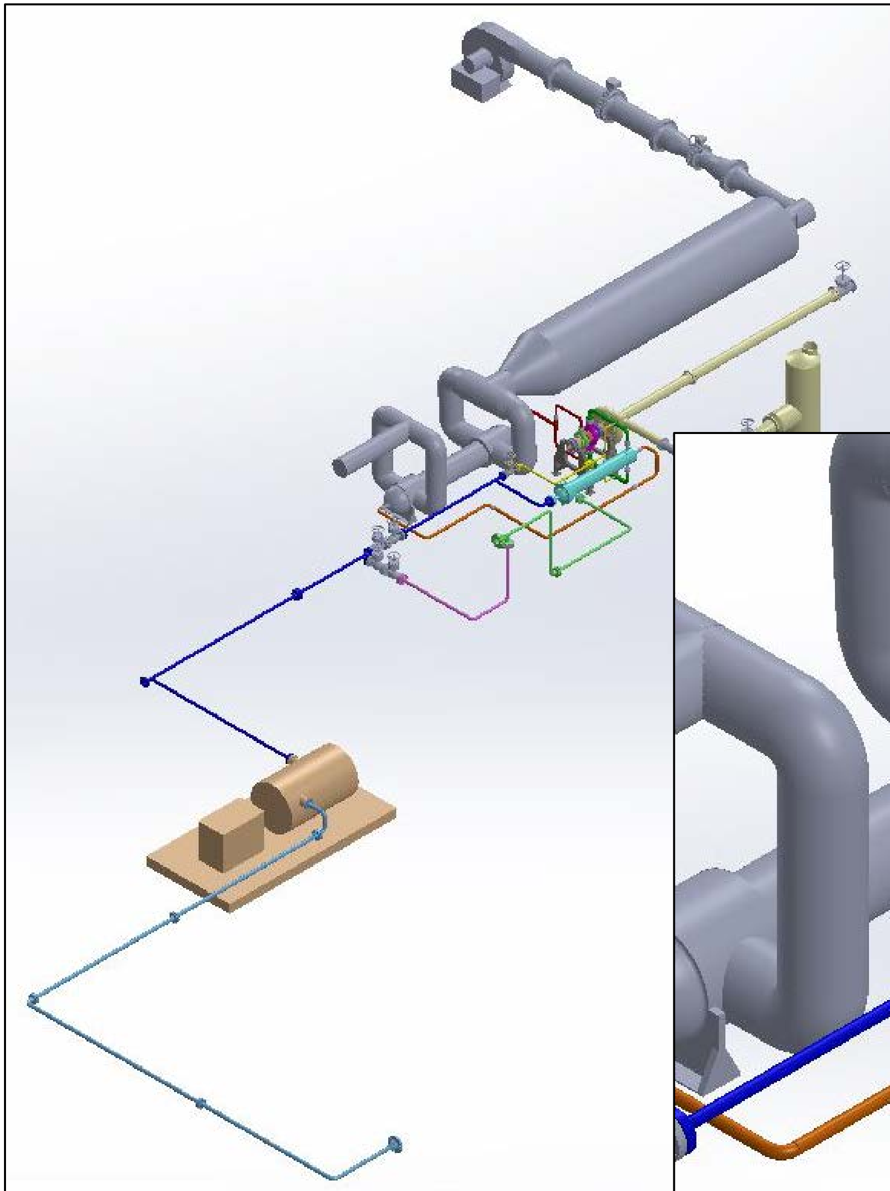
CO2 Compression Loop at SwRI

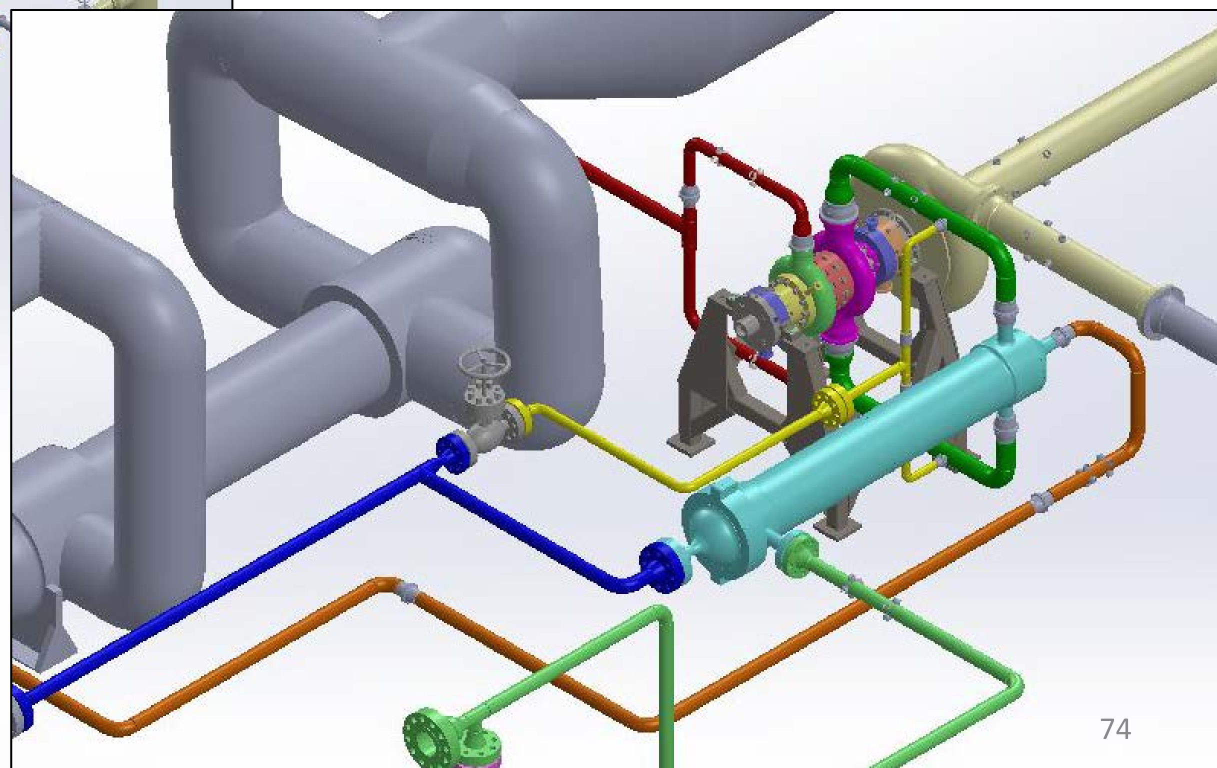
- 3 MW CO_2 compression under construction
- Will be used to perform aerodynamic testing of 10 MW SCO_2 turbine prototype
- High pressure portion of the loop will be used for full pressure-temperature testing of turbine

Test Configuration



Test Configuration

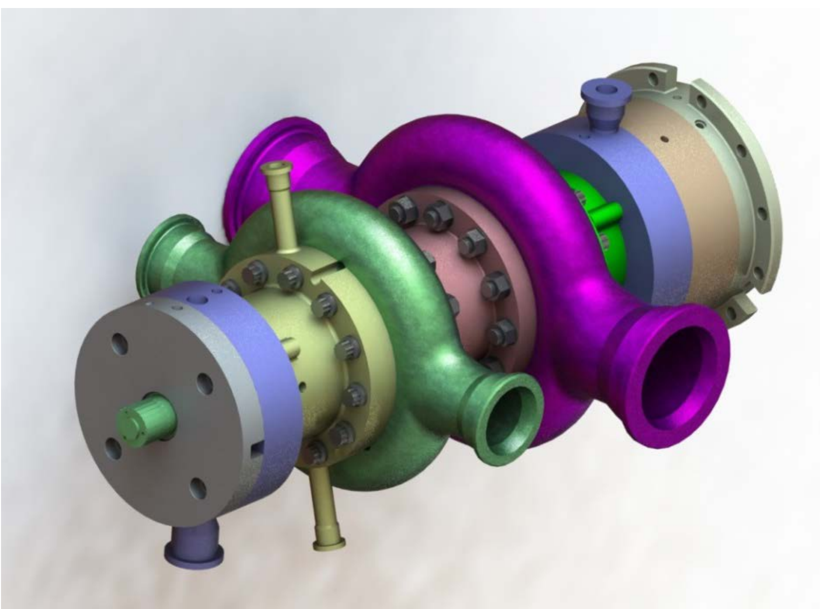


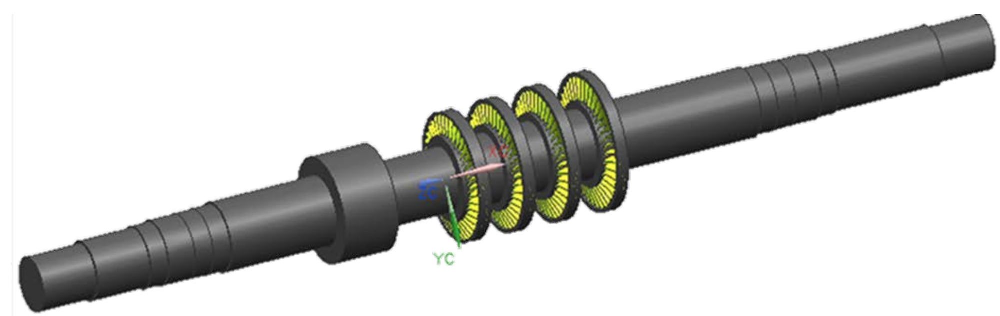


Pipe Section	Color
Pump to heater	Dark blue
Mixing line	Yellow
Recuperator to heater	Orange
HT heater to expander	Red
Expander to recuperator	Dark green
Recuperator to existing	Light green
Existing piping to pump	Light blue

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





Summary

Jeff Moore

Summary

- SCO2 Cycle can Provide over 50% Thermal Efficiency
- SCO2 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.

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