Tutorial: Turbo Machinery Design for Supercritical CO₂ Applications

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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
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**

- 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**

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

*Courtesy of MAN*

*Courtesy of Dresser-Rand*
Optimum Single Stage Pump Requires
N = 22000 rpm,
2.4MW CO2 Pump

For High Efficiency or Controllability, a High Speed Turbine May Drive a Single Stage Pump

Approximate Range of Existing Pump
Commercial Pump Option

Single Stage Pump Option
For Turbine Drive
High Speed Pump Head and Flow Coefficients for CO2 Over Wide Inlet Condition Range Changes Due to Density Range
Turbomachinery Elements-Main Compressor

Main Compressor/Pump-Saturated Liquid or Vapor Inlet

Figure 6.1 Recompression Brayton cycle layout

Recompression Brayton Cycle versus Condensing Brayton with Reheat

Eff = 50.6%

T = 500 C for Turbine Inlet

Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle
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
Turbomachinery Elements - Turbine

Figure 6.1 Recompression Brayton cycle layout

Figure 6.2 Temperature-entropy diagram of a recompression Brayton cycle
Turbine Performance Map

Main Compressor Turbine Performance Map

$P_{ref} = 1013.25$ [kPa], $T_{ref} = 428.65$ [°K], $\gamma_{ref} = 1.317078$ [], $2R_{ref} = 38.7055$ [ft-lbm/lbf-ft]

Corrected Specific Ideal Enthalpy Drop [BTU/lbm]

Corrected Massflow [lbm/s]
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

Dry Gas Seals
• Commercially available at the required pressure but limited to low temperature and smaller diameter.
• Requires clean, dry, filtered CO2 for seal buffer gas
  • Superheat required to prevent liquid and dry ice formation during expansion across face
Annular Gas Seals in Compressor

Image source [7-1]
Different Seal Geometries

Hole-Pattern Seal

Honeycomb Seal

Labyrinth Seal

Fluid properties affect rotordynamics!

Image source [7-2]
Dry Gas Seals

Rotating seal surface...

Image source [7-4]
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

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)
CO2 Hydrostatic Bearings

- Combined Thrust/Radial Bearing
- High Damping
- Good load capacity
- Hermetic Machine
- Moderate CO2 Flow Rates
- Self Supplied By Pump (Need Bootstrap)
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

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

- 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

Rotordynamic Modeling

Damper Seal Damping Test Data vs. Predictions

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


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 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
Supercritical CO2 Cycles
Materials

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# Supercritical CO2 Cycles

## Material Selection

<table>
<thead>
<tr>
<th>CO2 Metal Compatibility/Corrosion</th>
<th>Low Temperature -40°C to 150°C</th>
<th>Medium Chrome Steels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Medium Temperature 150°C to 300°C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>High Temperature 300°C+</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>CO2 Seal Material Compatibility</th>
<th>Elastomeric</th>
<th>Rotating Shaft Seals</th>
<th>High Temperature Seals</th>
</tr>
</thead>
</table>
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
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:

\[
\begin{align*}
M + CO_2 & \rightarrow MO + CO \\
2M + CO_2 & \rightarrow MO + C \\
M + CO & \rightarrow MO + C
\end{align*}
\]

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

Oxide Formation Increases Material Spalls (Corrosion and Erosion)
Alloy Corrosion Tests (UW-Madison)

<table>
<thead>
<tr>
<th>Alloy</th>
<th>C</th>
<th>Fe</th>
<th>Cr</th>
<th>Ni</th>
<th>Mn</th>
<th>Nb</th>
<th>Mo</th>
<th>Si</th>
<th>Cu</th>
<th>Co</th>
</tr>
</thead>
<tbody>
<tr>
<td>316L</td>
<td>0.045</td>
<td>64.3</td>
<td>17.4</td>
<td>13.3</td>
<td>1.7</td>
<td>-</td>
<td>2.7</td>
<td>0.43</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>347ss</td>
<td>0.051</td>
<td>68.5</td>
<td>17.7</td>
<td>9.62</td>
<td>1.66</td>
<td>0.72</td>
<td>0.38</td>
<td>0.77</td>
<td>0.38</td>
<td>0.20</td>
</tr>
</tbody>
</table>

200 hours exposure to CO₂ at 650C and 200 bar:

![Graph showing weight gain for 347ss and 316L alloys after exposure to CO₂.](image-url)
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

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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


**KTL - 3rd Annual Pressure Equipment Conference, 1999**

### Table: ASME Section VIII Divisions

<table>
<thead>
<tr>
<th>Section VIII Division 1</th>
<th>Section VIII Division 2</th>
<th>Section VIII Division 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Published</td>
<td>&lt; 1940</td>
<td>1968</td>
</tr>
<tr>
<td>Pressure Limits</td>
<td>Normally up to 3000 psig</td>
<td>No limits either way, usually 800+ psig</td>
</tr>
<tr>
<td>Organization</td>
<td>General, Construction &amp; Material U, UG, UW, UF, UB, UCS, UNF, UCI, UCL, UCD, UHT, ULT</td>
<td>General, Material, Design, Fabrication and others</td>
</tr>
<tr>
<td>Design Factor</td>
<td>Design Factor 3.5 on tensile (4&quot; used previously) and other yield and temperature considerations</td>
<td>Design Factor of 3 on tensile (lower factor under reviewed) and other yield and temperature considerations</td>
</tr>
<tr>
<td>Design Rules</td>
<td>Membrane - Maximum stress Generally Elastic analysis Very detailed design rules with Quality (joint efficiency) Factors. Little stress analysis required; pure membrane without consideration of discontinuities controlling stress concentration to a safety factor of 3.5 or higher</td>
<td>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 analyses considerations may be required unless exempted and guidance provided for in Appendix 4, 5 and 6</td>
</tr>
<tr>
<td>Experimental Stress Analysis</td>
<td>Normally not required</td>
<td>Introduced and may be required</td>
</tr>
<tr>
<td>Material and Impact Testing</td>
<td>Few restrictions on materials; Impact required unless exempted; extensive exemptions under UG-20, UCS 66/67</td>
<td>More restrictions on materials; impact required in general with similar rules as Division 1</td>
</tr>
<tr>
<td>NDE Requirements</td>
<td>NDE requirements may be exempted through increased design factor</td>
<td>More stringent NDE requirements; extensive use of RT as well as UT, MT and PT</td>
</tr>
<tr>
<td>Welding and fabrication</td>
<td>Different types with butt welds and others</td>
<td>Extensive use/requirement of butt welds and full penetration welds including non-pressure attachment welds</td>
</tr>
</tbody>
</table>
ASME SECTION VIII-For Rotating Machinery

1. Useful for Defining Safety Margins
   - 1.5X on Yield Strength, 3.5X on Ultimate Tensile Strength
2. Useful for Defining Hydrostatic Test Requirements
   - 1.3X MAWP (Temperature Rated)
3. Useful for Material Selection and Temperature/Stress De-rating
4. Not Cognizant of Complicated Geometry Found in Turbomachinery (Can use Div 2 for FEA)
5. Transient Thermal Stresses

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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;
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
Radial Turbine Housing – Transient Thermal Profile

Time (s)
- $t = 400 \text{ s}$
- $t = 800 \text{ s}$
- $t = 1200 \text{ s} (20 \text{ min})$
- $t = 9 \text{ hours}$

Temp (F)
- $T_{\text{MIN}}$
- $T_{\text{MAX}}$

Thermally Induced Stresses Can Limit Startup Time
Thermal Management

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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

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

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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

<table>
<thead>
<tr>
<th>Pipe Section</th>
<th>Color</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump to heater</td>
<td>Dark blue</td>
</tr>
<tr>
<td>Mixing line</td>
<td>Yellow</td>
</tr>
<tr>
<td>Recuperator to heater</td>
<td>Orange</td>
</tr>
<tr>
<td>HT heater to expander</td>
<td>Red</td>
</tr>
<tr>
<td>Expander to recuperator</td>
<td>Dark green</td>
</tr>
<tr>
<td>Recuperator to existing</td>
<td>Light green</td>
</tr>
<tr>
<td>Existing piping to pump</td>
<td>Light blue</td>
</tr>
</tbody>
</table>
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

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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.