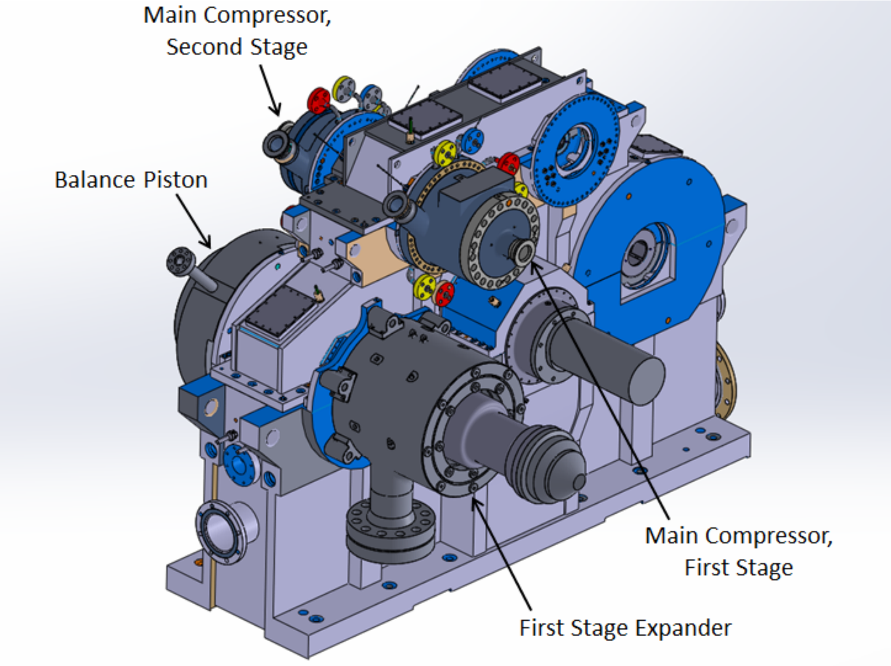
Approach for Ultra-Low Piping Stress in a High-Pressure/High-Temperature Integrally Geared Compander

Jason Wilkes, Ph.D., Kelsi M. Katcher,
Southwest Research Institute

Karl Wygant, Rob Pelton, Jon Bygrave
Hanwha Power Systems Americas

sCO_2 Symposium
Feb 22-24, 2022, San Antonio, TX



Turbomachinery Development for CSP

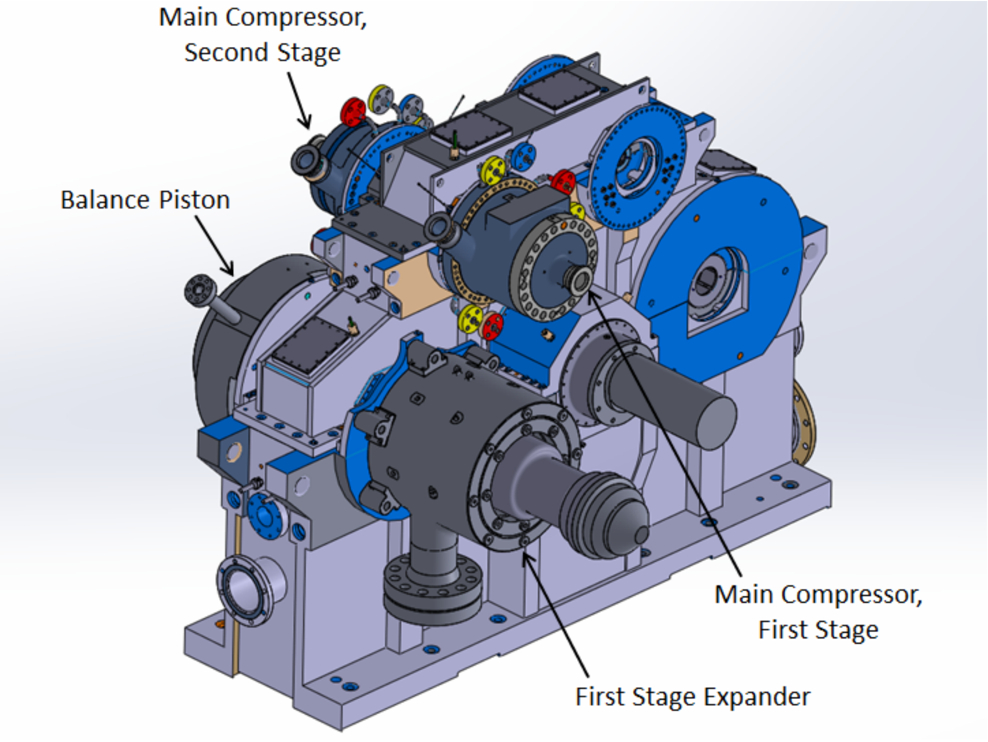
Acknowledgment: This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) under the Solar Energy Technologies Office Award Number DE-EE0007114. This presentation is based on project work supported by Vacuum Process Engineering, Inc.

Overview

- Project Overview
- Test Facility
- Test Loop Design
  - First pass design based on operating conditions
  - Design validation:
    - Thermal modeling
    - Modal analysis
    - Piping vibration screening
    - Heater/turbine piping material analysis
- Final Process Loop Design
- Conclusions and Lessons Learned

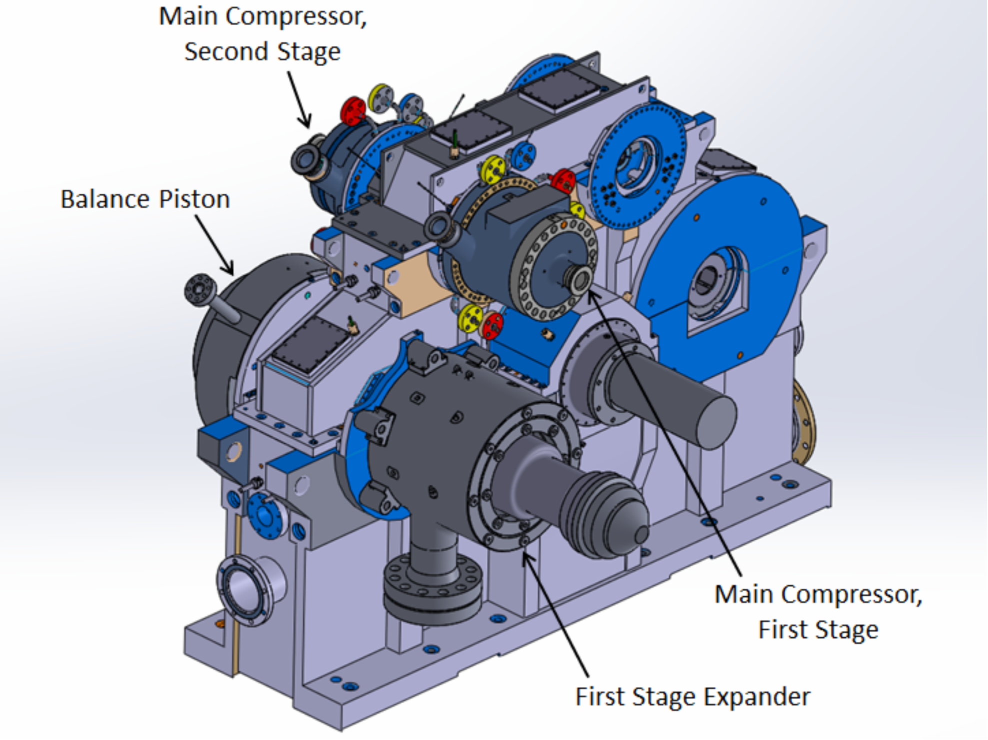
Integrally-Geared Compressor-Expander

- The goal of this project is to develop and validate an integrally-geared compressor-expander (compander)
- The full compander design
  - 10 MW net
  - Four pinion shafts
    - main compressor, 2 stages
    - recompressor, 2 stages
    - 4 expander stages
- High-risk items were selected for performance testing
  - Main compressor (wide operating range)
  - First expander stage (highest temperature and pressure)

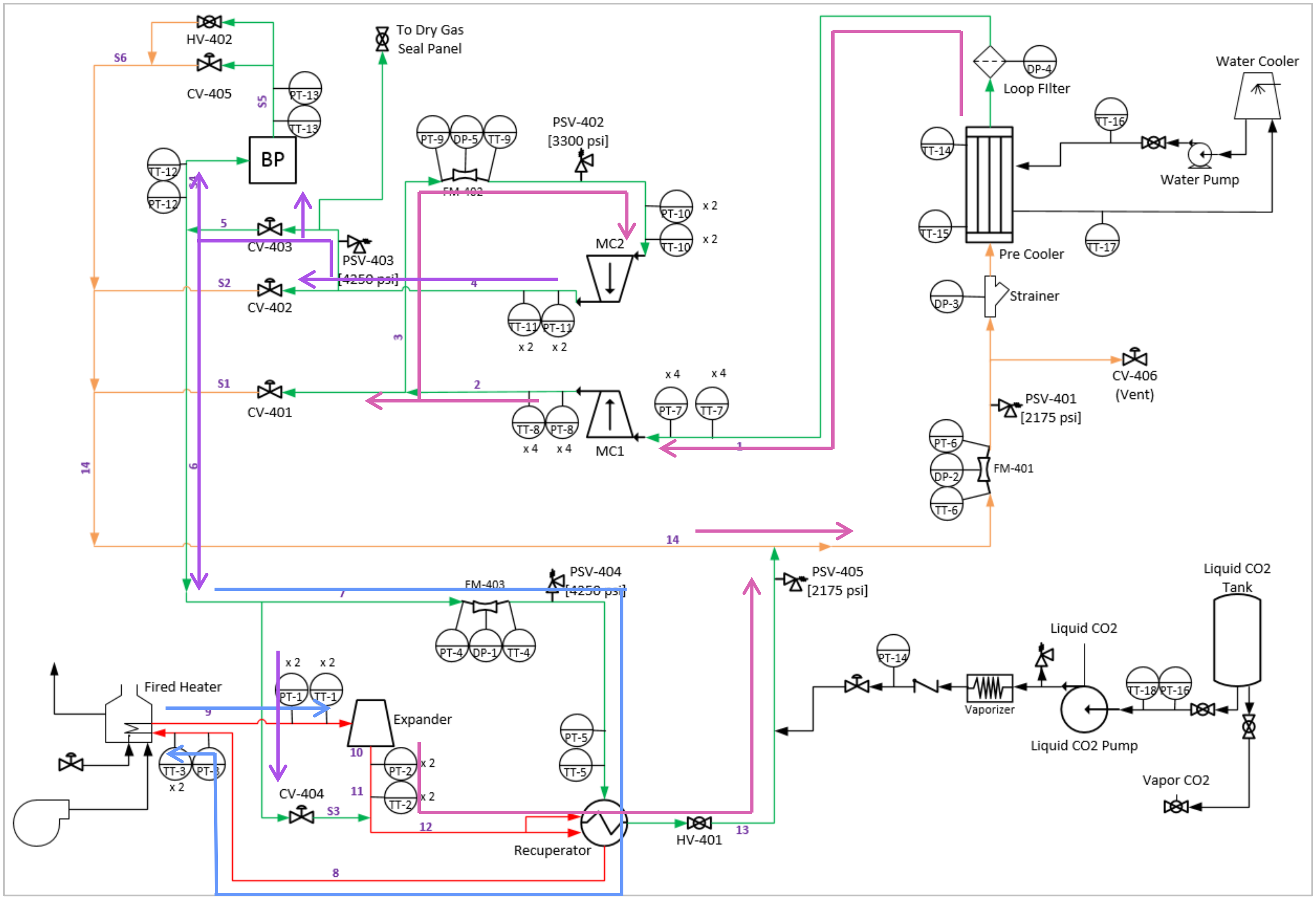


Compander Testing Campaign Objectives

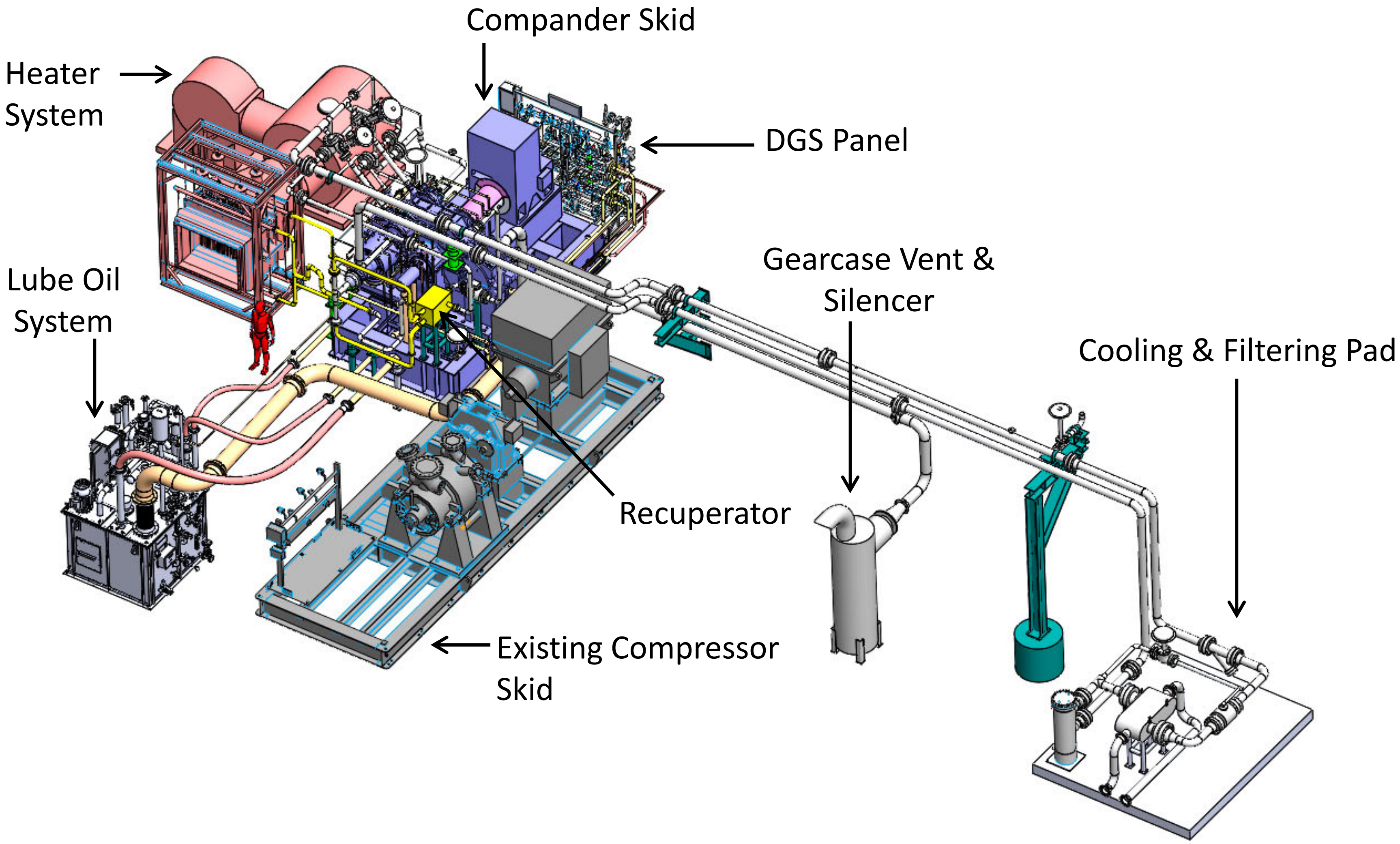
- Full-scale bull gear and case
- Main compressor test
  - Full pressure
  - Full power
  - Wide operating range
- First expander stage
  - Full pressure
  - Full temperature
  - Full speed
  - Reduced mass flow (limited by available heat input)



Process & Instrumentation



Compander Test Facility



Compander Skid

Heater System

DGS Panel

Lube Oil System

Gearcase Vent & Silencer

Cooling & Filtering Pad

Recuperator

Existing Compressor Skid

Piping Design Considerations – First Pass

- ASME PTC-10 standard for machinery instrumentation
- Material selection based on pressure/temperature
- Diameters sized to limit flow velocities to 30 m/s or less
  - SwRI has experience with operating sCO_2 test loops with flow velocities over 60 m/s.
  - The elevated piping in this test loop is highly susceptible to vibration. Velocities were limited to reduce excitation energy.
- Diameter, wall thickness, material selections based on piping codes
  - ASME B31.3 piping code (2008 version) for carbon steel / stainless steel
  - ASME B31.1 (2012 version) for Inconel 625 / Inconel 740
- Major pressure losses in each line were limited
  - < 0.34 bar [5 psi]
  - Allowed to exceed this value in lines where a large pressure drop is intended across a control valve (i.e. compressor recycle, expander bypass)

Design Temperatures & Pressures

- The compander cycle operating conditions were predicted using an NPSS fully-defined cycle model.
- Design pressure defined as the PSV set point pressure
  - PSV set-point was defined by the greater of:
    - cycle-predicted process pressure or
    - loop settle-out pressure,
    - ➢plus a 7% margin for PSV set-point avoidance
- Corrosion allowance:
  - Stainless steel / Inconel: 0.020 inches
  - Carbon steel: 0.060 inches
- Maximum allowable temperature was also calculated for each line (for the selected material, pipe diameter, and pipe schedule) to quantify the operable temperature margin

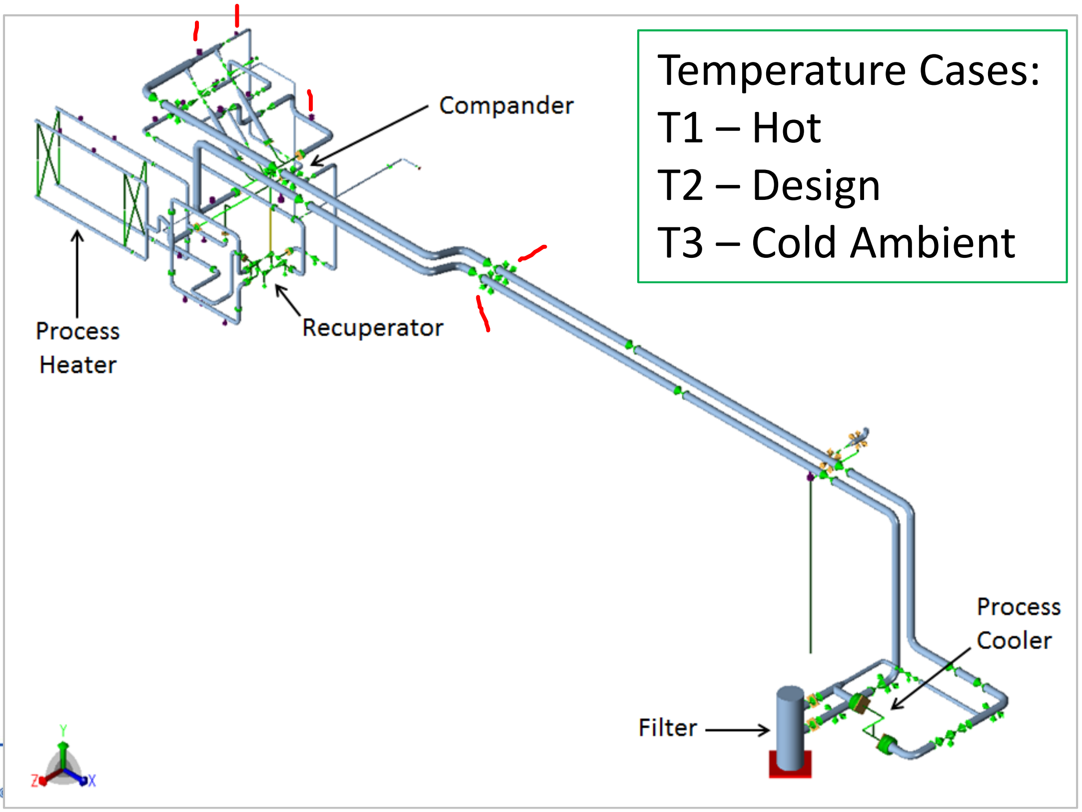
Test Loop Design Validation: Thermal Modeling

Caesar II thermal pipeline modeler

- predict the piping stresses and displacements caused by weight, pressure, and thermal growth
- calculate the nozzle loads on the compander compressors and expander as well as on the heat exchangers
- calculate the modal frequencies of the piping system

Piping model includes:

- Concentrated mass (valves, flanges)
- Compander case and HXs modeled as rigid
- Support locations and stiffnesses



Temperature Cases:
T1 – Hot
T2 – Design
T3 – Cold Ambient

Test Loop Design Validation: Thermal Modeling

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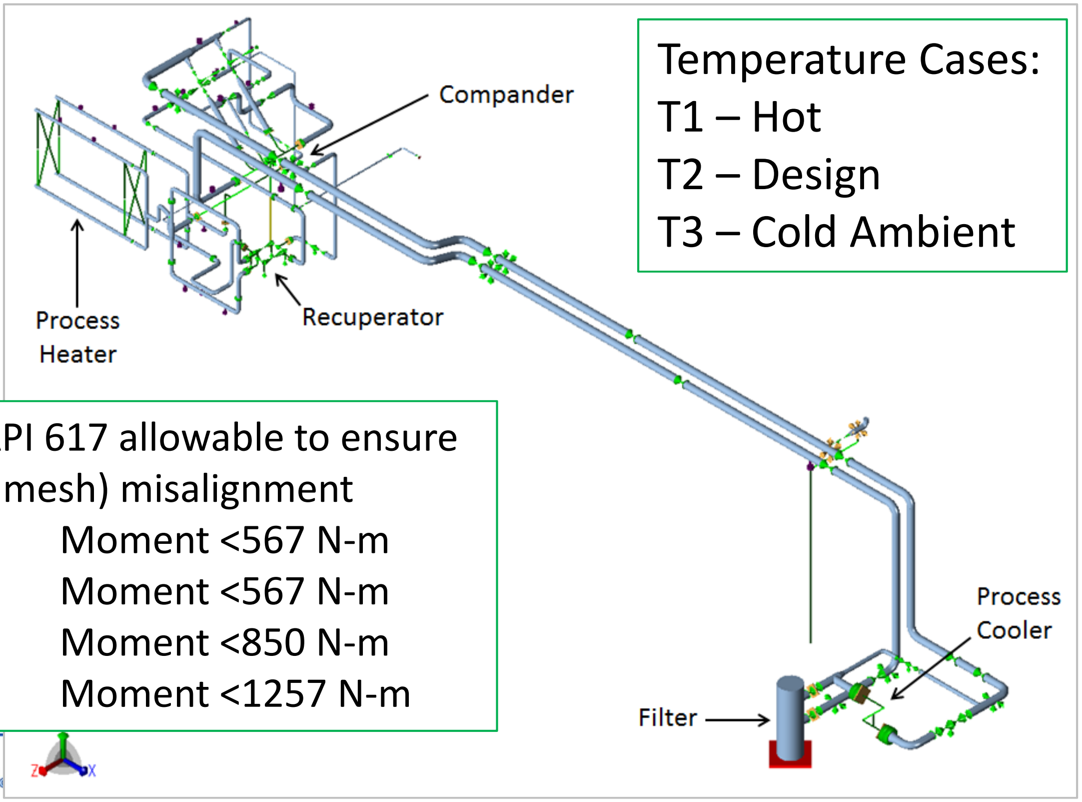


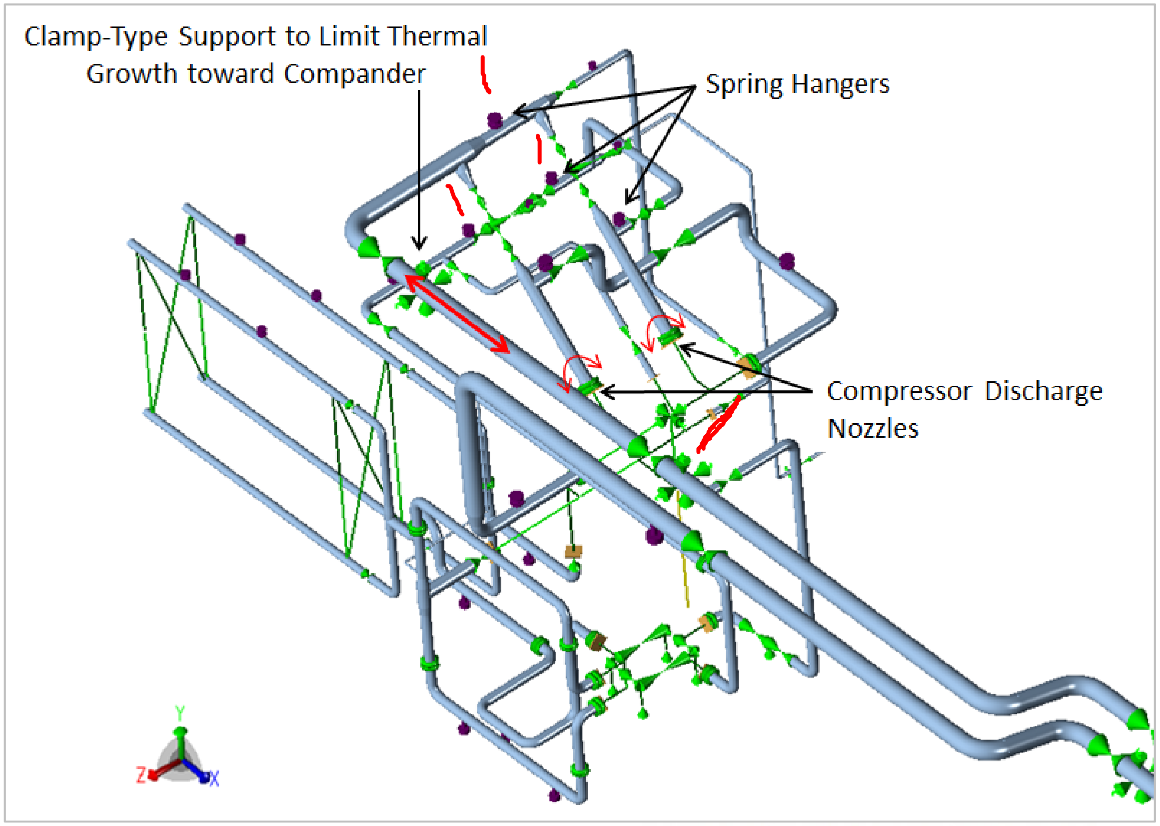
Temperature Cases:
T1 – Hot
T2 – Design
T3 – Cold Ambient

Maximum loads are taken to be 30% of API 617 allowable to ensure bolting integrity and to limit pinion (gear mesh) misalignment

Compressor Inlet:	Force <1032 N	Moment <567 N-m
Compressor Outlet:	Force <1032 N	Moment <567 N-m
Expander Inlet:	Force <1548 N	Moment <850 N-m
Expander Outlet:	Force <2286 N	Moment <1257 N-m

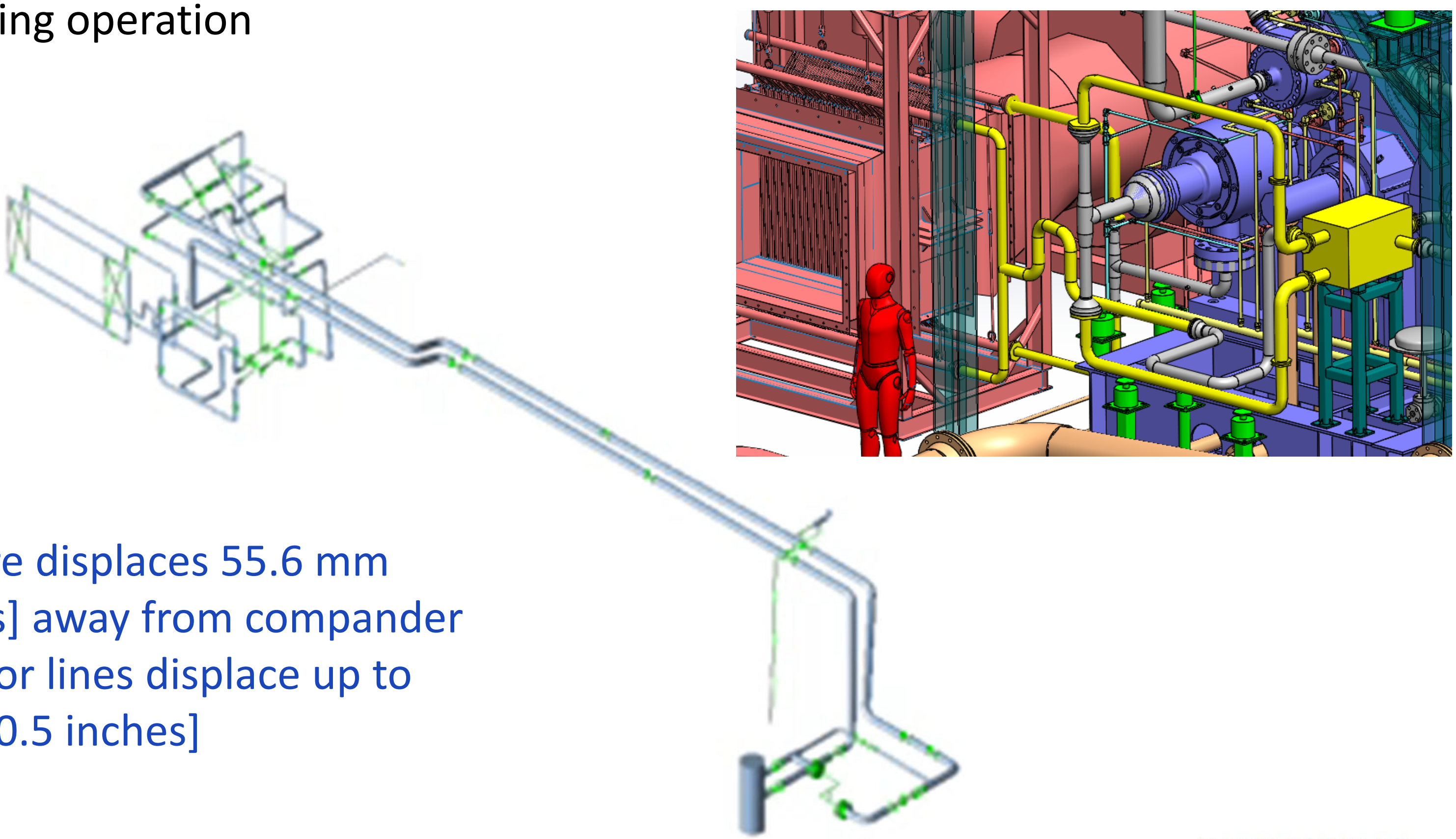
Test Loop Design Validation: Thermal Modeling

- After several design iterations within Caesar II, it was evident that the compander nozzle loads were the limiting factor in the test loop design.
- The majority of the piping was supported with spring hangers to provide weight support, but allow thermal growth away from the compander.
- Clamp-type supports were installed on the long runs to and from the cooler to force thermal growth away from the compander nozzles, requiring a rigid support structure.
- Locations of valves/flanges was also iterated to unload the compressor nozzles.
- **Balancing game..**



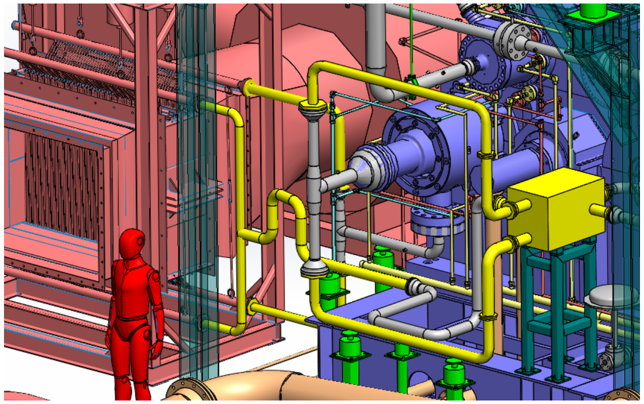
Test Loop Design Validation: Thermal Modeling

- Significant thermal growth in the hot section
- Heavy-wall piping required is very stiff
- The combination can cause elevated piping stresses and elevated equipment nozzle loads during operation



- Heater core displaces 55.6 mm [2.2 inches] away from compander
- Recuperator lines displace up to 12.5 mm [0.5 inches]

Test Loop Design Validation: Thermal Modeling



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- Heavy-wall piping required is very stiff
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➢ The primary heater and the recuperator were softly supported to allow movement.

Heater supported with spring hangers

Recuperator Supports – insulated footing with polished stainless steel contact

Modal Analysis

- Using Caesar II, the lowest MNF was predicted at 2.1 Hz.
- Generally, SwRI recommends that all MNFs in the main process piping should be above 15 Hz to minimize vibration risk.
- Here, 39 modes were predicted below 15 Hz.
- The low nozzle load allowables on the compressors were driving, so stiffening the system was not practical.

Energy Institute Vibration Screening

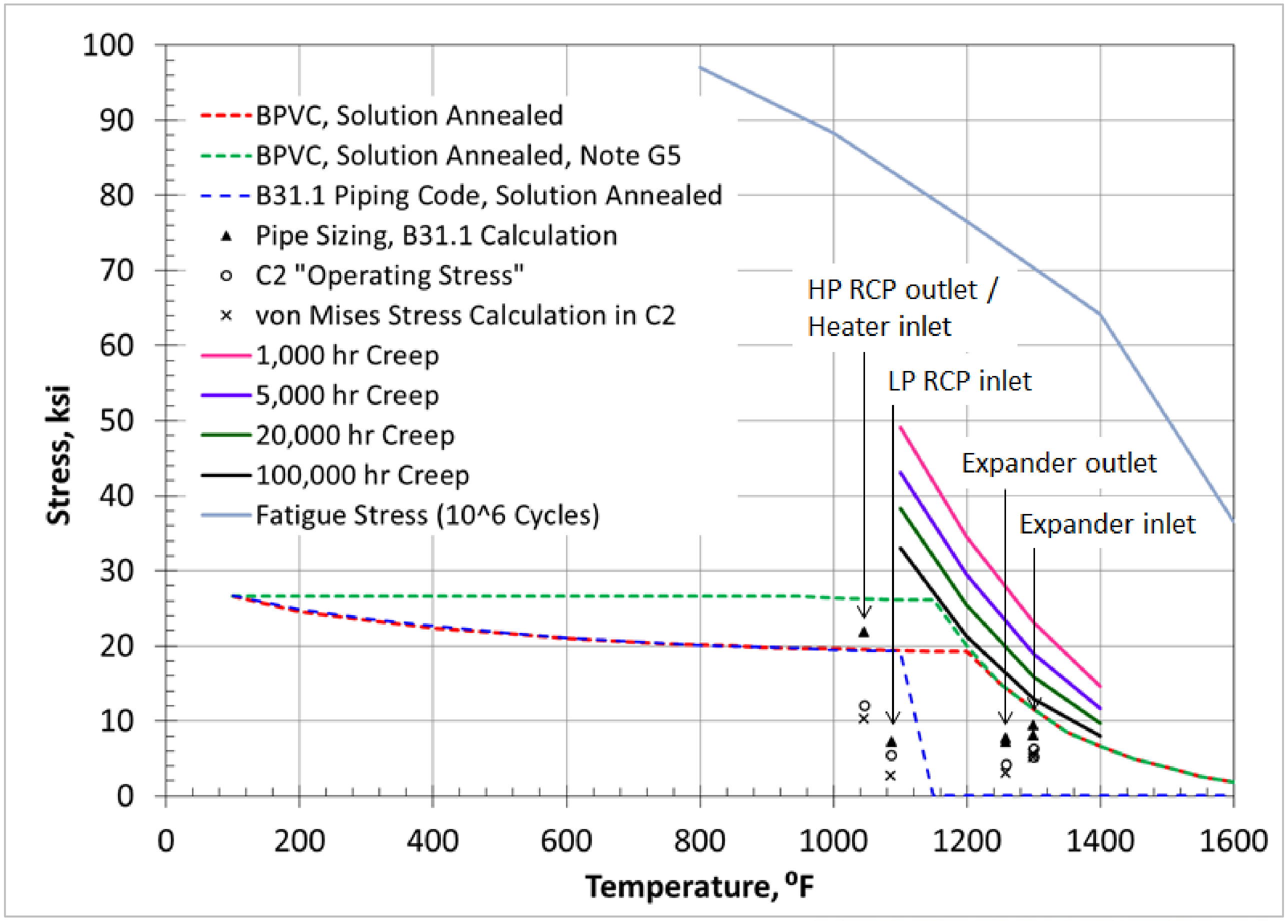
- EI Screening Analysis was completed to minimize risk of excessive vibrations and fatigue failure.
  - velocity, viscosity, and density as a measure excitation energy
  - lowest MNF of each section of pipe
  - determine a likelihood of failure (LOF) factor.
  - Note that "failure" is referring to the presence of vibrations and not necessarily a fatigue failure in the piping.

➢ Pipe diameters were increased in several locations to reduce velocities

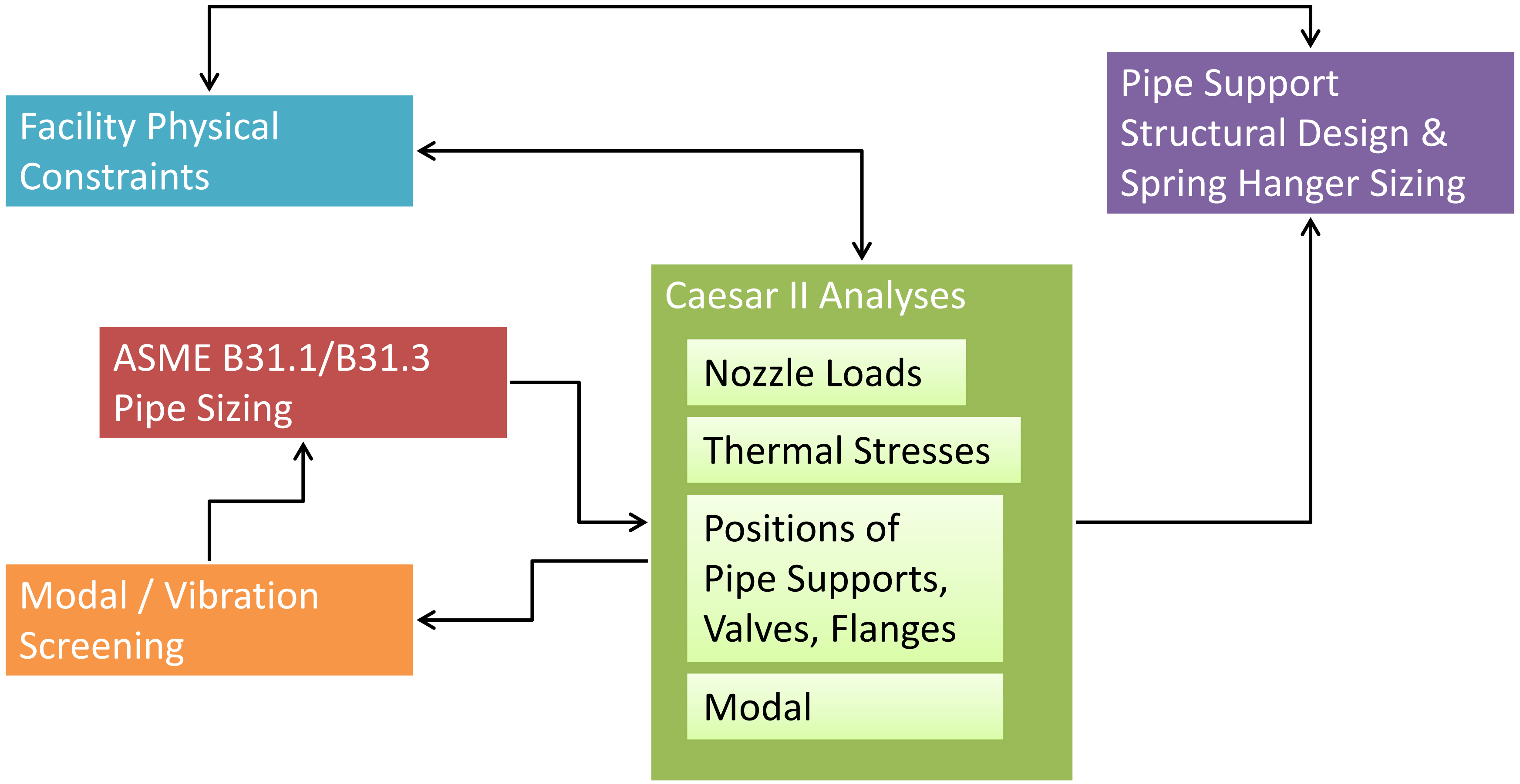
LOF	Action
LOF ≥ 1	Main line **shall** be re-supported or redesigned for relevant piping. Small bore connections actions **shall** be undertaken
1 > LOF ≥ 0.5	Main line **should** be re-supported or redesigned for relevant piping as far as practicable, or vibration monitoring of the main line **should** be undertaken after commissioning to ensure mitigation has been successful or to identify if further modification (e.g. bracing) is required. Small bore connections actions **shall** be undertaken.
0.5 > LOF ≥ 0.3	Modifications to the main line are not required. Small bore connections actions **should** be undertaken.
LOF < 0.3	Acceptable LOF value. No action required.

Validation of Hot Piping Material Selection

- INCO 625 selected
- Strength from B31.1 Piping Code goes to 0 at these conditions.
- Strength values validated with B&PV code, material creep strength, and fatigue strength
- Actual material stresses pulled from pipe sizing analysis and Caesar II pipe stress results



Iterative Design!



Pipe Support Structural Design & Spring Hanger Sizing

Facility Physical Constraints

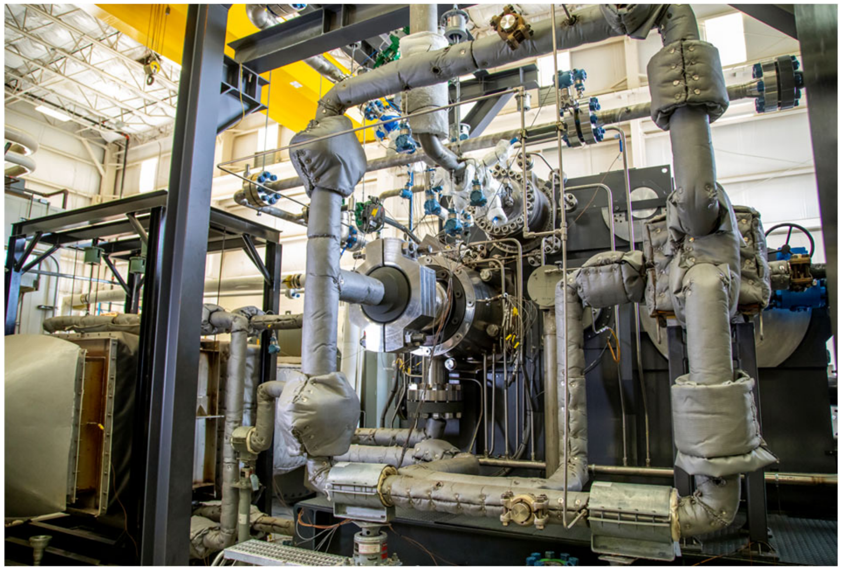
Caesar II Analyses

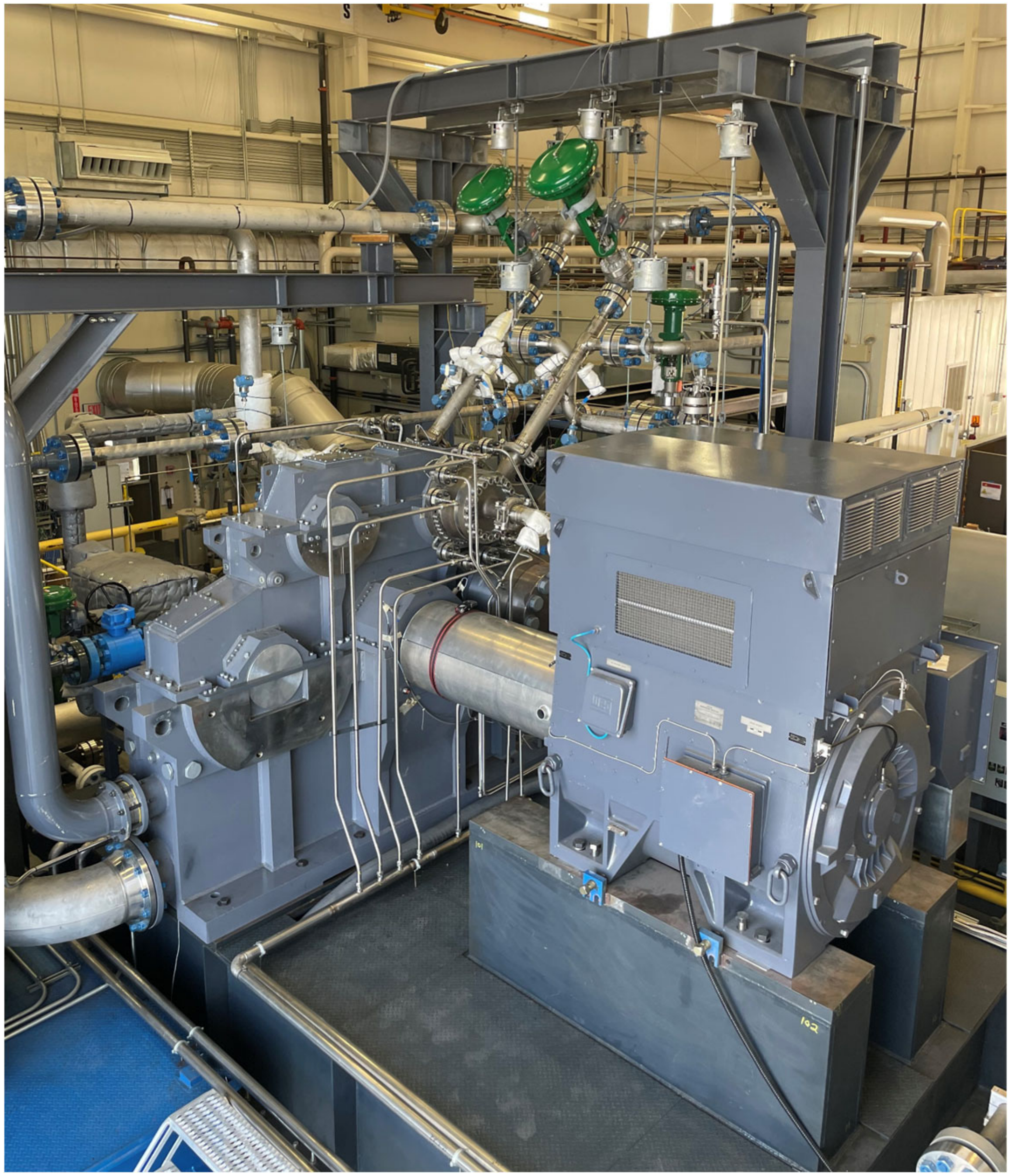
- Nozzle Loads
- Thermal Stresses
- Positions of Pipe Supports, Valves, Flanges
- Modal

ASME B31.1/B31.3 Pipe Sizing

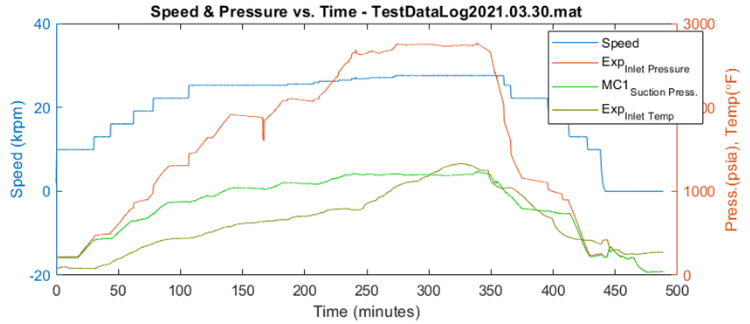
Modal / Vibration Screening

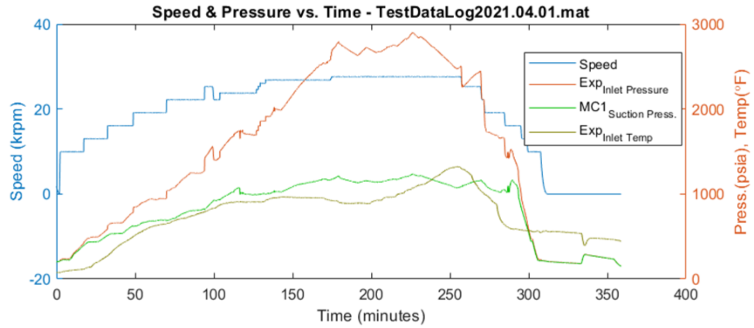
Compander Loop

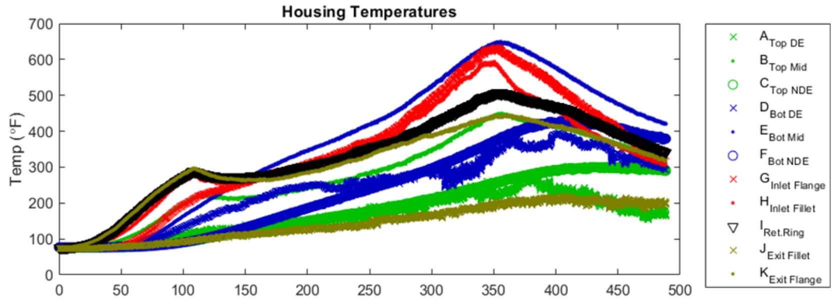


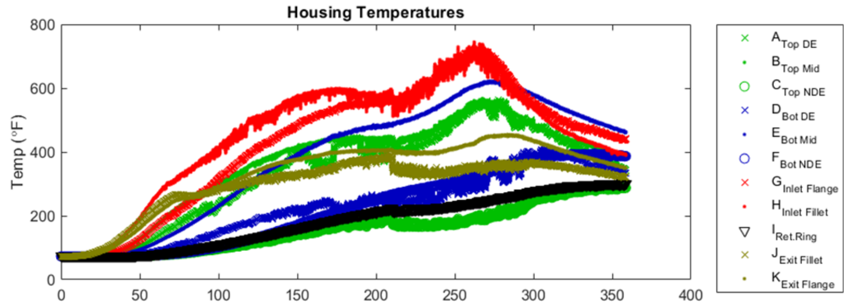


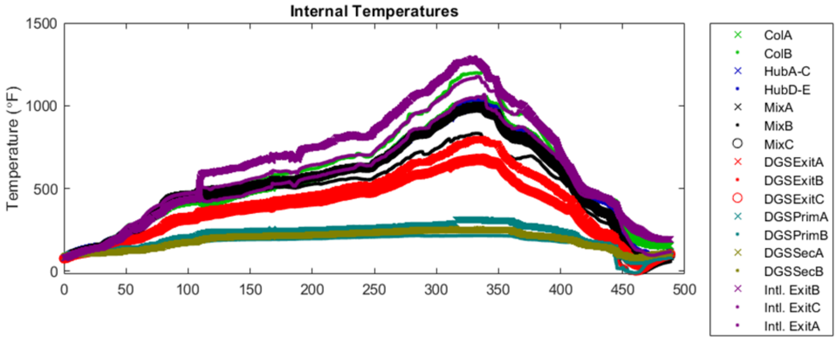
On the first run to temperature we achieved a turbine inlet temperature of 1329°F, or 720°C

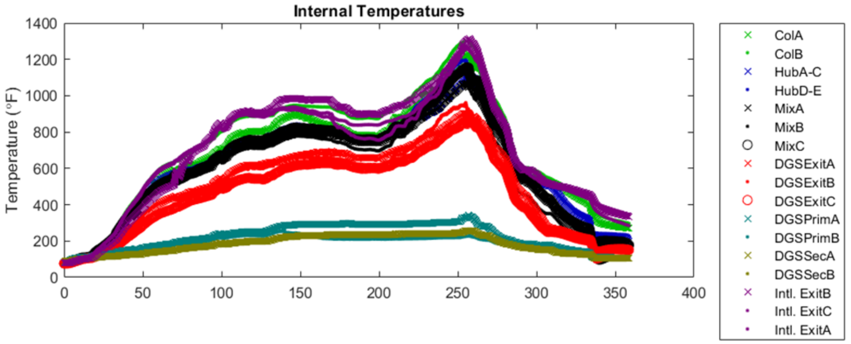






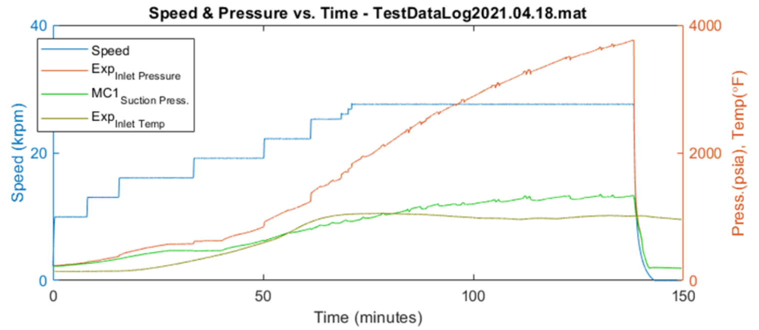


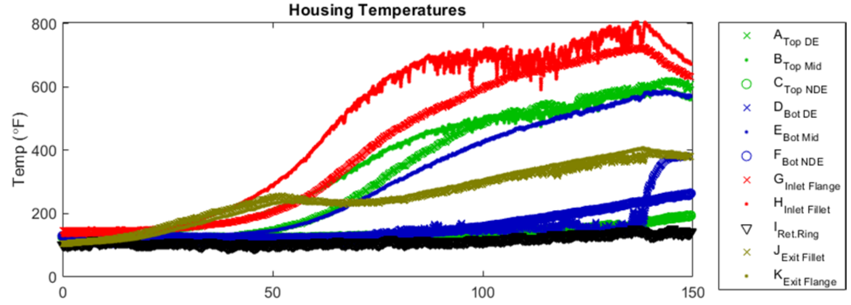




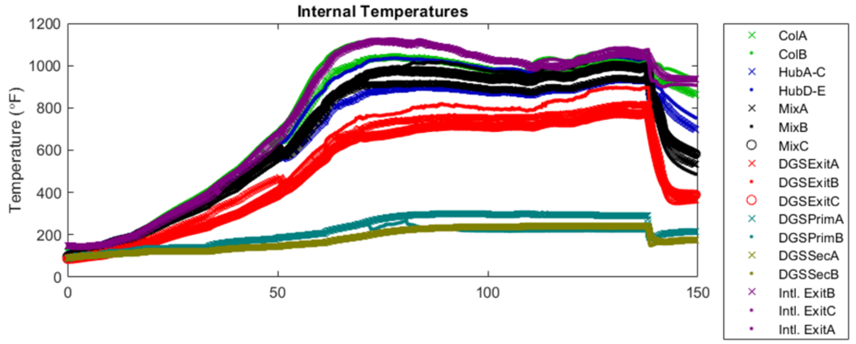
After some water side improvements, we successfully achieved 3765 psi at 1076 F

- Compressor discharge pressure of 3765 psi is the highest turbine pressure recorded to the teams knowledge
- During testing, extrusion on a backup sealing ring in the primary DGS led to the requirement for teardown and inspection of the DGS





Run	Maximum Pressure (psi)	Maximum Temperature (°F)	Reason for Termination
1	2739	1329	Cooler Limitation
2	2898*	1320*	Cooler Limitation
3	3654	985	Failed thermocouple
4	3765	1076	Seal Stationary Ring Extrusion



In conclusion

- We didn’t lose any pipes in Mexico during the test!

ACCOMPLISHMENTS/ CONCLUSIONS

Phase I Accomplishments

- Optimized the thermodynamic cycle and cost model for a CSP plant utilizing a compander (Achieved an LCOE of 6.0¢/kWh)
- Achieved 50% cycle efficiency
- Estimated 670 $/kWh capital cost for installed power plant items
- Designed a novel compander that is predicted to meet or exceed performance targets
  - Predicted 86% compressor efficiency with a range of 73.6%
  - Predicted 92% turbine efficiency with a targeted 50,000 hr. life

Phase II Accomplishments

- Successfully designed a reduced flow test facility to enable testing of the compander
- Procured facility components
- Fabricated and assembled the compander
- Performed mechanical testing of the compander

Phase II Accomplishments

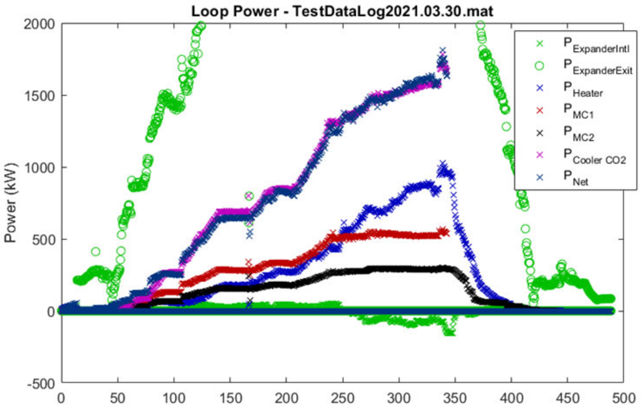
- Commissioned the test facility
- Performed multiple compressor builds and performed multiple tests
    - Achieved 69.5% range on the compressor
    - Power derived compressor measurements suggest an efficiency greater than 80%
- Turbine Inlet temperature successfully operated above 1320F (>720C)

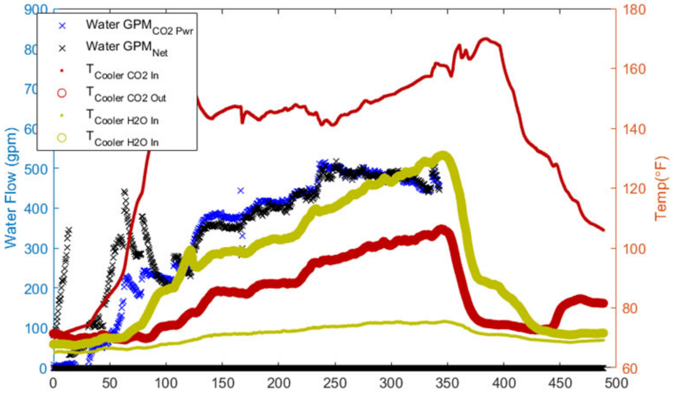
Program Firsts

- Highest pressure sCO_2 DGS in the world
- Highest pressure integrally geared expander
- Highest density integrally geared expander
- Highest density radial expander
- Highest temperature radial expander at pressure > 100 bar
- Highest pressure integrally geared compressor
- Highest density integrally geared compressor
- First functional sCO2 compressor driven indirectly fired turbine power cycle loop > 1MW

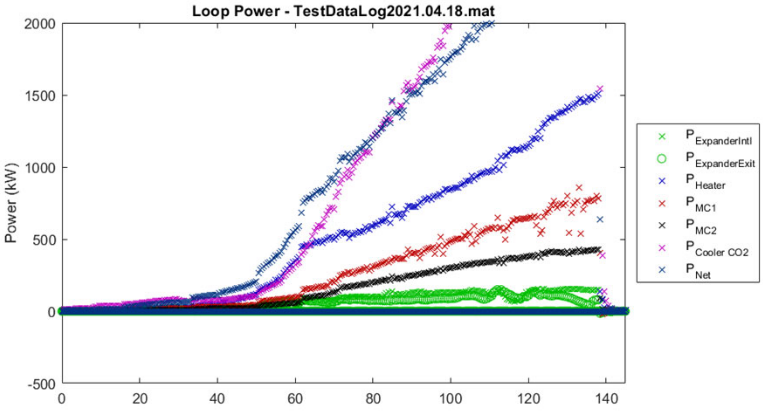
LESSONS LEARNED

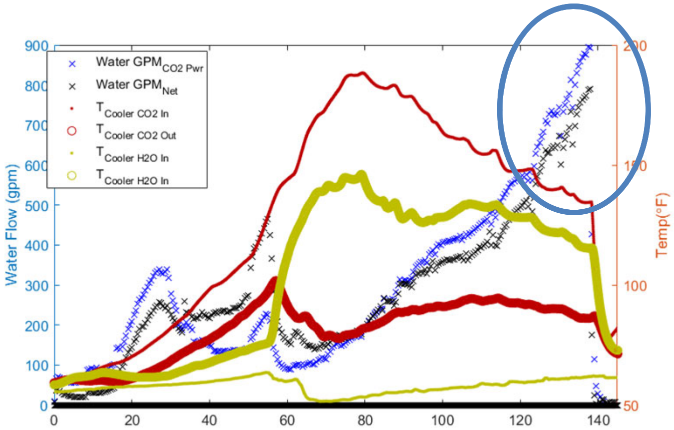
Loop Power balance simulations showed that water flow rate was drastically lower than anticipated

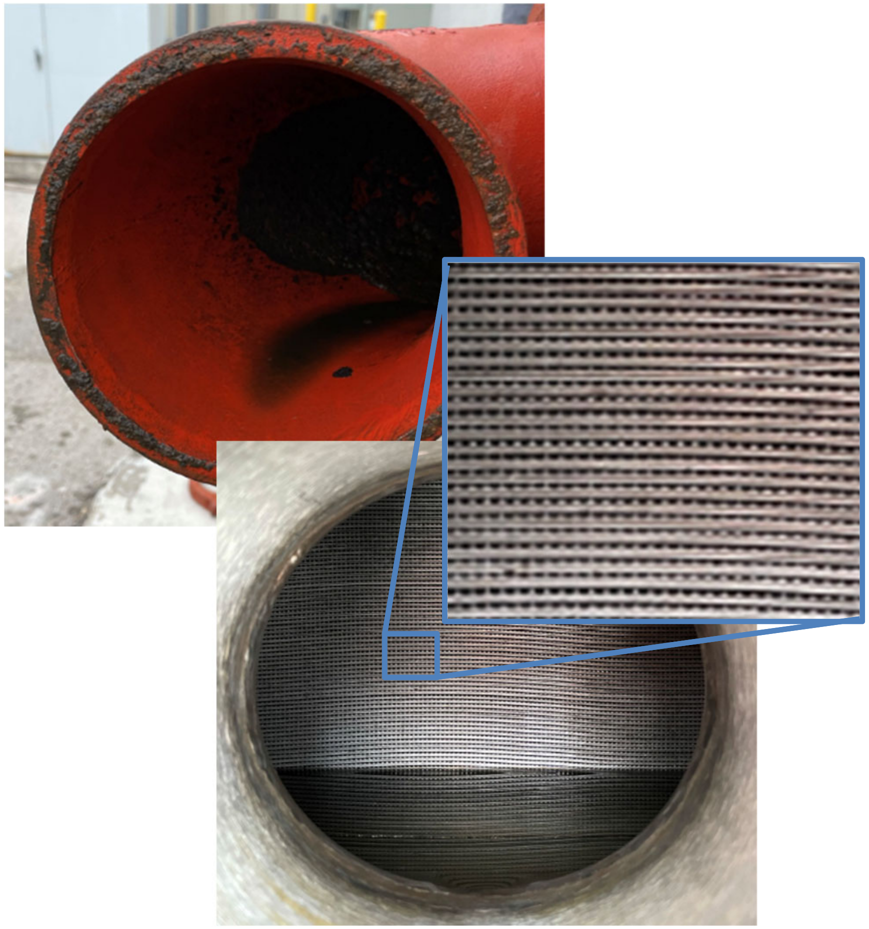




Cleaning the cooler with CLR showed a drastic improvement in water flow, allowing increased operating power for the loop



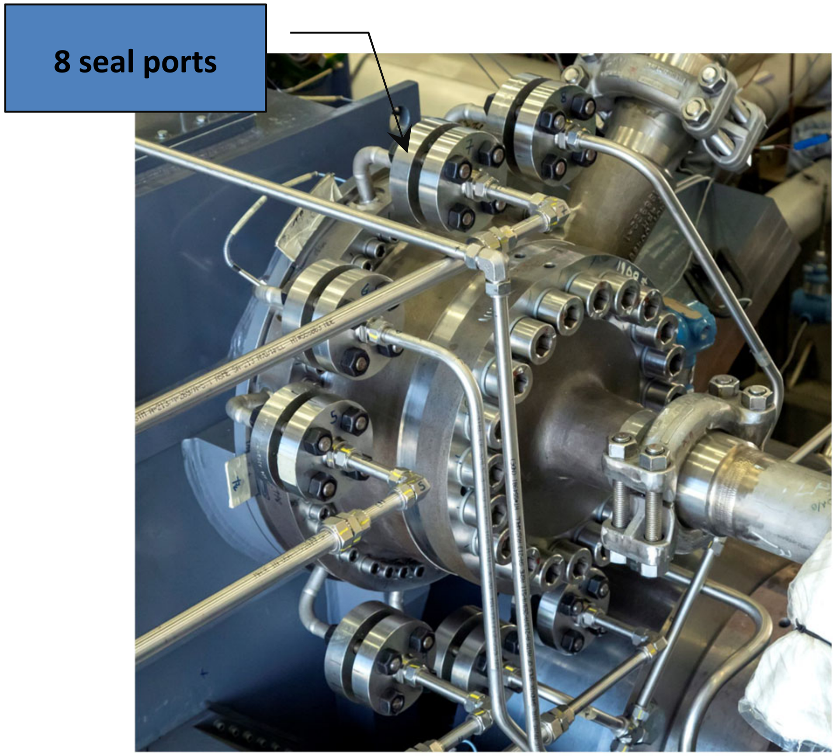




The damage valve was sent back for inspection, where a bent stem was observed

Machining debris found in the compressor seal ports

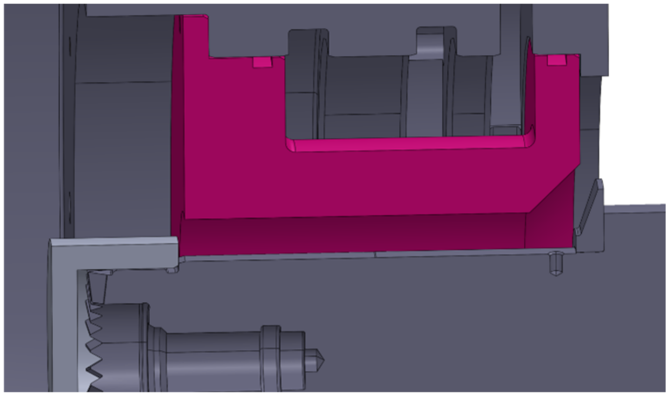
- During inspection of DGS ports to determine if oil was present, large burrs were found in the DGS ports that presented debris risk



Using a flexible choke cord, a deburring tool was constructed to debur the ports in place

Typical Machining Burr

DGS Port Plug



Flexible Deburring Tool

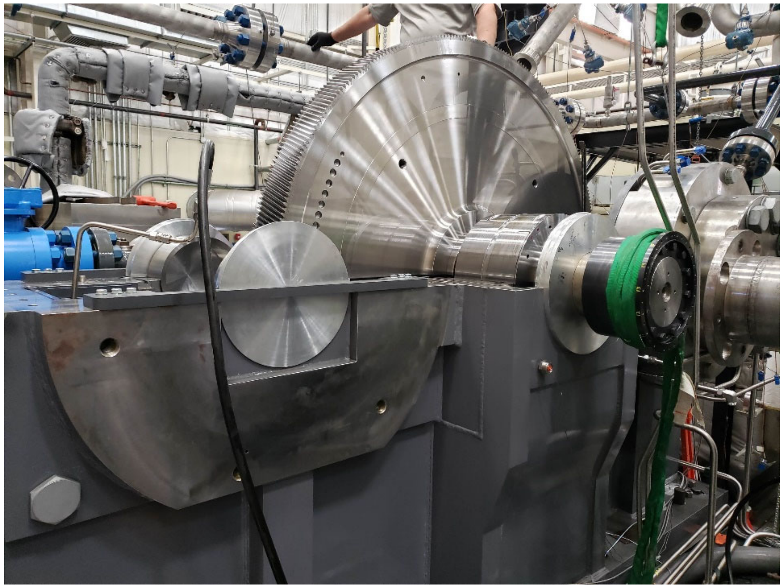
Post-cleaned holes

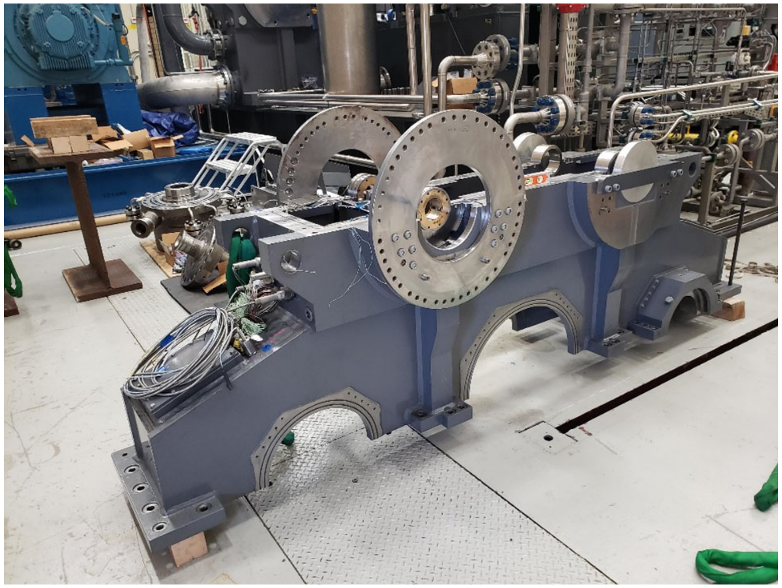
Dry Gas Seal Failure

1. **While operating it was observed that main gearbox vent had opened**
   - Vibration and temperature was all within normal operating conditions at the time of the DGS failure

→ 2. **Inspection found that the stage 2 DGS had failed**

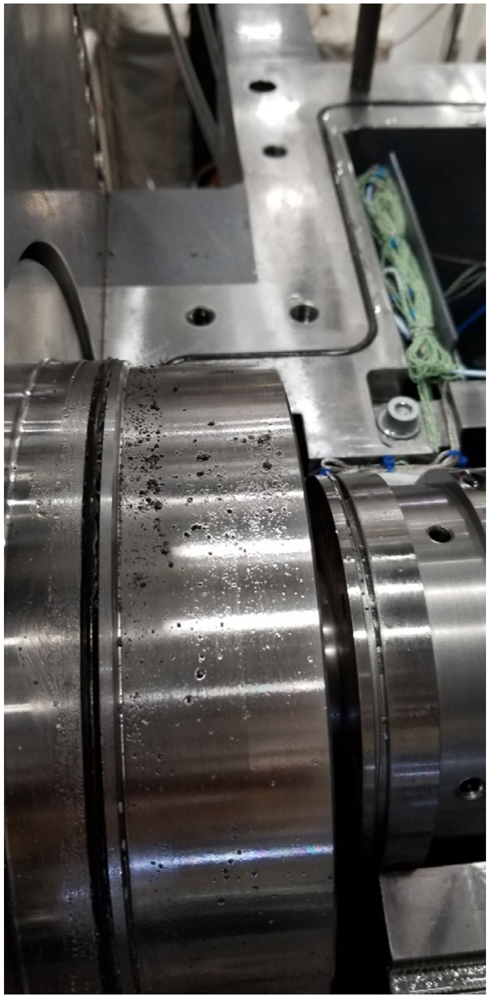
Disassembly



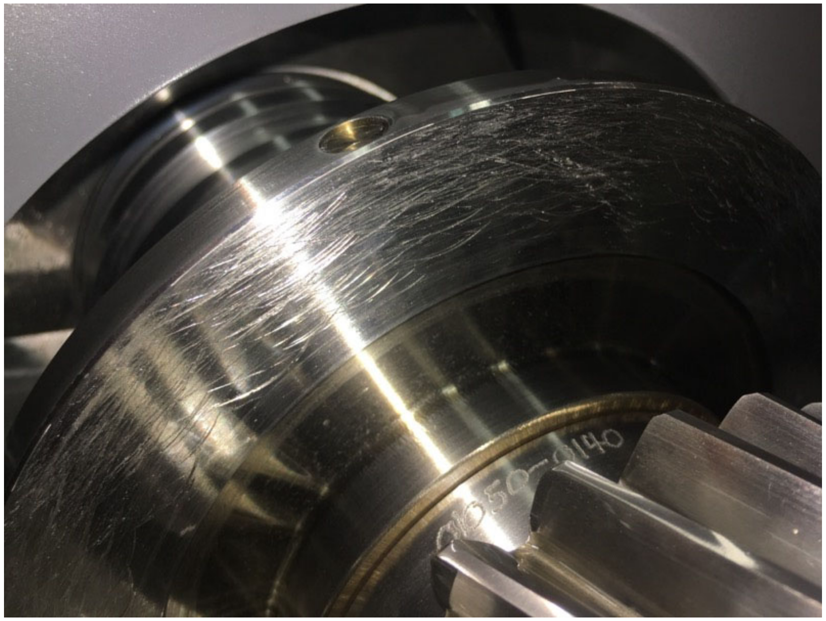


- **The core was disassembled to fully assess the damage**
  - Top cover of the gearbox was removed
  - Compressor pinion removed
  - Center section of the gearbox housing removed
  - Expander pinion removed
  - Bull gear removed.

Unfortunately, the seal was not the only damage



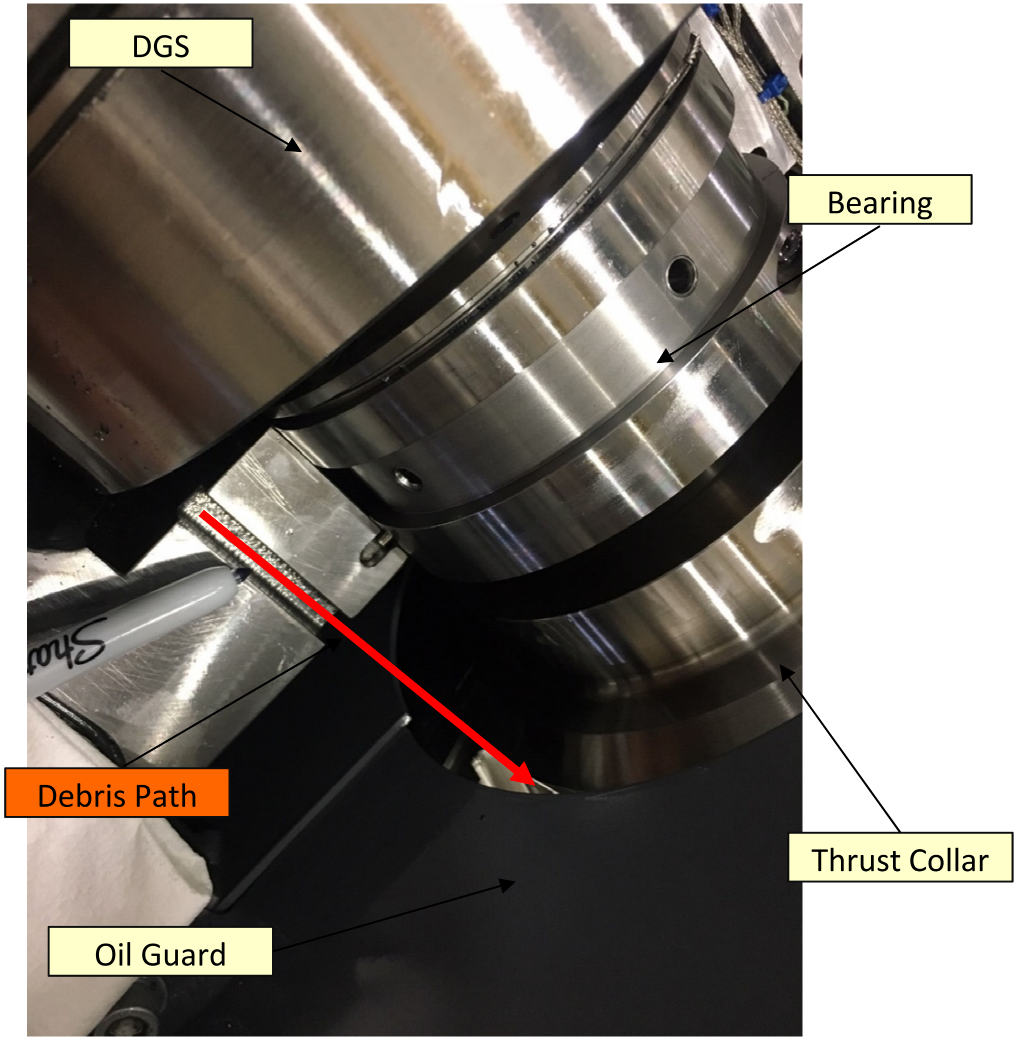
Debris from the MC1 DGS rings passed into the gearbox



Debris from the DGS rings caused scoring and scratches on the thrust surfaces

DGS Failure Lead to Secondary Damage

- Additional damage to the pinons was observed
- It was concluded that an open instrumentation port allowed some of the DGS debris to pass into the gearbox, impinging on the thrust face
- Damaged parts (Rotors, DGS and Bull Gear) were removed
- DGS will be replaced
- Bull gear and pinons will be refinished to remove scoring and scratches



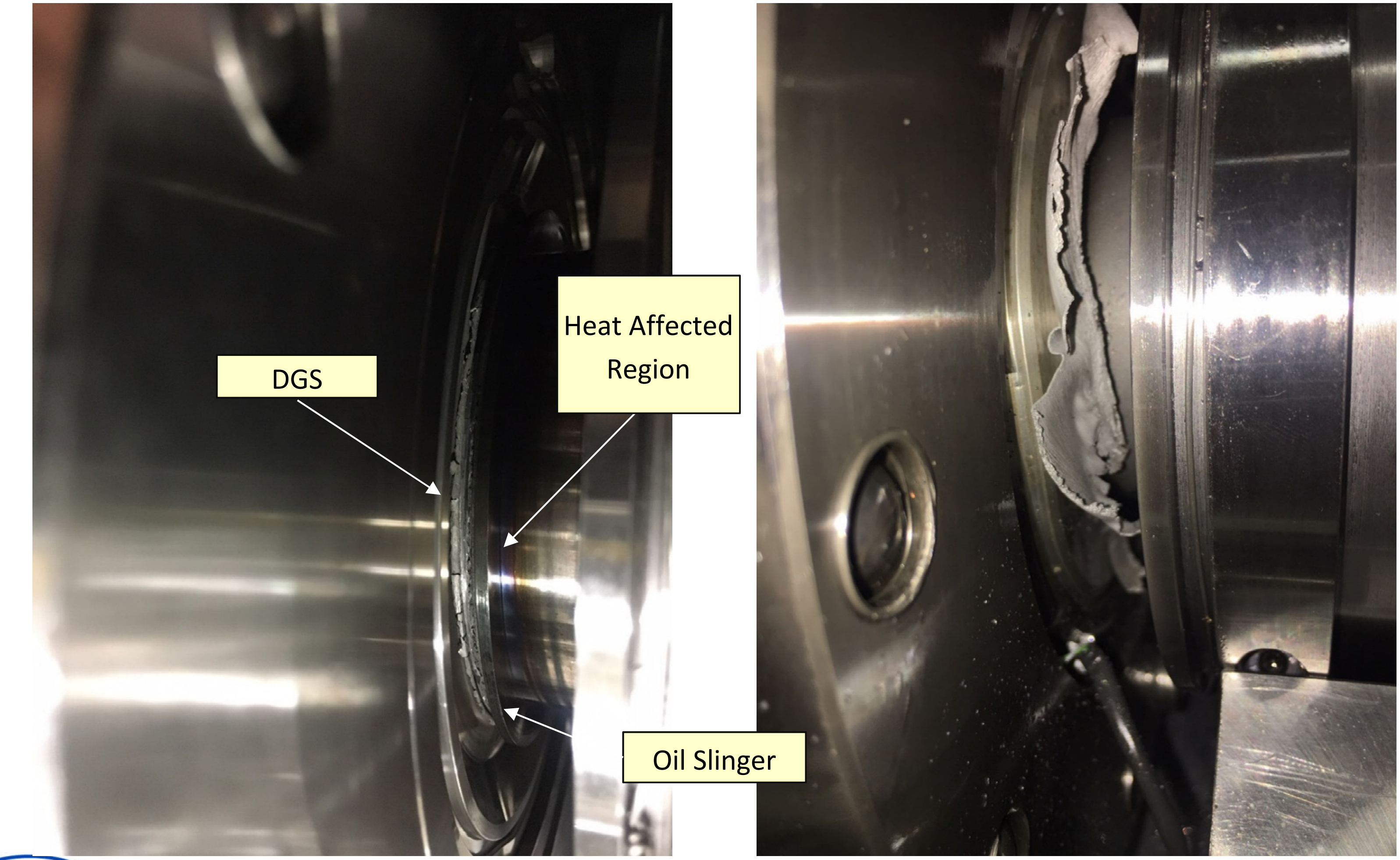
Root Cause Analysis of DGS Damage

A detailed Fault Tree Analysis is being conducted to determine root cause. There is much data to review but a general consensus is forming around the following chain of events:

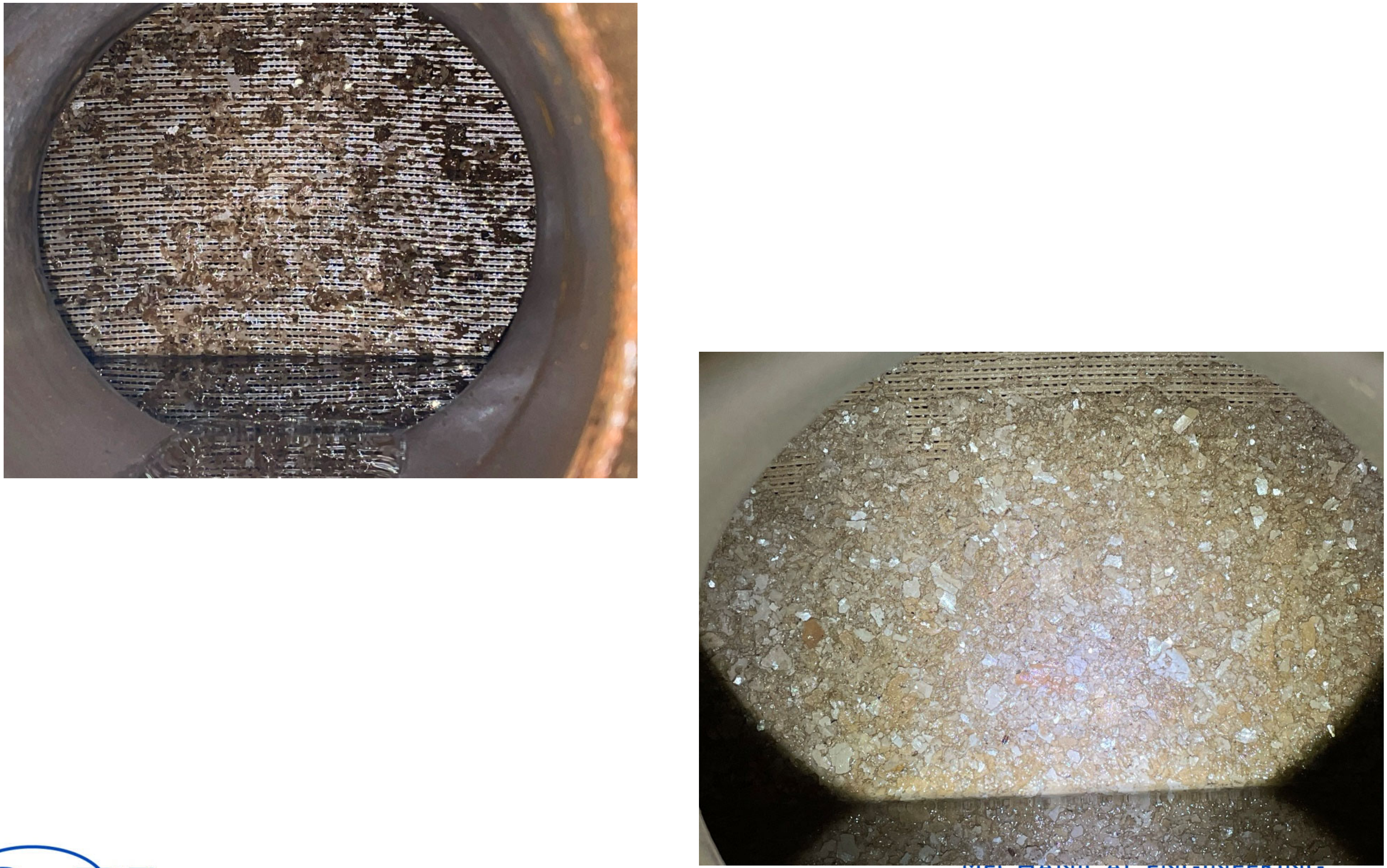
1. The shimming for the MC2 DGS was found to be incorrect during disassembly following the seal failure. This would have allowed contact between the stationary separation seal laby and the rotating slinger on the shaft once a back pressure had been established on the primary vent.
2. The rubbing will have produced axial vibration in the compressor pinion.
3. In combination with this perturbation, the check valve separating the seals was leaking and a flange gasket was found missing on the DGS panel, resulting in insufficient backpressure.
4. DGSs are susceptible to contact between the rotating and stationary seal rings if the gas film stiffness is insufficient to maintain separation. It was not possible to establish the desired back pressure on the MC1 primary vent at the low pressure operating conditions (~10 to 13 bar) present during these early runs, though it was possible to do so on the MC2 primary vent. This meant that the MC1 secondary seal would have had a much lower gas film stiffness between its seal faces than its primary seal nor both the primary and secondary seals of the MC2 DGS.
5. Data trending indicates that at around 3:20 pm on 14th August, sufficient damage had occurred to the MC1 secondary seal that it could no longer maintain any back pressure at all.
6. Despite the damage and rubbing taking place, temperature measurements from within the DGSs remained within limits and so, though no back pressure could be generated on the vent, no alarms were generated. Vibration trends also did not seem out of place.
7. Testing continued intermittently due to control system debugging until MC1 seal ring total failure occurred on 21st August as a result of accumulated damage.

Shim found missing on the DGS

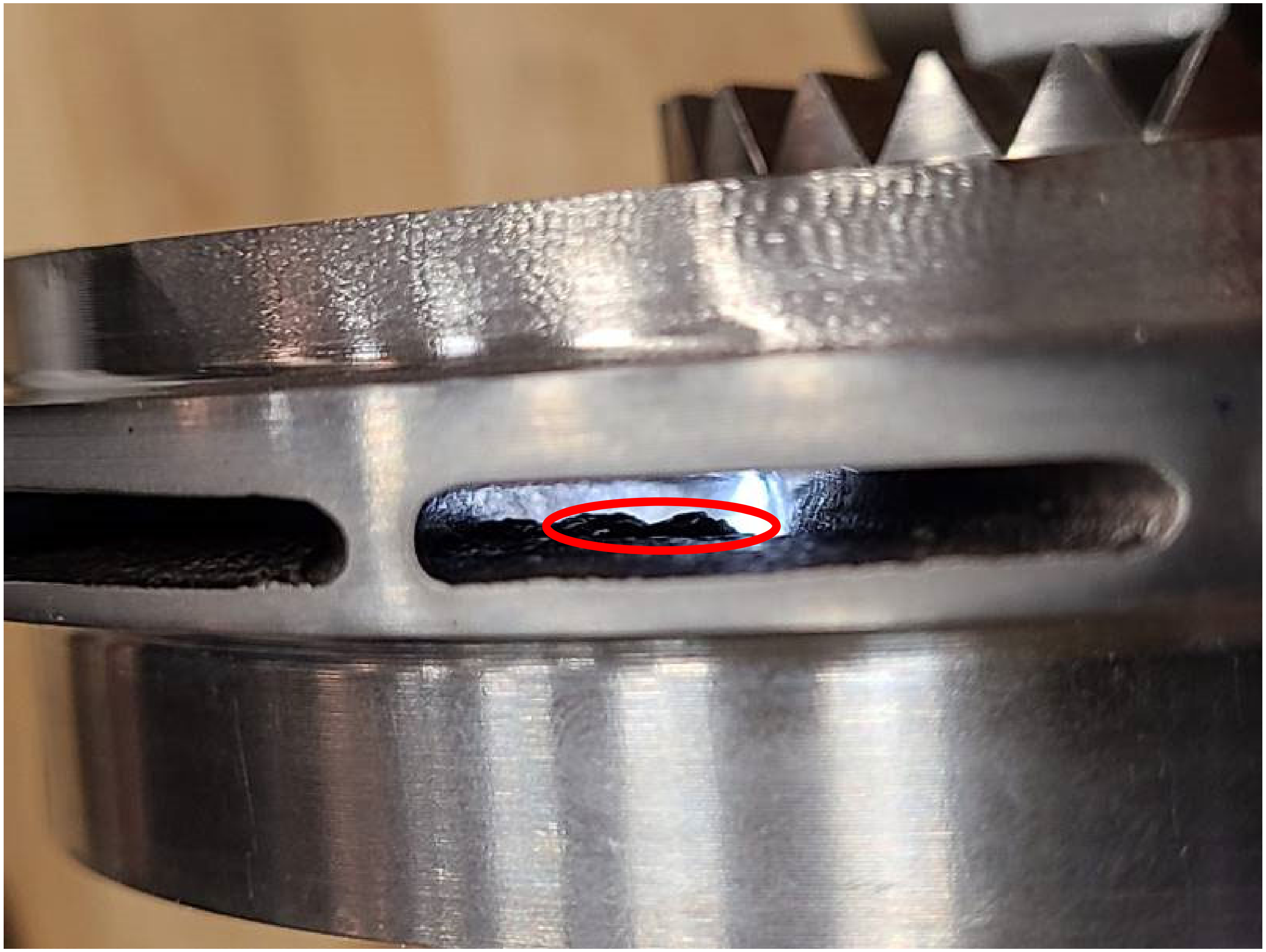
Incorrect Shimming Resulted in Rubbing



PCHEs are great for closed water systems – our facility was not originally design with the use of PCHEs in mind

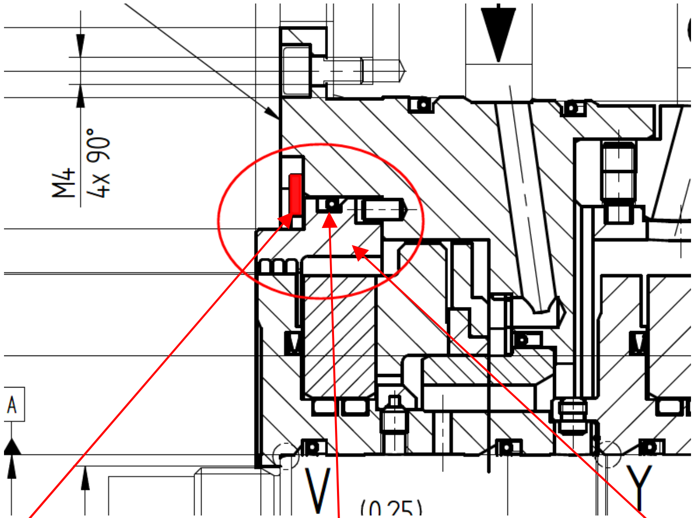


With a development project with several thousand moving parts, sometimes even an obvious flaw gets missed.



With relatively little industry experience in sCO2, vendors are also performing prototype work for "standard parts"

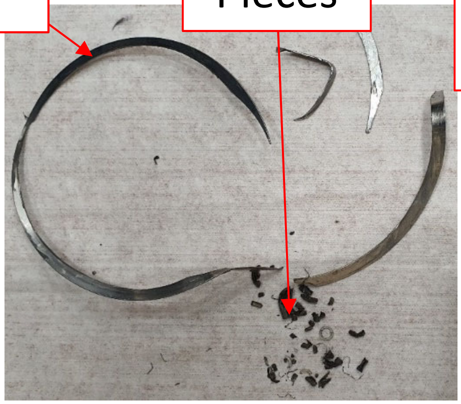
DGS Still Installed – Process Side Laby and Retaining Snap Ring Missing

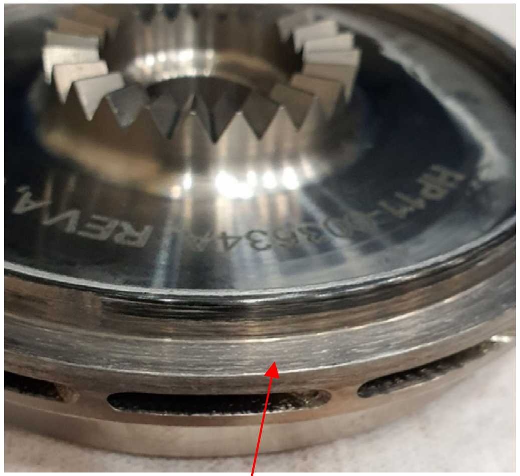


Snap Ring

O-ring Pieces

Process Side Laby





Process Side Laby Has rubbed on Impeller

Questions?

Jason Wilkes
Jason.wilkes@swri.org

Lessons learned

- Mitigation of expander wheel locking on shaft due to high-temperature/high-pressure
- Compressor performance measurement requires atypical instrumentation beyond PTC-10
- DGS remain a reliability risk for sCO2 machines
- First-time AM parts are still risky (tolerances not dialed in, heat treat)
- Off design excitation of subsynchronous vibration (due to impact type excitation, not traditional instability)
- Main compressor performance is still difficult to predict with typical CFD approach (discrepancy between Numeca and craft-tech)
- Piping is still a significant system cost (at least for an R&D loop)
- Observed stable turbine pressures even as comp inlet T fluctuated causing other loop transients - that's an interesting result.

Lessons learned

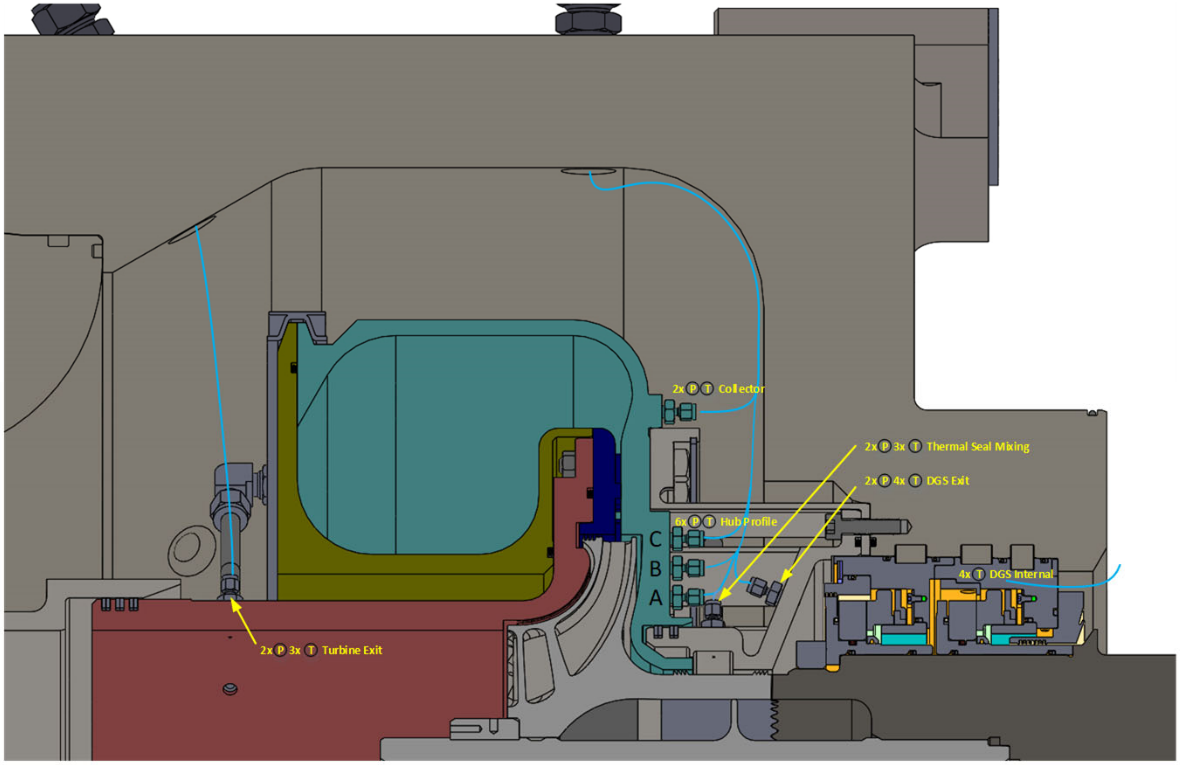
- Compressor eye seal leakages were higher than expected based on standard design criteria
- All seals in the system should be resistant to RGD failure
- The mechanical design should have provisions to mitigate damage in case of a DGS failure
- Casing internal cross-drilling should be inspected with a bore scope to ensure they are clear after manufacturing
- Critical to ensure that the 3D printed impellers bore is machined concentric to the axis of rotation of the printed flowpath
- The amount of material removed by surface finishing of 3D printed parts must be understood and accounted for
- Use of a PCHE as a cooler in an open water loop is prone to clogging
- Compressor performance is very sensitive to inlet temperature. Alternate methods should be considered to measure compressor efficiency.

Lessons learned

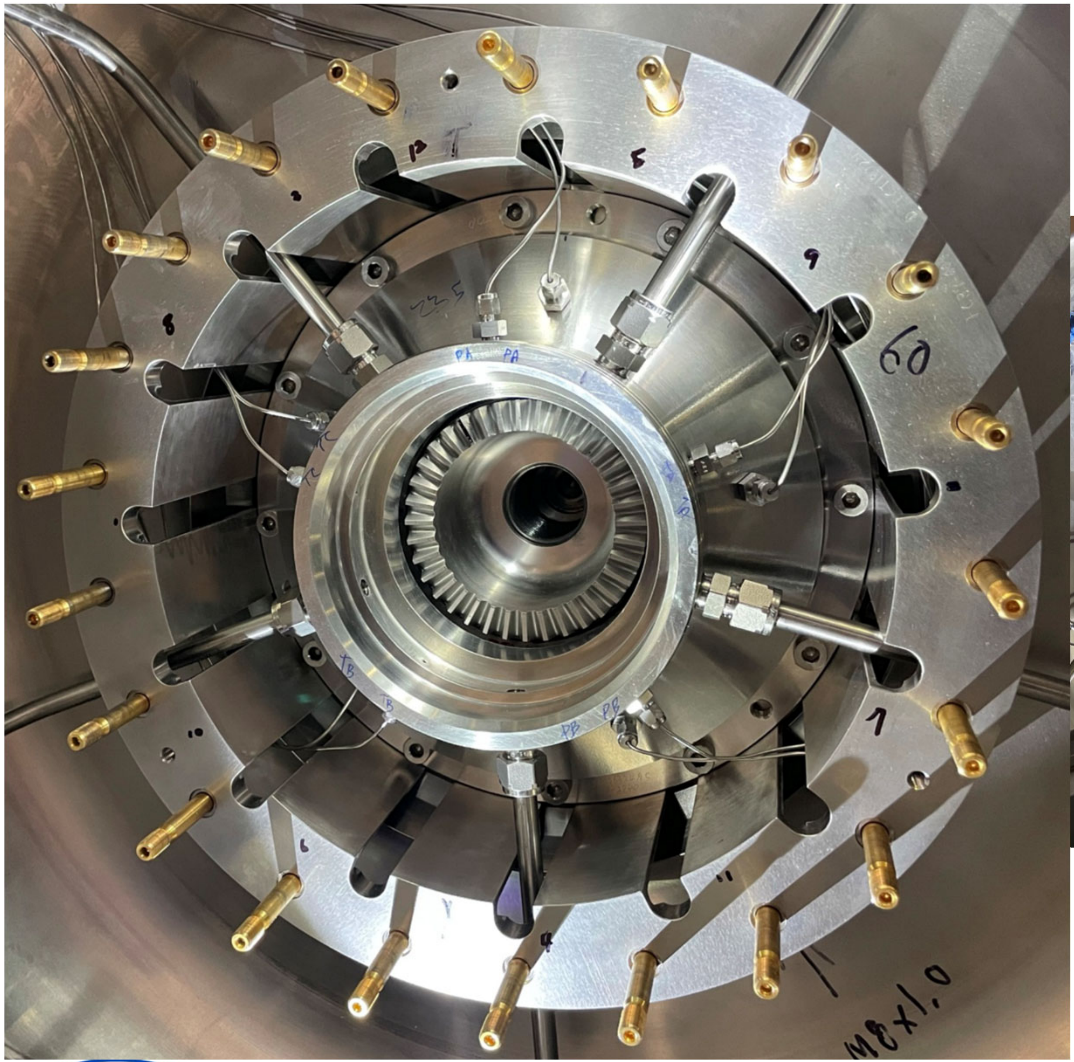
- Evaluate dry gas seal panel design at low pressures as well as supercritical pressures to ensure sufficient flow capacity for the pressure differences available.
- Due to range of conditions covered and safety concerns, include more remote control of equipment on the dry gas seal panel.
- Evaluate loop control valves for capacity at low and supercritical pressures to ensure sufficient flow capacity for the pressure differences available. Also plan for off-design and reduced performance on components to ensure that even if component performance suffers, a high degree of flexibility is still obtained.

How do we validate thermal management

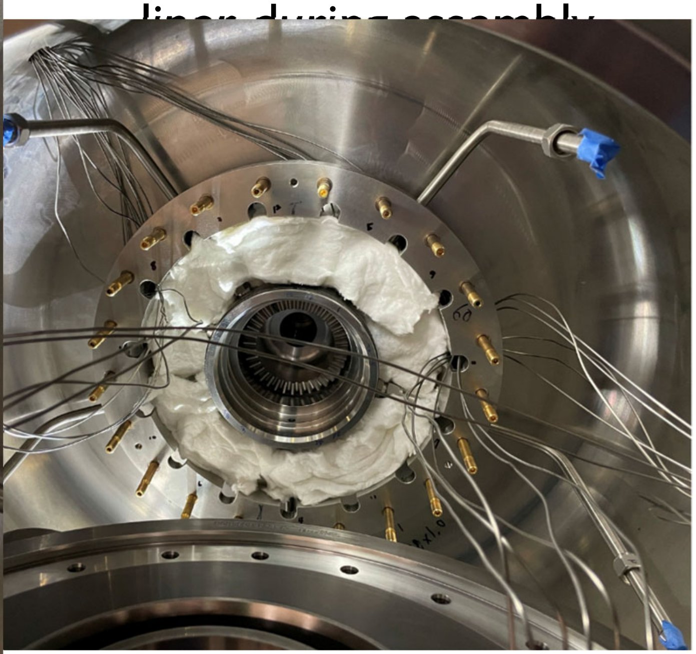
- 19 internal temperatures
- 18 internal pressures



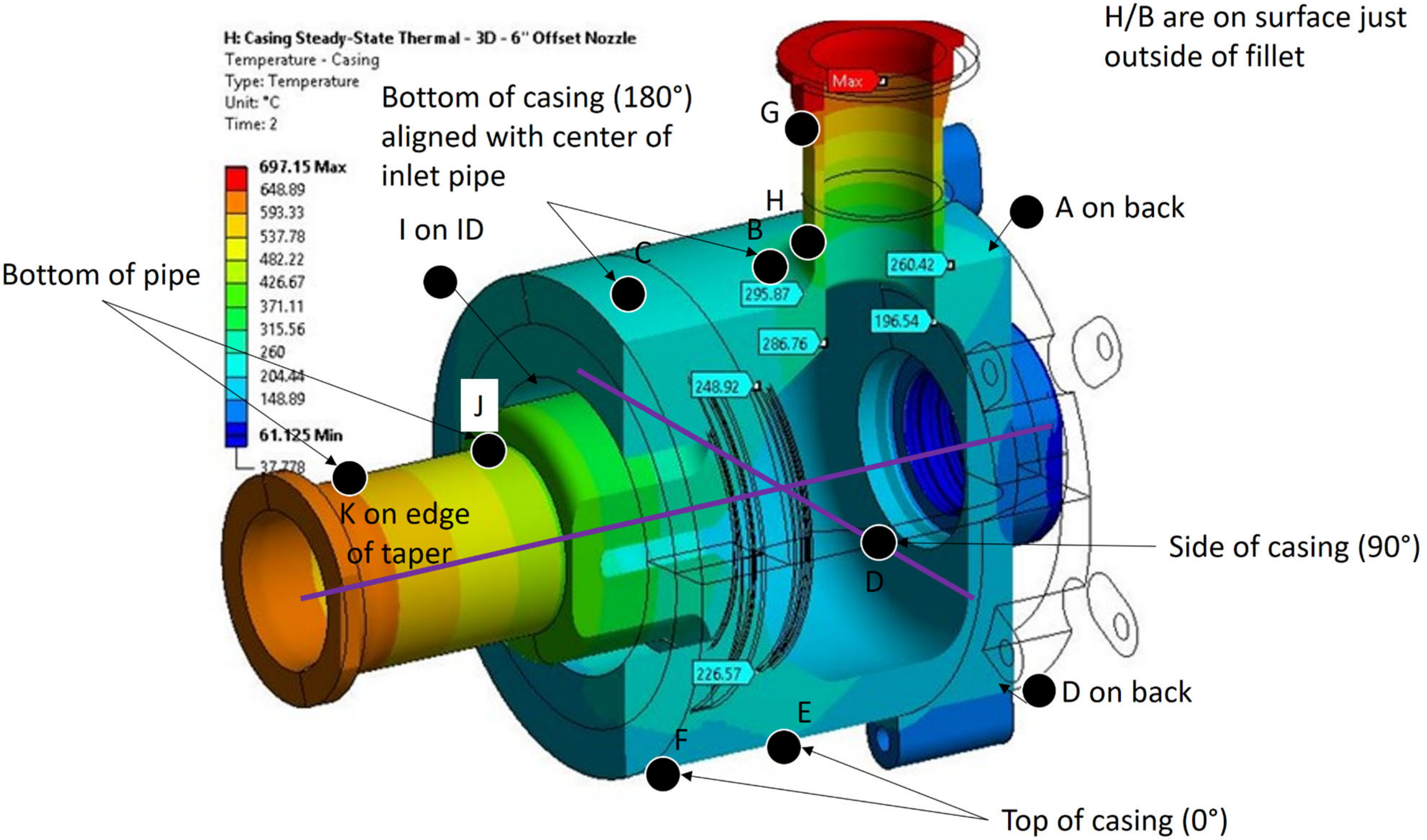
Internal instrumentation was added to help validate our thermal management approach (insulated casing)



- Thermocouples and pressure transducers were installed on the liner during assembly



Thermocouples were also added to the housing to determine housing temperatures



H/B are on surface just outside of fillet

Bottom of casing (180°) aligned with center of inlet pipe

I on ID

Bottom of pipe

A on back

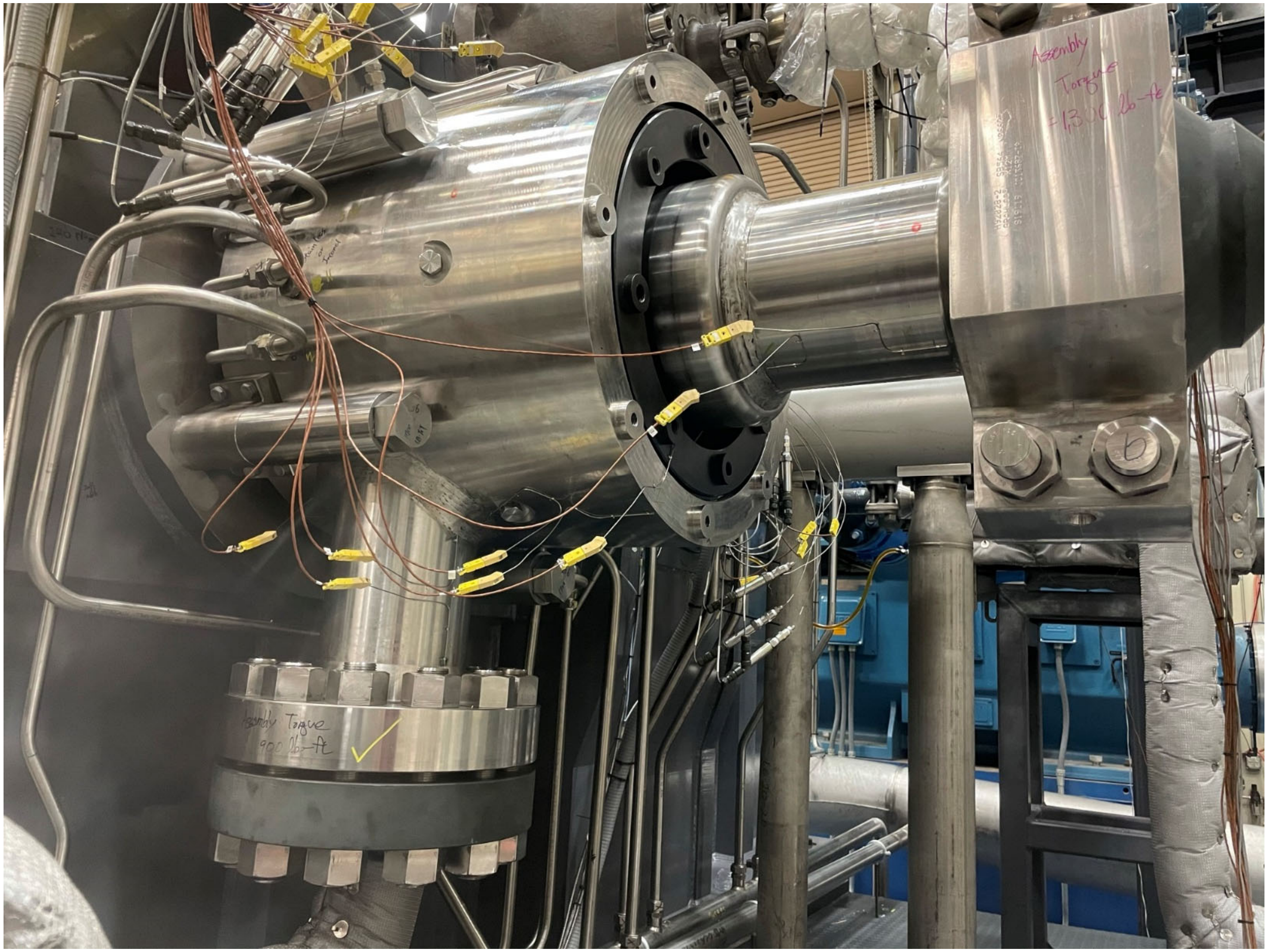
K on edge of taper

Side of casing (90°)

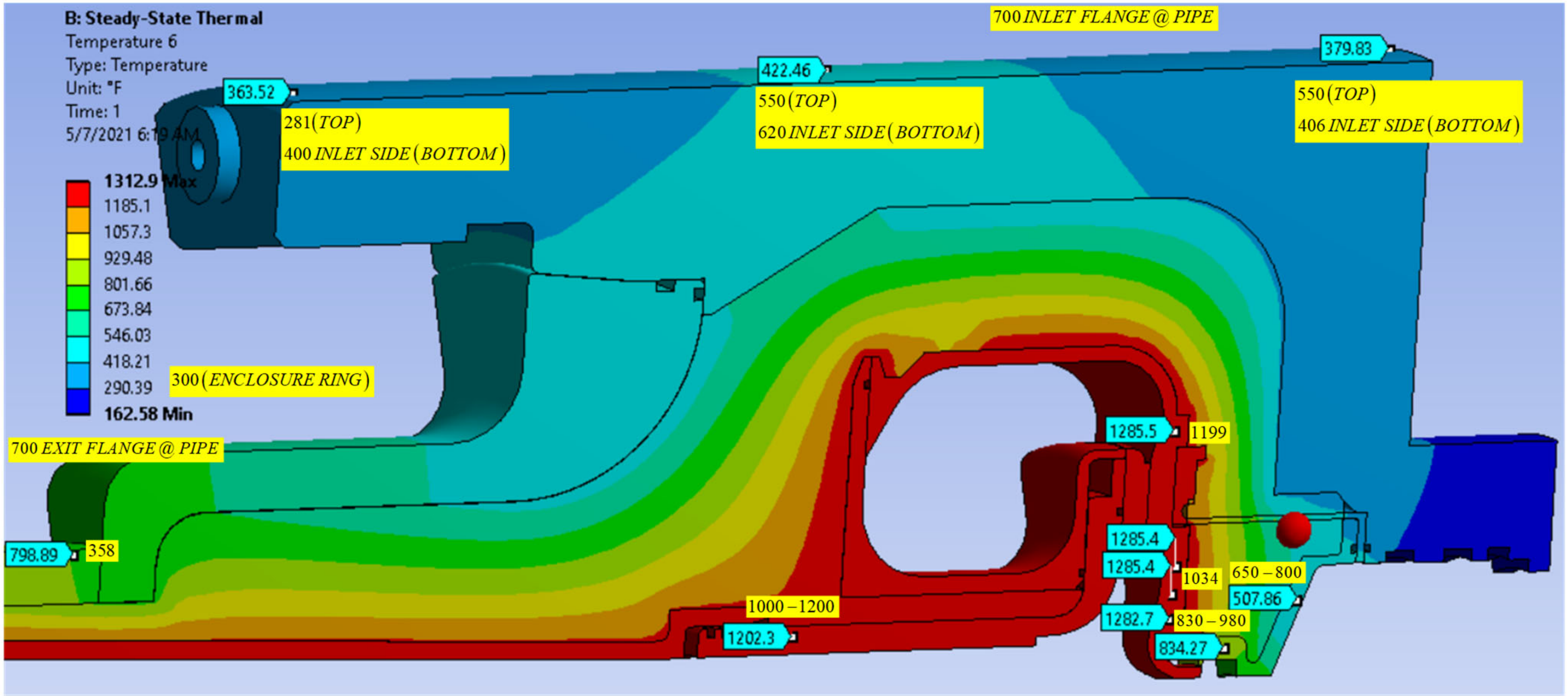
D on back

Top of casing (0°)

Turbine assembly completed with casing thermocouples attached



Comparison between measured and predicted expander temperatures shows gooed agreement



Turbine disassembly showed good condition of parts