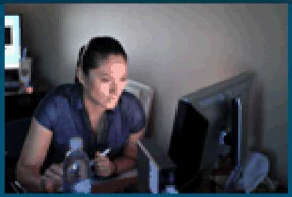
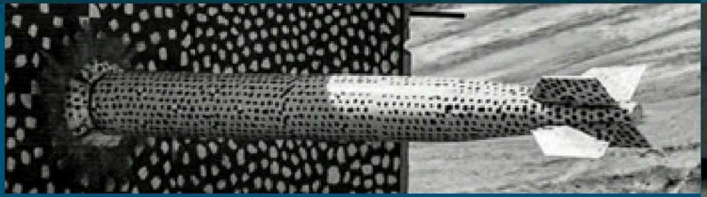
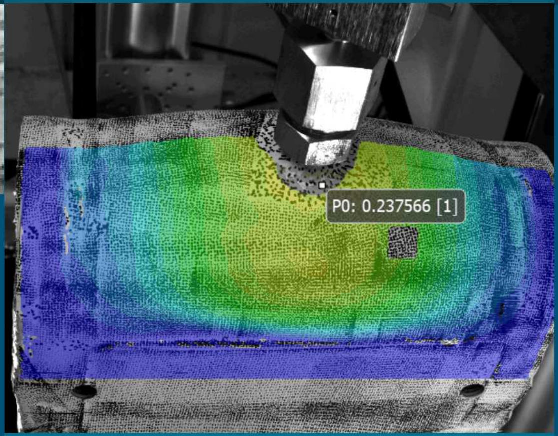
SAND2019-6884C

Compact Heat Exchanger Semi-circular Header Burst Pressure and Strain Validation







PRESENTED BY

Blake W. Lance, Matthew D. Carlson

SAND2019-XXXX

ASME Turbo Expo 2019, Phoenix, AZ, June 17-21, 2019

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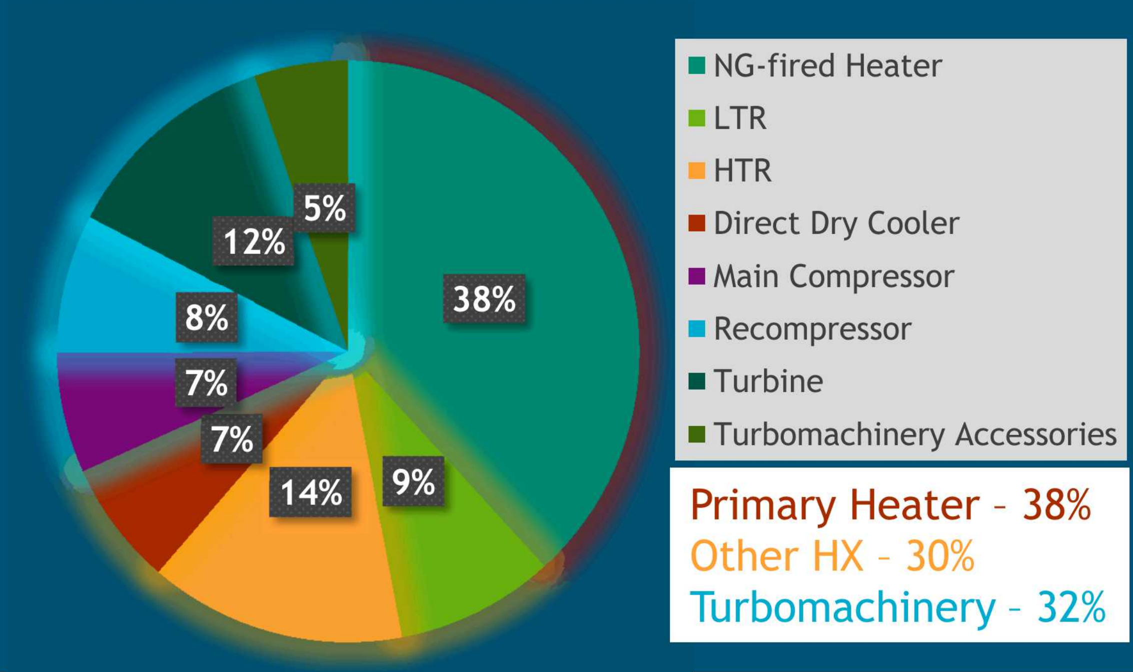
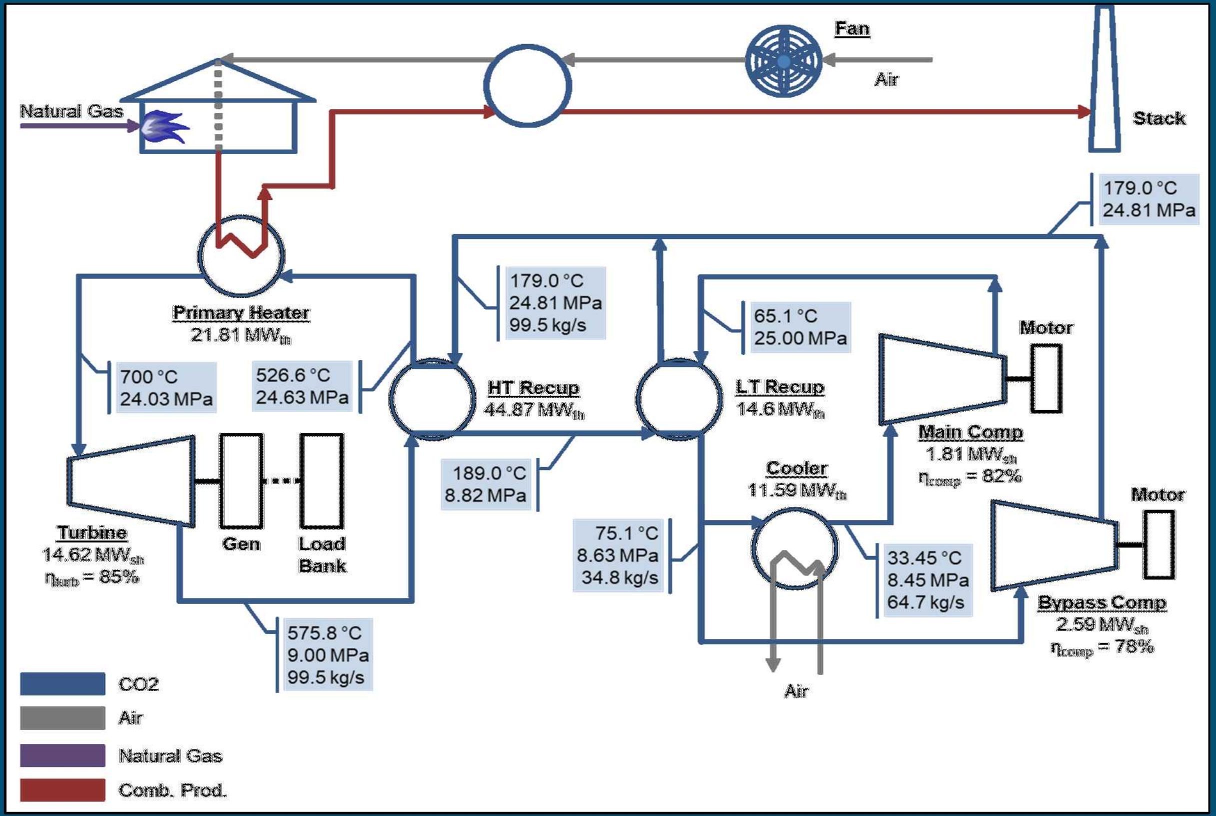
Heat exchangers in sCO_2 power cycles are a sizeable portion of component costs

- The heat duties of the recuperators is about 10x larger than the net power output
  - 10 MWe net power
  - 22 MWth Primary Heater
  - 45 MWth High Temperature Recuperator
  - 15 MWth Low Temperature Recuperator
  - 12 MWth Cooler



Primary Heater – 38%
Other HX – 30%
Turbomachinery – 32%

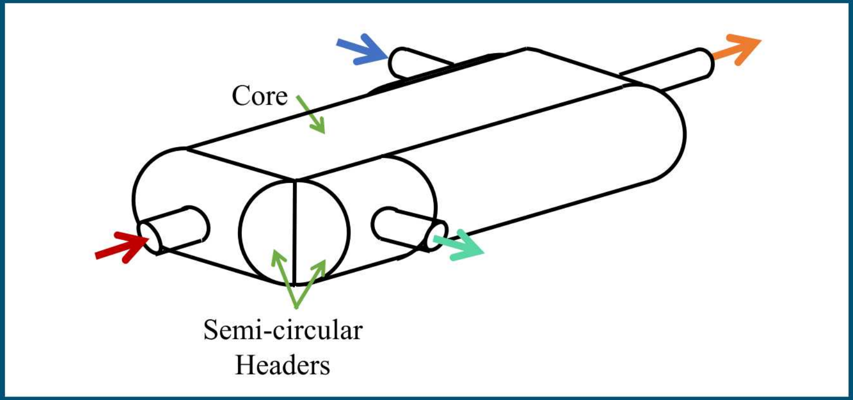


10 MWe STEP Project Conditions, NETL

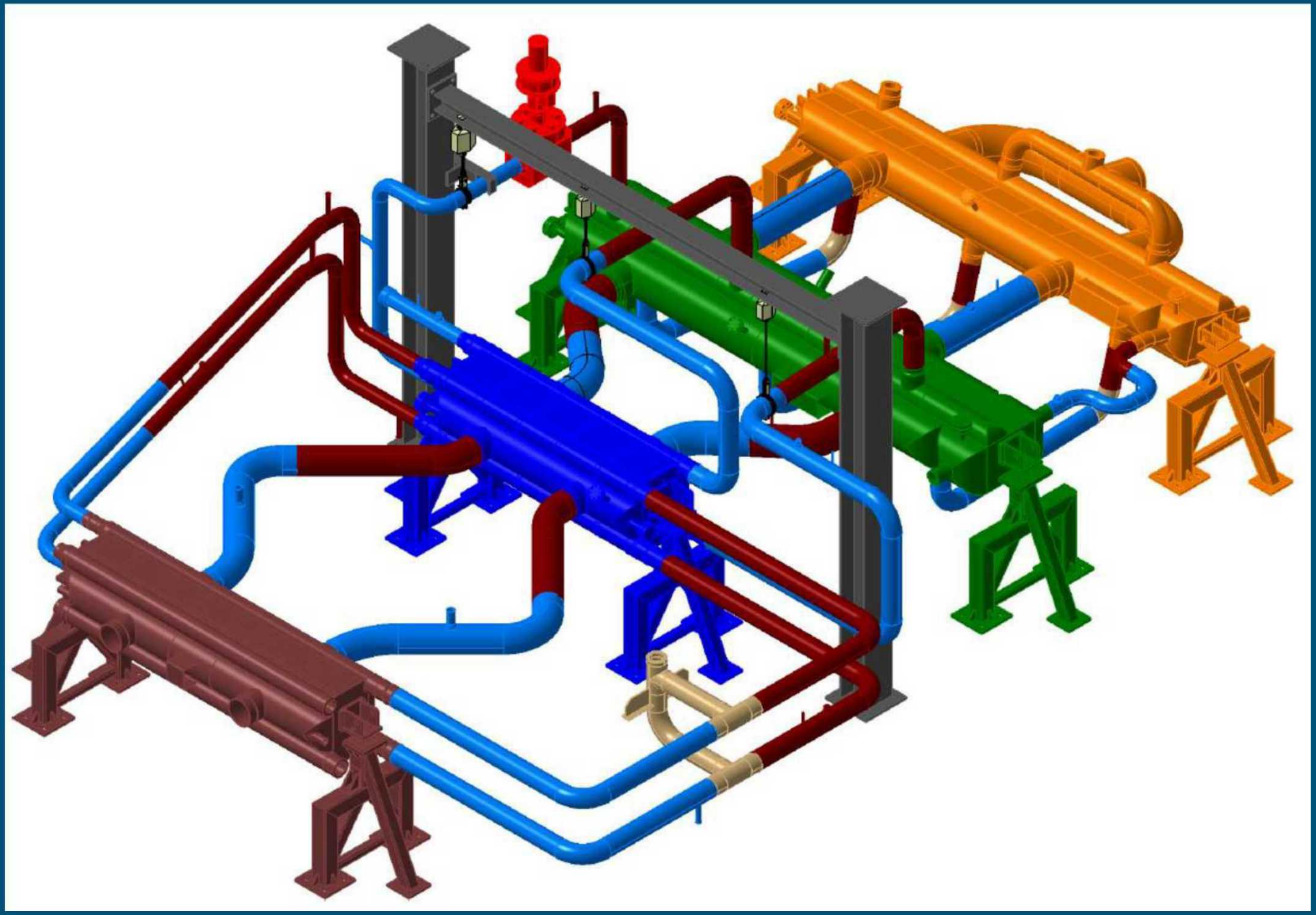
Weiland, Nathan T., Lance, Blake W., and Pidaparti, Sandeep R., "sCO2 Power Cycle Component Cost Correlations from DOE Data Spanning Multiple Scales and Applications", 2019 ASME Turbo Expo, June 17-21, 2019, Phoenix, AZ.

Semi-circular headers are a critical part of compact heat exchangers for sCO_2 power cycles

- sCO_2 power cycles are heat exchanger intensive
- Semi-circular headers are a common header design



Simplified compact heat exchanger design with semi-circular headers



Heat exchanger train initial design for Net Power direct-fired plant.
Allam, R., Martin, S., Forrest, B., Fetvedt, J., Lu, X., Freed, D., Brown, G. W., Sasaki, T., Itoh, M., and Manning, J., 2017, "Demonstration of the Allam Cycle: An update on the development status of a high efficiency supercritical carbon dioxide power process employing full carbon capture", Energy Procedia.

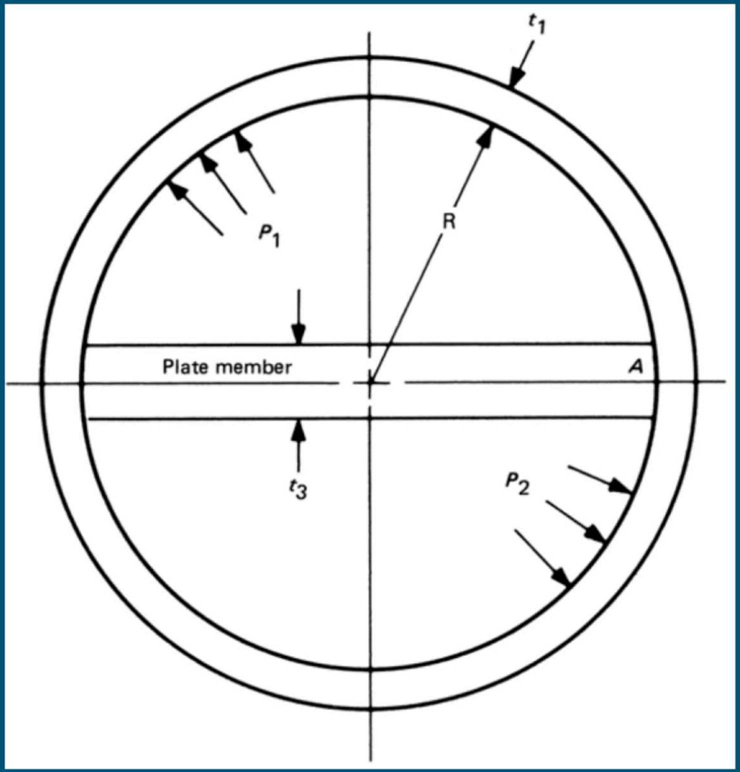
The ASME Boiler and Pressure Vessel Code (BPVC) is often used to design semi-circular headers

- There are likely several design methods that apply
- This work used design by equation methods of the ASME BPVC, Section VIII, Division 1, Mandatory Appendix 13, section 13 for "Vessels of Circular Cross Section Having a Single Diametral Staying Member"
- The Plate in the BPVC can be considered the diffusion-bonded core

13-13 VESSELS OF CIRCULAR CROSS SECTION HAVING A SINGLE DIAMETRAL STAYING MEMBER [FIGURE 13-2(C)]

For the equations in these paragraphs, the moments of inertia are calculated on a per-unit-width basis. That is, $I = bt^3/12$, where b = 1.0. See 13-4(k).

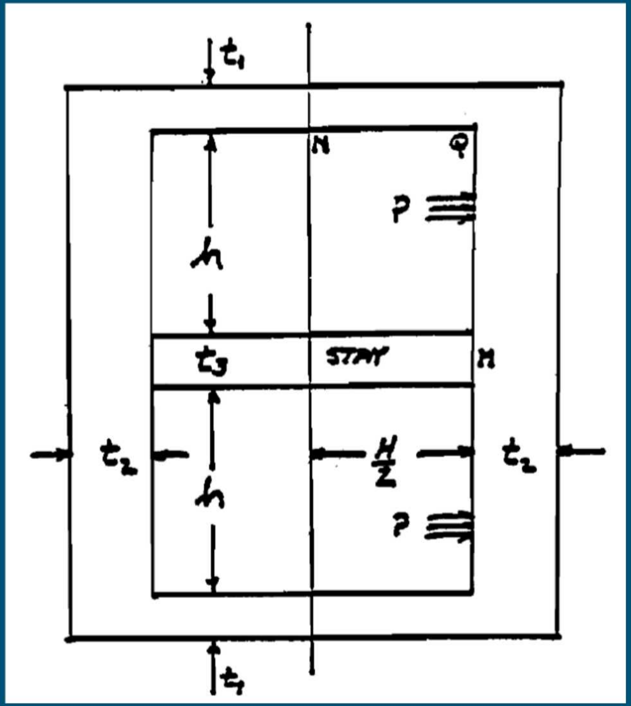
(a) The cylindrical shell and diametral stay plate are sized such that the various vessel members will not be overstressed when there is full pressure in both vessel compartments or when there is full pressure in one compartment and zero pressure in the other compartment.



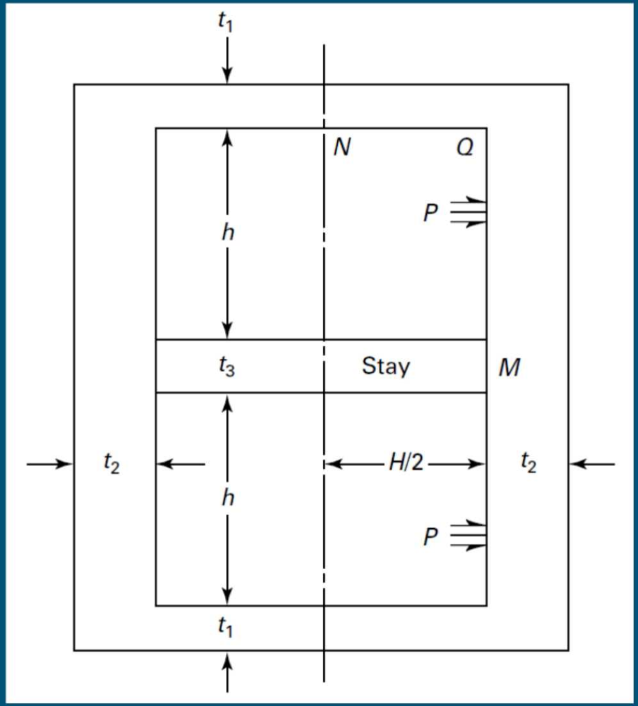
Vessel of circular cross section with diametral stay plate. The American Society of Mechanical Engineers, 2017, "2017 ASME Boiler & Pressure Vessel Code," Section VIII, Division 1, New York.

The ASME BPVC is based on Castigliano's Theorems

- Castigliano's Theorems were originally documented in his publication from 1879 and written in French
- Southwell published some discussion on this theorem in 1923 in English
- Simply restated, "material will elastically deform in a way that minimizes the total strain energy"
- …But there is no known validation of the theorems and the implementation in the BPVC



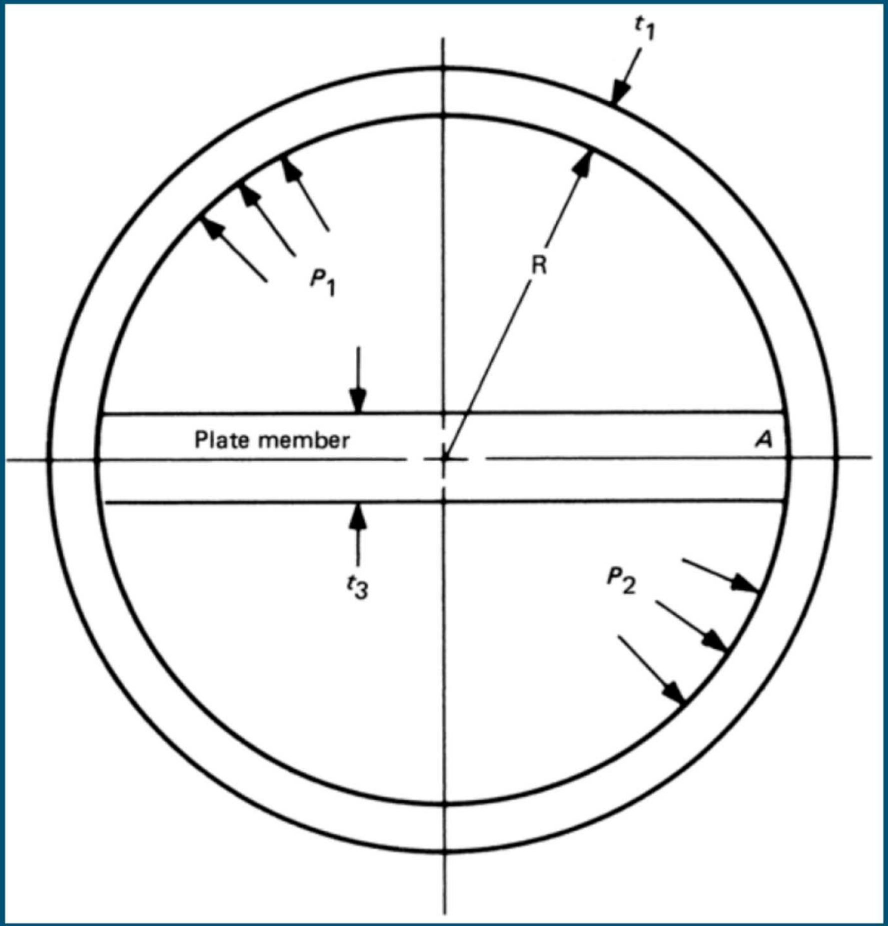
Faupel, J., 1979, "Pressure Vessels of Noncircular Cross Section (Commentary on New Rules for ASME Code)," Journal of Pressure Vessel Technology, 101(3), pp. 255-267.



2017 ASME BPVC, Sec. VIII, Div. 1

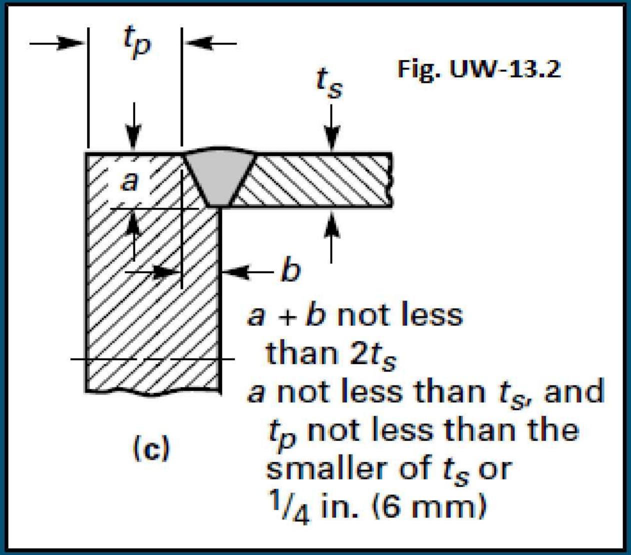
The stress equation requirements drove the design

- Three geometric design parameters
  - Shell thickness, t_{shell}
  - Plate thickness, t_{plate}
  - End cap thickness, t_{cap}
- Stress equations
  - $S_{T,\text{shell}} = \frac{PR}{t_{\text{shell}}} + \frac{c}{I}\left[\frac{2Pt_{\text{shell}}^2}{3(\pi^2-8)}\right],\ c = \frac{t_{\text{shell}}}{2},\ I = \frac{bt^3}{12},\ b = 1.0$
  - $S_{T,\text{plate}} = \frac{2\pi P t_{\text{shell}}^2}{3Rt_{plate}(\pi^2-8)}$
- Stress requirements
  - $S_{\text{membrane,shell}} = \frac{PR}{t_{\text{shell}}} \le SE \Rightarrow t_{\text{shell}} \ge \frac{PR}{SE}$
  - $S_{T,\text{shell}} = \frac{PR}{t_{\text{shell}}} + \frac{4P}{\pi^2-8} \le 1.5SE \Rightarrow t_{\text{shell}} \ge \frac{PR(\pi^2-8)}{1.5SE(\pi^2-8)-4P}$
  - $S_{T,\text{plate}} = \frac{2\pi P t_{\text{shell}}^2}{3Rt_{\text{plate}}(\pi^2-8)} \le SE \Rightarrow t_{plate} \ge \frac{2\pi P t_{\text{shell}}^2}{3R(\pi^2-8)SE}$
  - $t_{cap} = d\sqrt{\frac{ZCP}{SE}},\ Z = 3.4 - \frac{2.4d}{D},\ Z \le 2.5,\ C = 0.2 \Rightarrow t_{cap} \ge R\sqrt{\frac{0.44P}{SE}}$



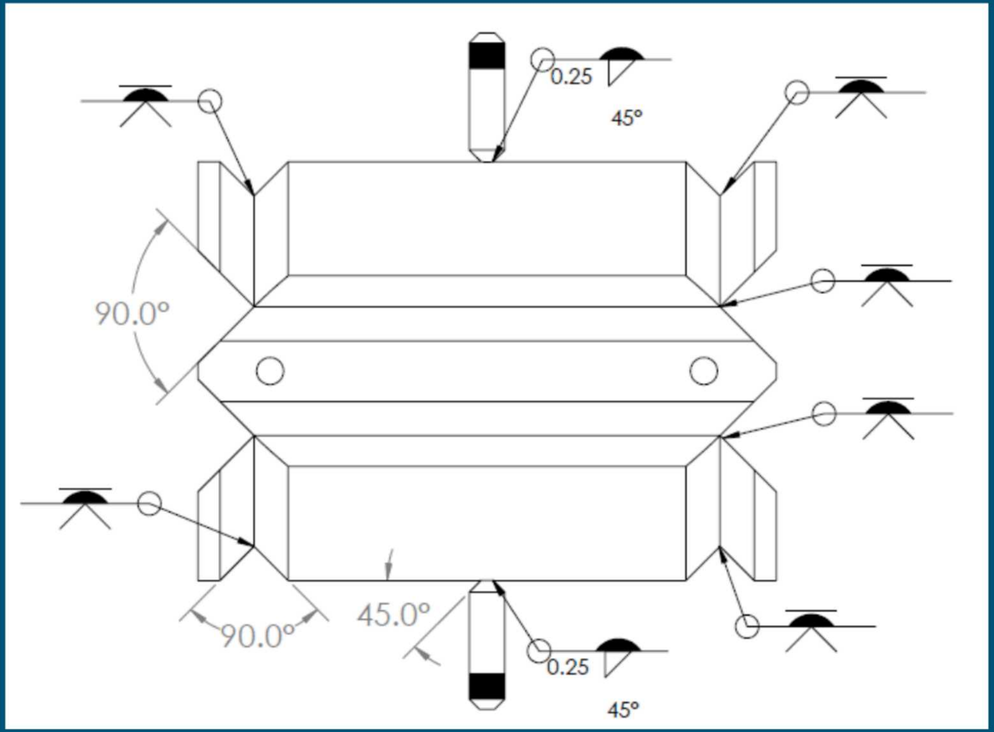
ASME BPVC

Weld requirements increased some thicknesses



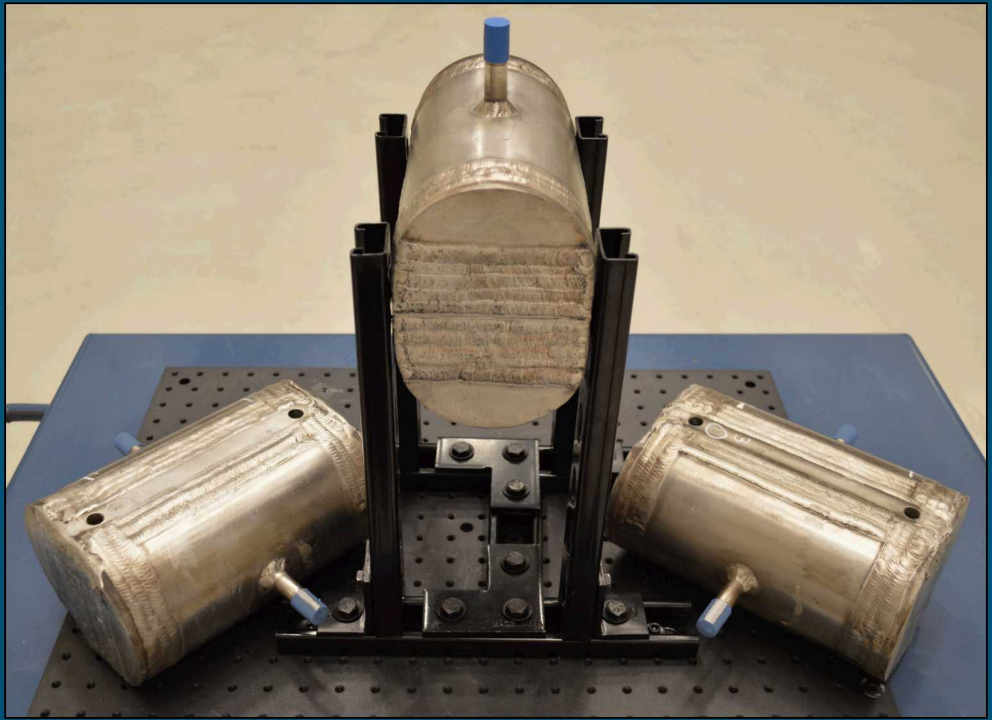
ASME BPVC

- The shell thickness t_{shell} was not increased by weld requirements
- The plate thickness t_{plate} was significantly increased from 0.15 inch to 2.0 inch
- The end cap thickness t_{cap} increased from 0.60 inch to 0.78 inch

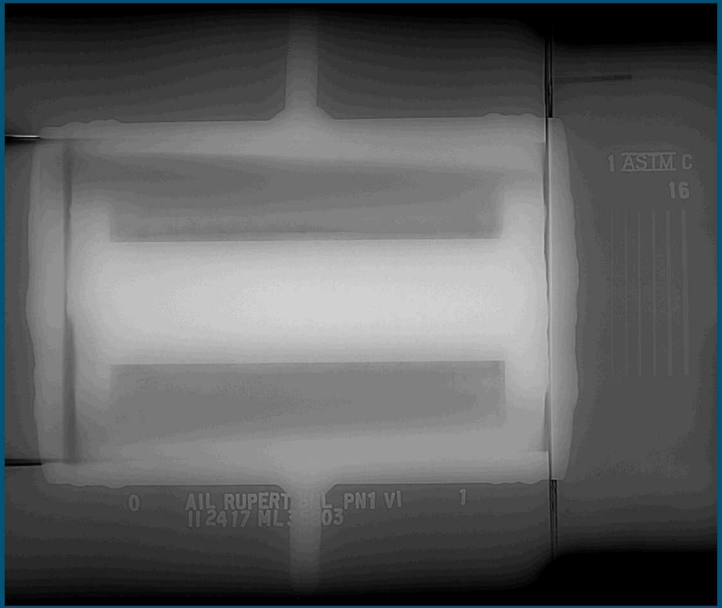


- Materials were selected from readily available options
- Shell made from 4” schedule 160 pipe with 0.531” thickness
- Plate made from 2” thick sheet
- End cap made from 0.875” sheet

Three prototypes were designed by the BPVC



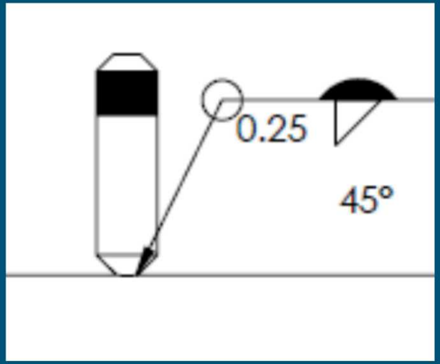
- Three Test Articles (TAs) allowed for experiment to be repeated in triplicate
- The material is 316/316L with allowable stress from ASME BPVC Section II, Part D of 20 ksi (138 MPa).
- The pressure rating is 3,900 psi (26.9 MPa) assuming no weld inspection (weld joint efficiency factor $E = 0.7$).

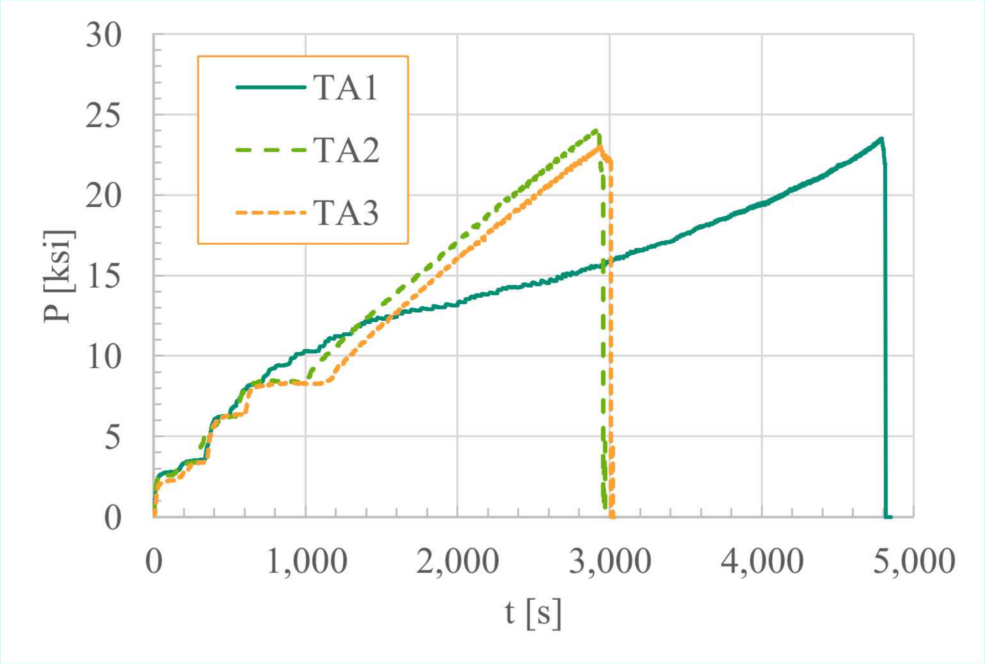


- Full X-ray inspection was performed to check for voids or defects
- Increased pressure rating is 5,550 psi (38.3 MPa) assuming $E = 1.0$

The burst pressure results were very consistent

- The pressure was increased slowly over about one hour to minimize transient effects
- The failure mode for all three TAs was a pin-hole formed in the nozzle-header weld.





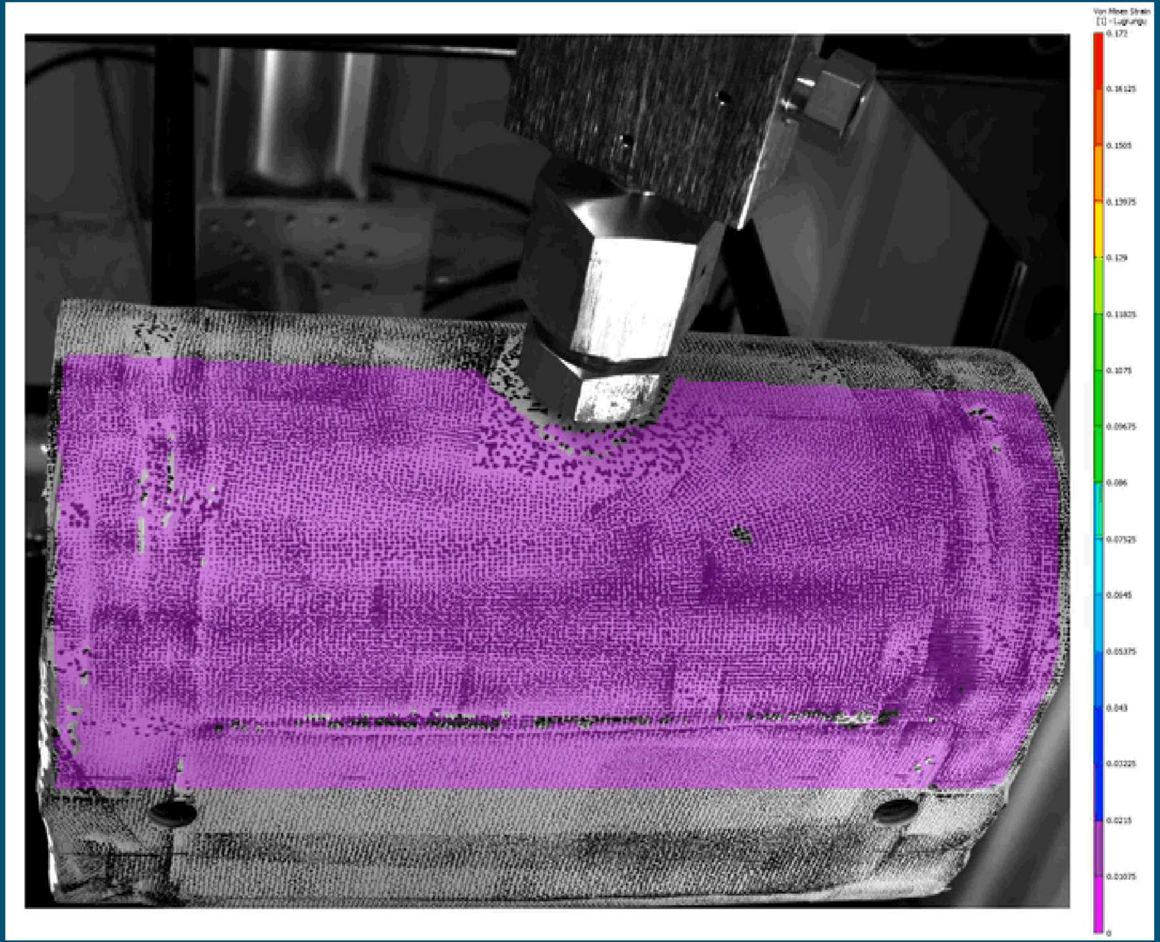
TA	Burst Pressure (ksi / MPa)	Uncertainty (ksi / MPa)	Burst Pressure Rating/ Design Rating
1	23.50 / 162.0	0.351 / 2.42	1.06
2	24.10 / 166.2	0.353 / 2.43	1.09
3	23.03 / 158.8	0.349 / 2.41	1.04
Ave	23.54 / 162.3	1.37 / 9.45	1.06

- Burst Pressure Rating

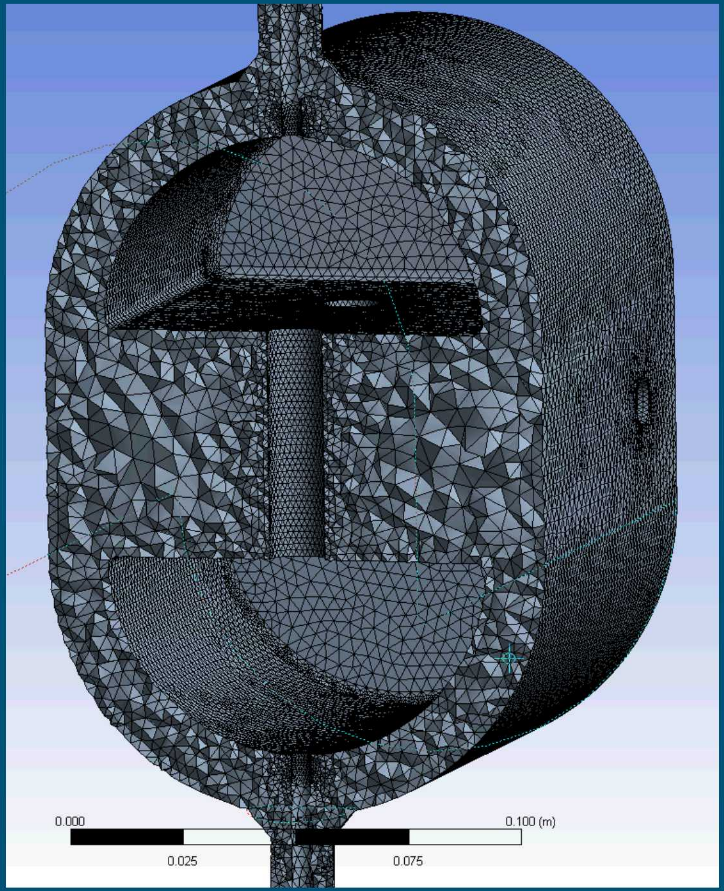
$$P = \frac{BE}{4}\frac{S_{DesignTemp}}{S_{TestTemp}}$$

Digital Image Correlation (DIC) was used to optically measure strain fields through the burst experiments

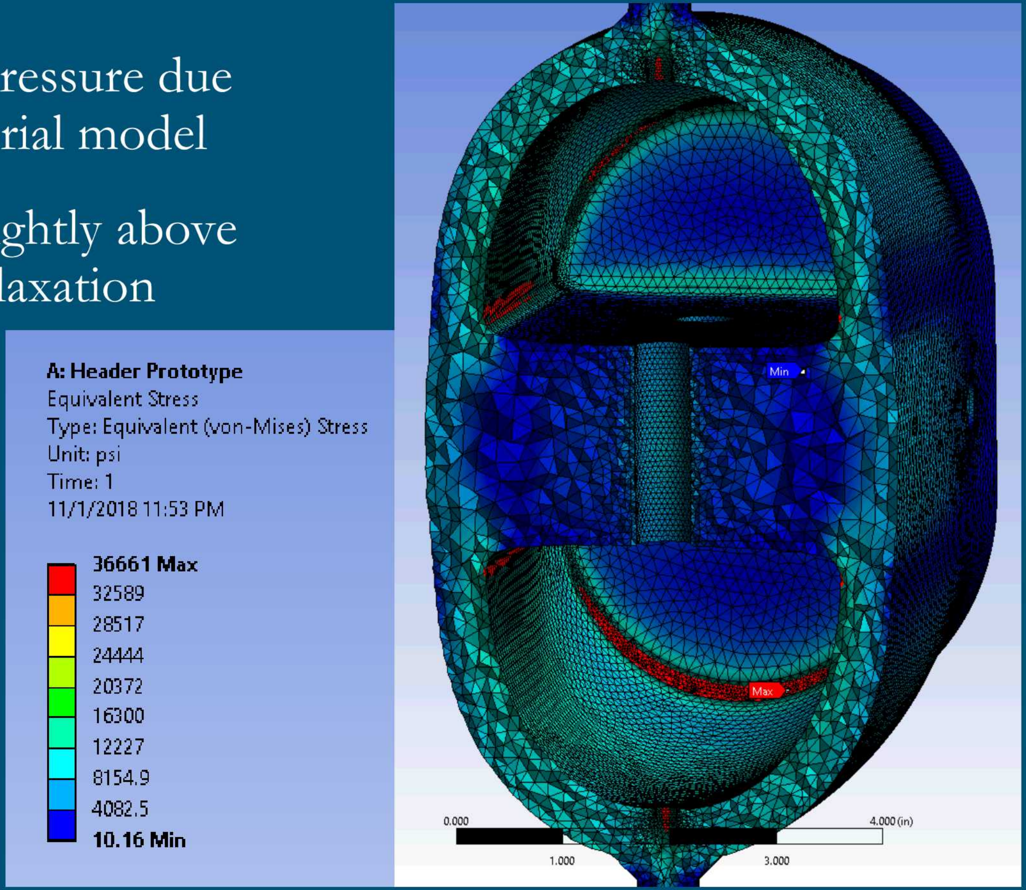
- Knowledgeable staff at Sandia Labs
- 8 cameras for 4 stereo DIC measurement systems
- VIC-3D software



Finite Element Analysis (FEA) was also performed

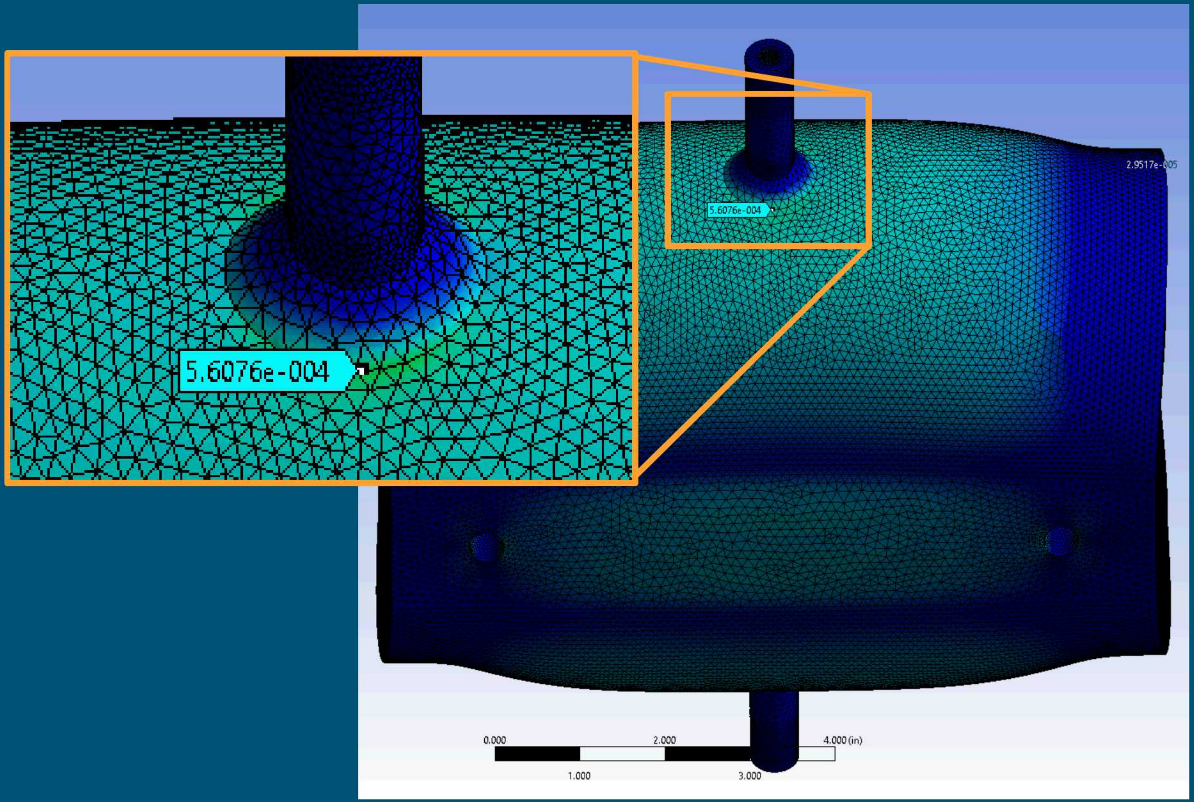


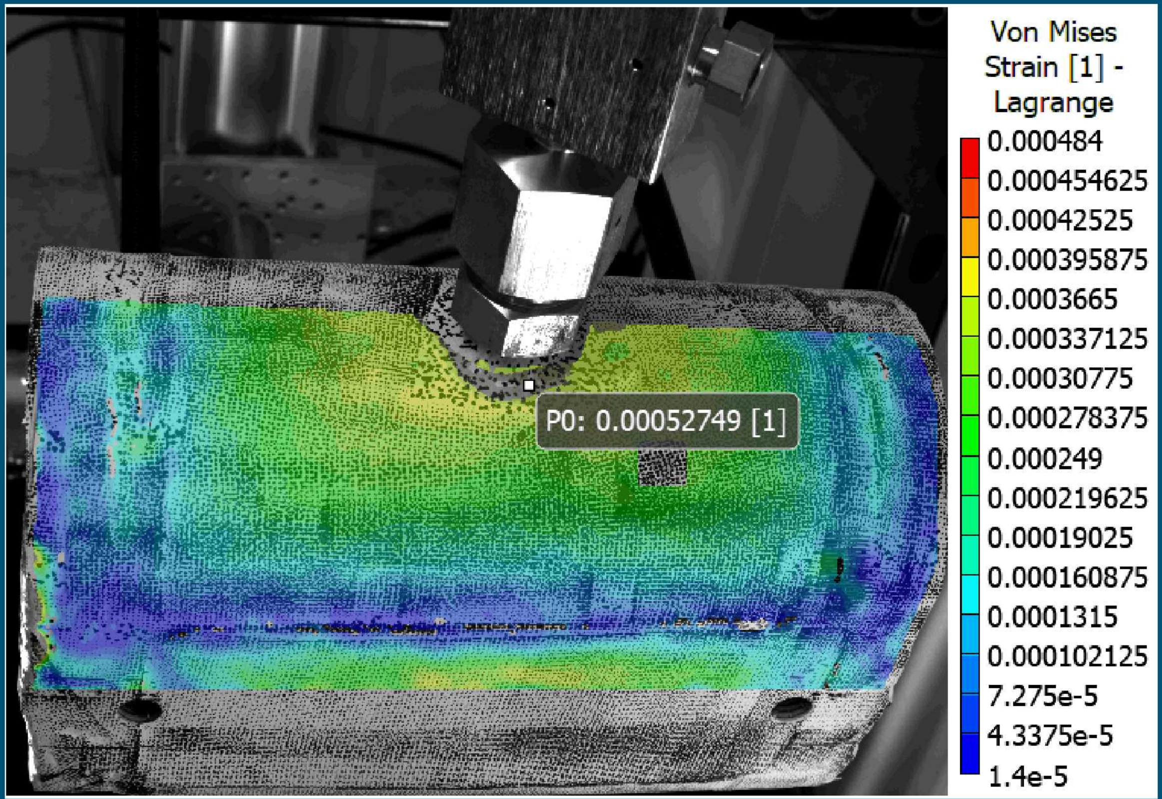
- ANSYS Mechanical 19.1
- 1.09M elements
- Local refinement based on curvature
- Pressures limited to rated pressure due to linear elastic default material model
- Local von-Mises stresses slightly above yield at 30 ksi (205 MPa), relaxation expected



The measured strain was very consistent with FEA predictions

- The maximum strain was predicted and measured at the nozzle-header weld at pressure rating
- This is also the failure area on all three TAs
- Predicted strain was 6.3% higher than measured





Conclusions and future work

- Conclusions
  - Pressure safety design equations in ASME BPVC for semi-circular headers validated
  - Measured burst pressure 6.1% higher than predicted by ASME BPVC
  - Measured strain 6.3% lower than predicted by FEA at design pressure
- Future Work
  - Extend FEA to plastic material model to compare with measured strain fields at higher pressure
- Related Work
  - Pressure fatigue on a Hydrogen Pre-cooler PCHE