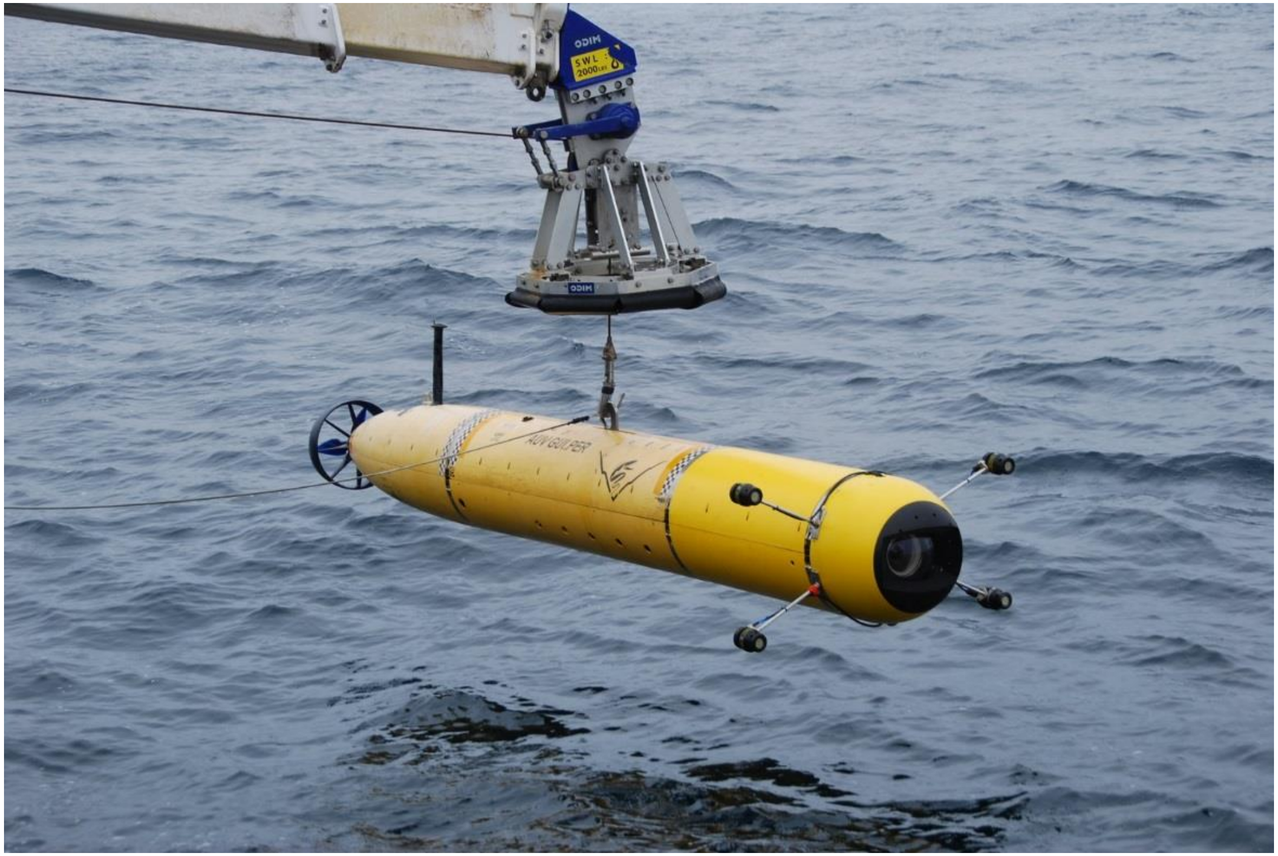
Proof testing of deep water camera port

Francois Cazenave

Monterey Bay Aquarium Research Institute

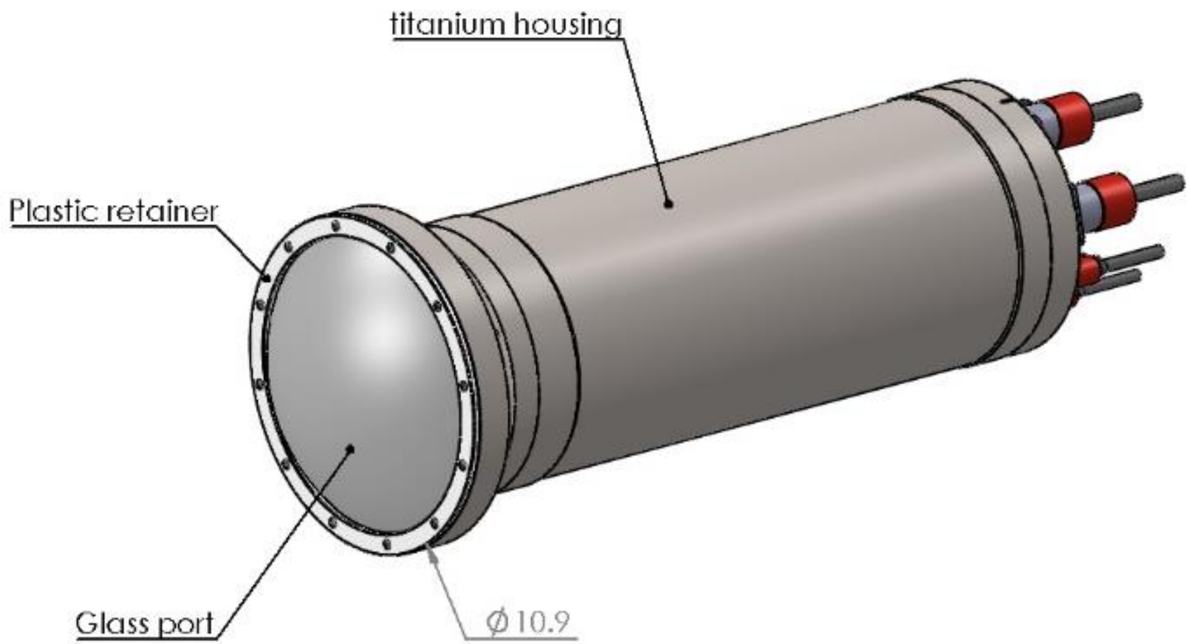
May 24, 2016

Application



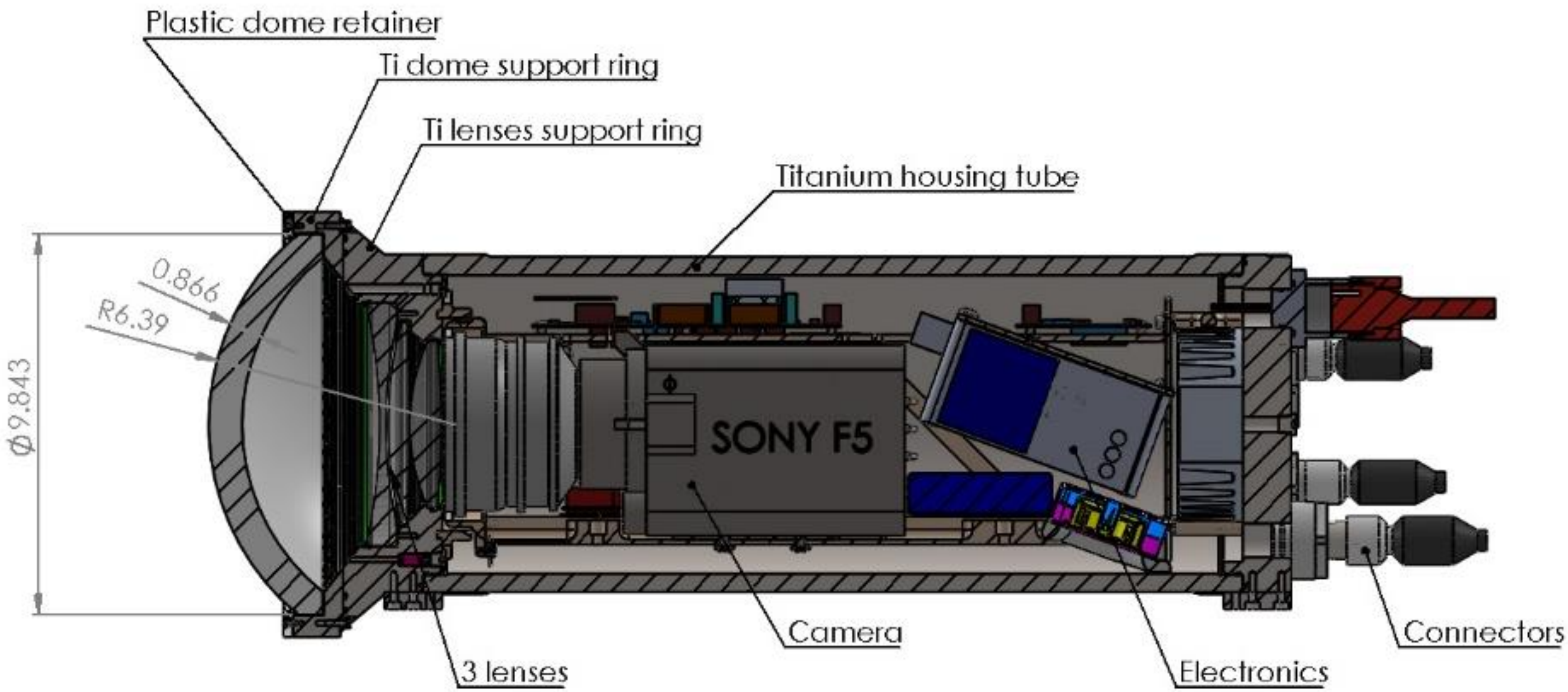
AUV (autonomous underwater vehicle) based imaging of midwater animals from 50m to 1500m depth.
Only 1 unit will be built.

Camera Housing design



Depth rating: 1500m
Temperature: Max 40 degC in the lab, minimum 4 degC at depth.
12 dives per year. 1 hour per dive at 1500m.
Minimum service life: 100 hours at depth.

Camera housing cross section



Dome mechanical design

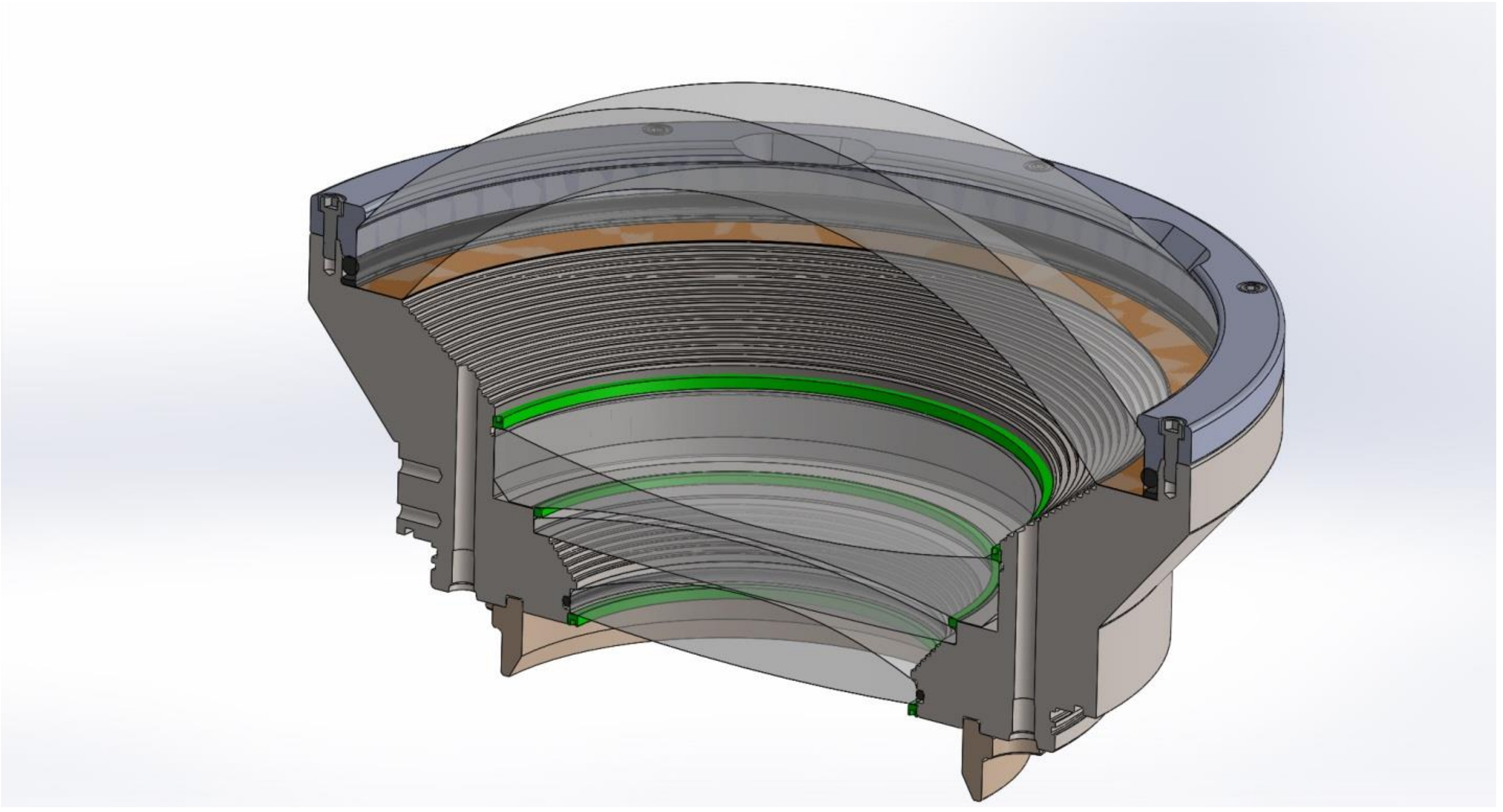
- FEA was performed using Solidworks simulation
- Materials:
- Grade 5 titanium
- H-K9L glass (Chinese BK7)
- Kapton (polyimide)
- Polyurethane resin

- Design was optimized for optical quality, not for mechanical properties.

- 5 domes were purchased ($1800 each). 2 were broken during pressure testing.

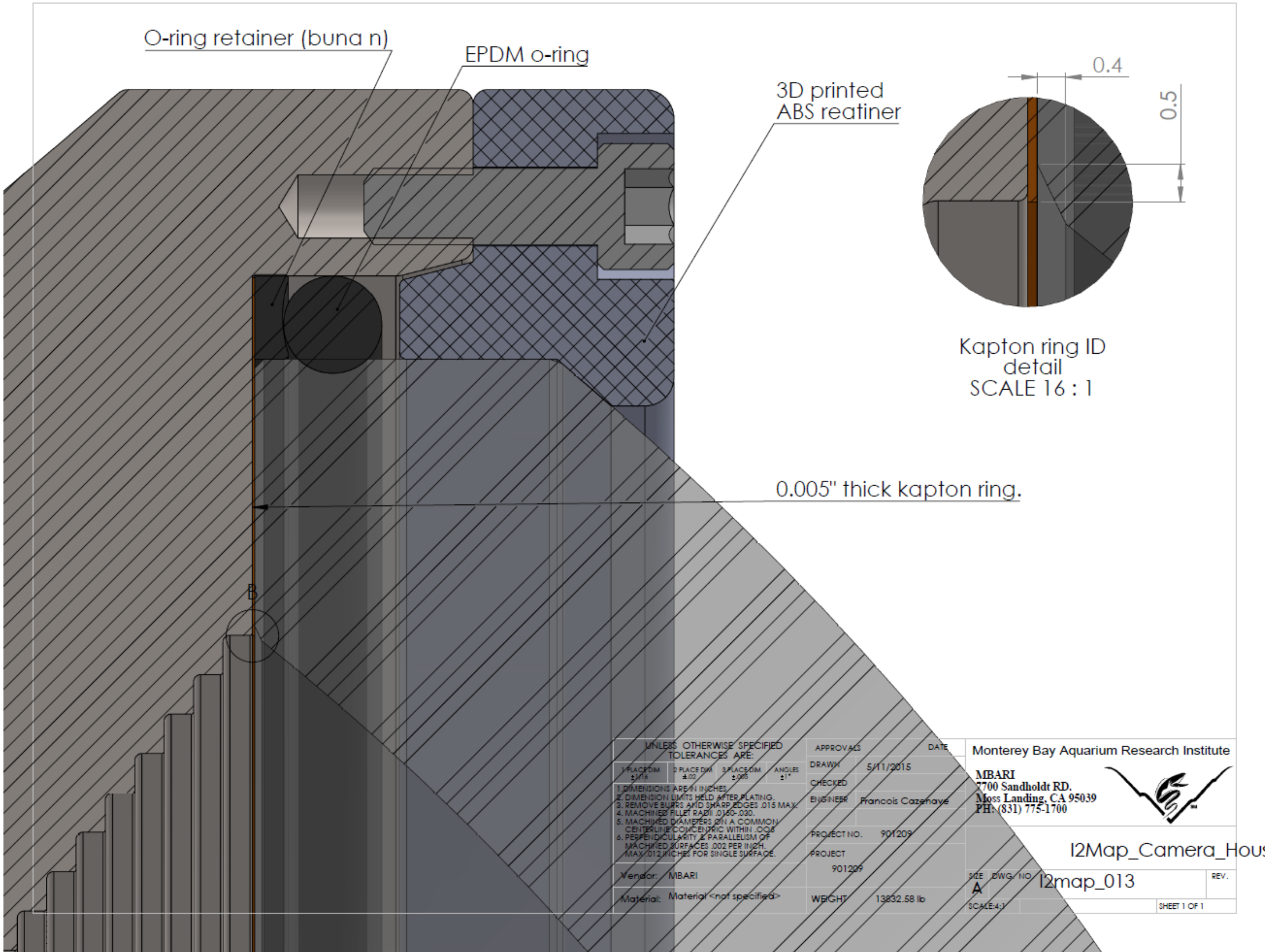
- The following slides shows previous designs that failed during pressure testing.

Dome Seat Design #1

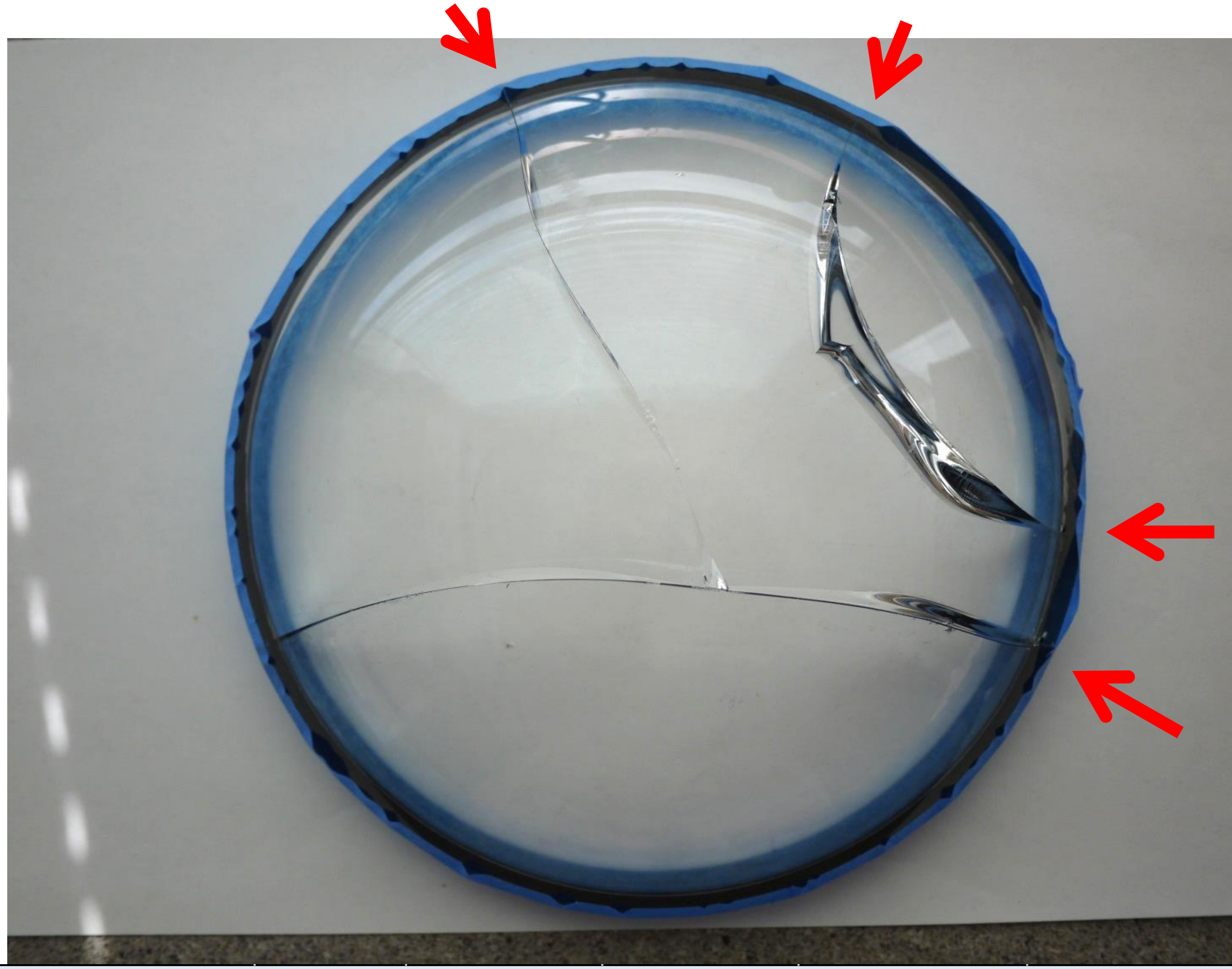


Dome material: H-K9L glass (similar to BK7)
Outer Diameter: 250mm
Thickness at apex: 22mm
Design depth: 1500 meters seawater (~2175 p.s.i)
Kapton support gasket thickness: 0.005”

Detail of Design #1 Dome Seat



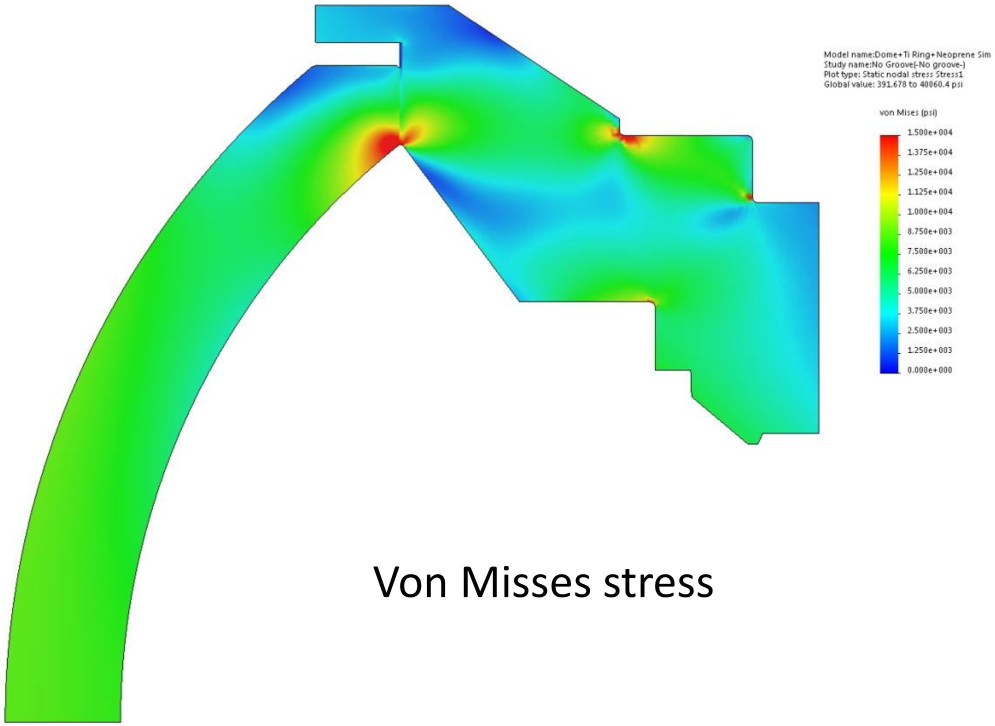
Design #1 Dome Failure



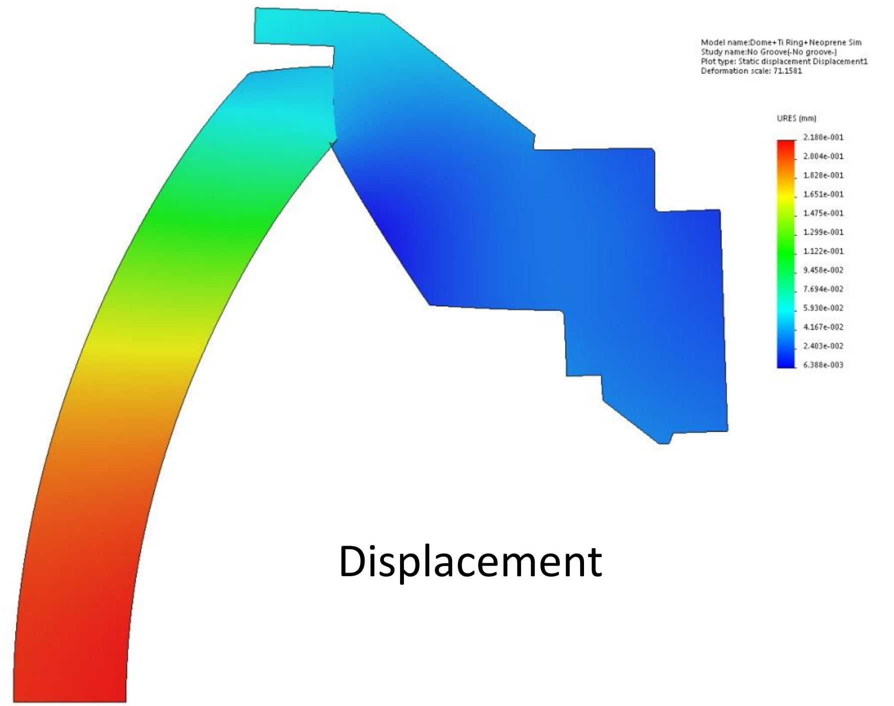
Initial failure points?

i2MAP Pressure Tests & Operations						
Pressure Test Description	**Date**	**Max. pressure (meters depth)**	**Max. pressure (PSI)**	**Number of Cycles**	**Failure pressure (meters depth)**	**Failure pressure (PSI)**
Test design #1 : Solid support ring, .005" Kapton. Problems identified; dome lacked specified inner corner break.	4/30/2015	1020	1500	1	pass	
	4/30/2015	1883	2750	9	pass	
	5/4/2015	1883	2750	Soak 72 + hours	1883	2750

FEA of first design



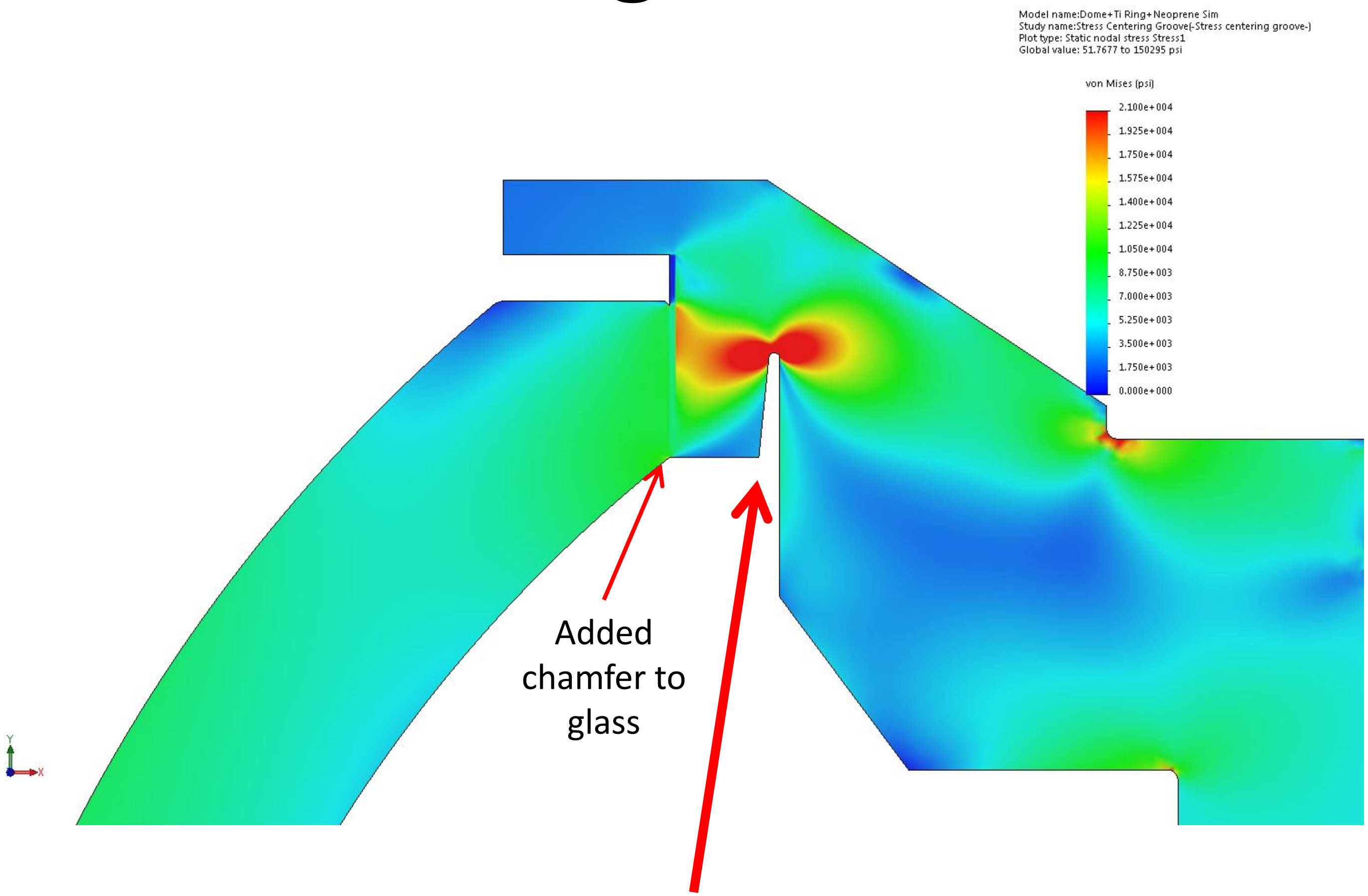
Von Misses stress



Displacement

- The coefficients of friction between glass and titanium, and between titanium and kapton are unknown. When using a medium (0.5) COF in SW Simulation, no sliding would occur between the glass and support ring so I assumed it was the case in reality.
- I was most concerned with compressive stress near the inner edge and I thought tensile stress was low.
- During the first test, there was no sliding and tensile stress was low (glass exhibits low stress failure). The failure occurred at 2750PSI after 10 cycles and several hours under pressure.

Dome Seat Design #2



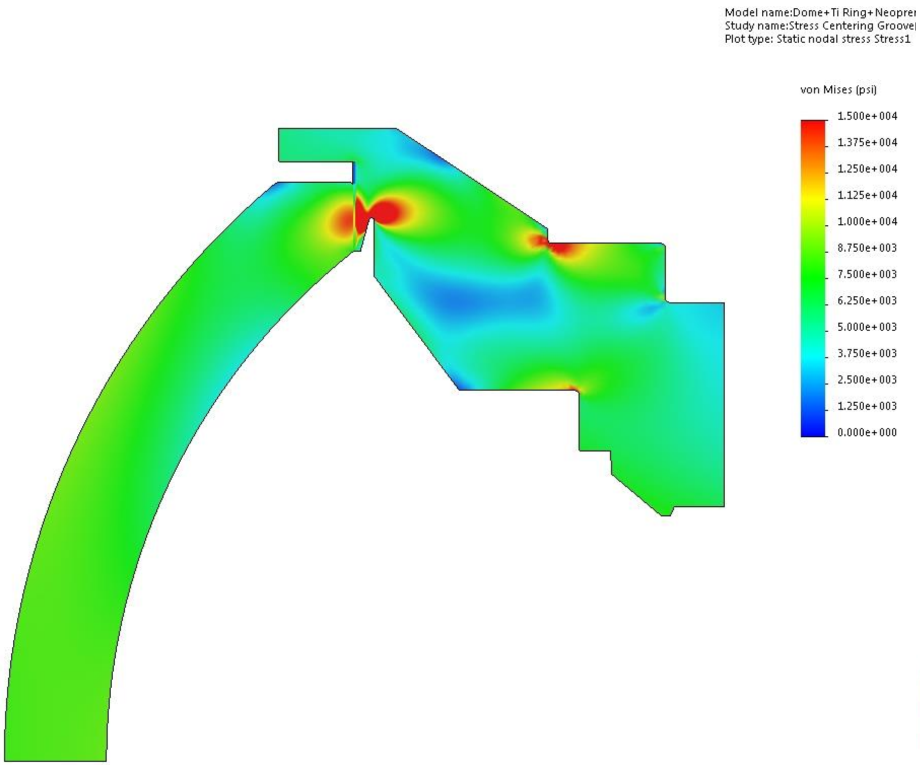
Same design as #1 except for; addition of groove below the seat to allow slight flexure of dome seat under hydrostatic pressure, Kapton ring thickness under dome seat was increased from 0.005 to 0.020 thickness, the seat areas was optically polished, and a small polished chamfer was added to the dome inside radius (outside radius already had a ground chamfer).

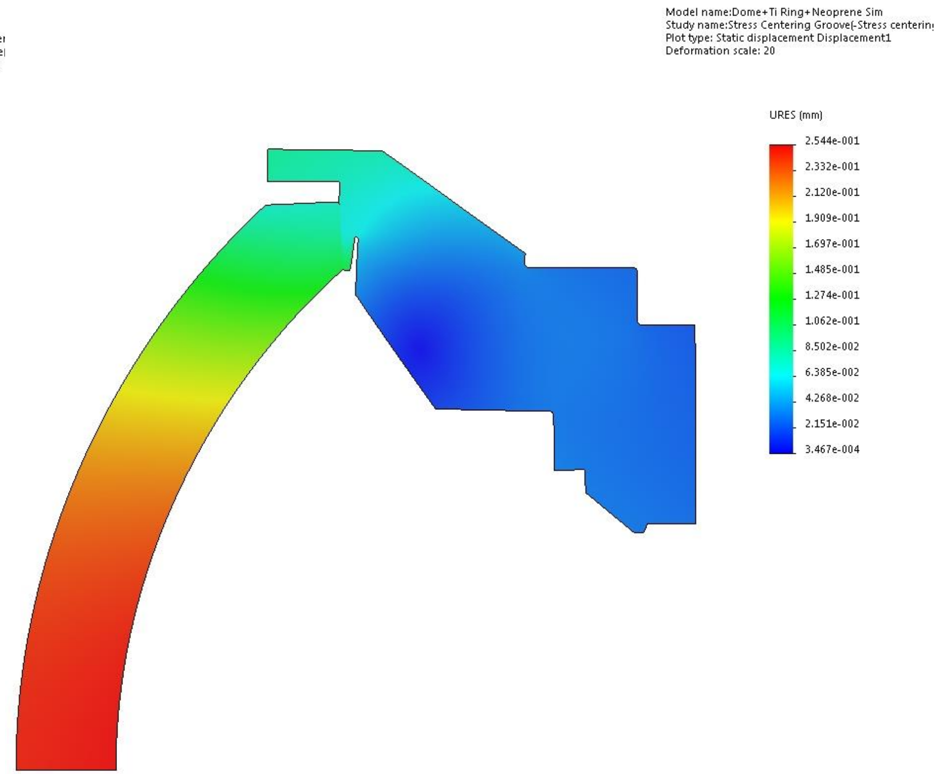
Design #2 Dome Failure

Initial failure point?

Pressure Test Description	Date	Max. pressure (meters depth)	Max. pressure (PSI)	Number of Cycles	Failure pressure (meters depth)	Failure pressure (PSI)
Test design # 2: Dome inner corner chamfered, no tampoprint ink, dome mating surface polished, .005" Kapton replaced with .020" Kapton gasket, Titanium support ring changed to flexure. Test date 4FEB16	2/4/2016	?	?	1	1064	1563

FEA of second design:

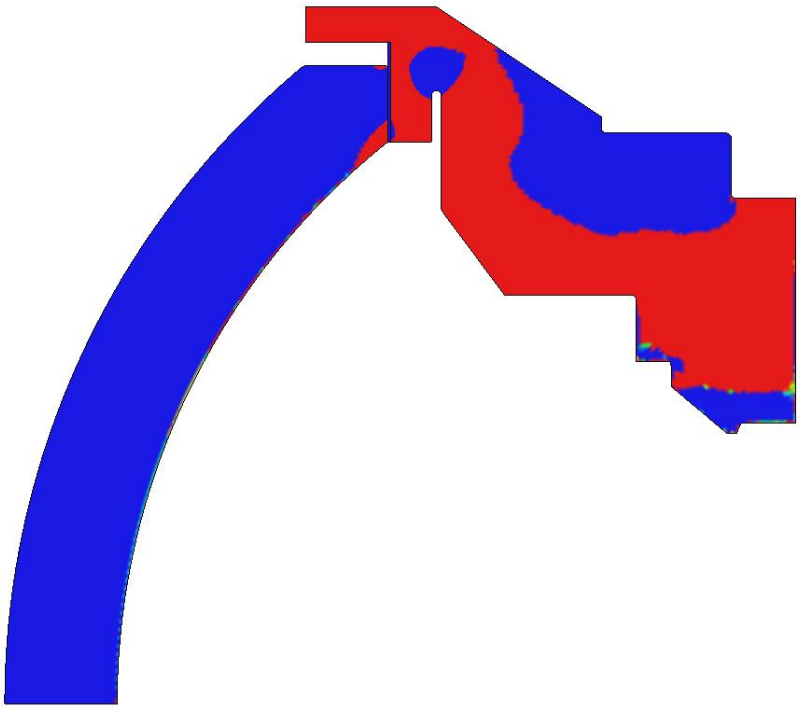


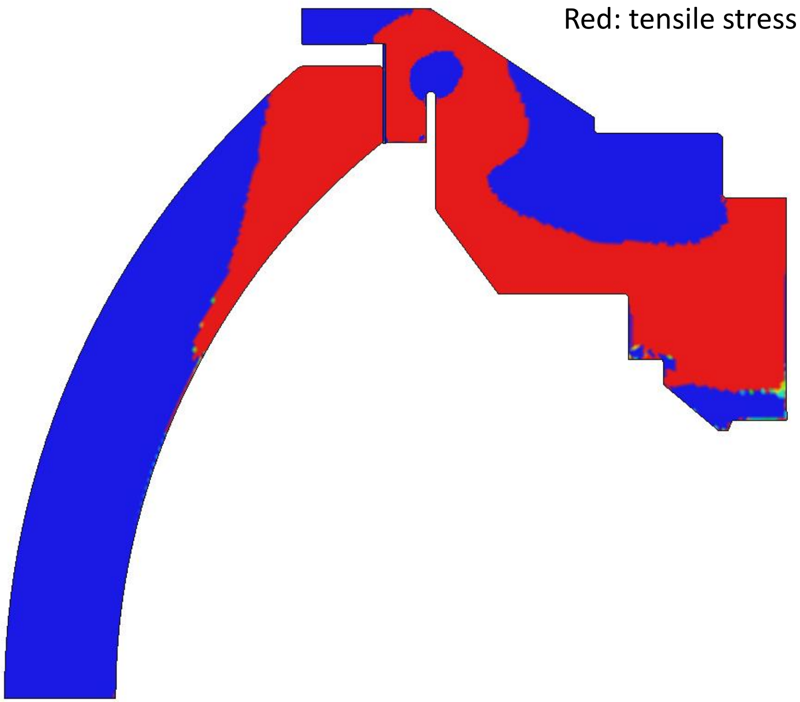


- we added a groove in the titanium to decrease compressive stress and to center it.
- We had the glass polished and increased the kapton thickness. The smoother surface and even stress across the seat decreased friction and resulted in sliding.
- Glass failed in tension due to hoop stresses on the periphery. Hoop tensile stress at failure was about 9000 PSI and radial displacement was 0.0045”, according to FEA (6000 PSI according to glass facture analysis). The failure occurred at 1570PSI, during the first cycle.

Comparison of sliding/no sliding cases

Blue: compressive stress
Red: tensile stress





Case with friction: no sliding
There is almost no tensile stress.
Mean 1st principal stress (at 1577 psi hydrostatic pressure):
on periphery: -188PSI
on seat: -355PSI

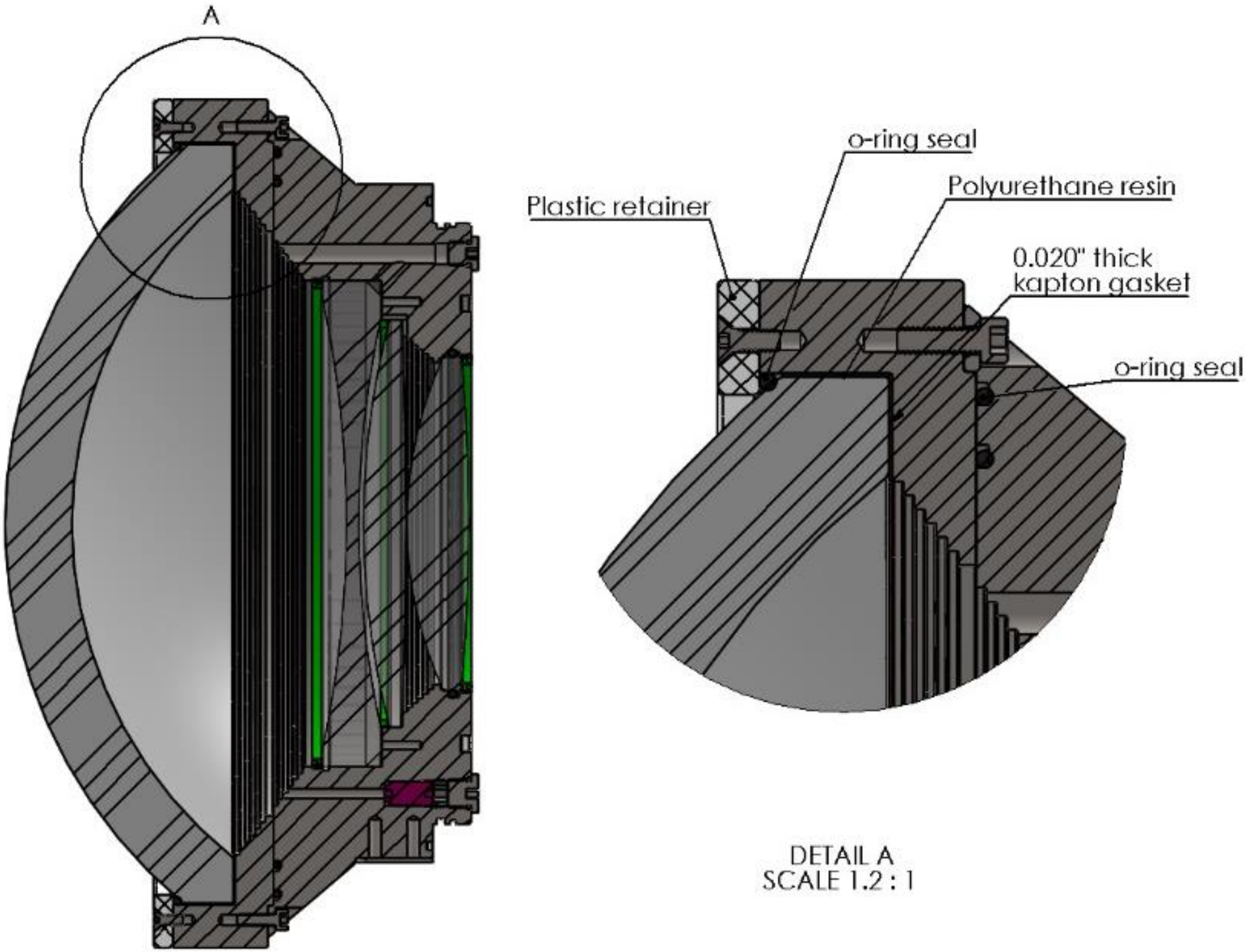
Case with no friction: sliding occurred.
High tensile stress on periphery
MFPS on periphery: 6513PSI
on seat: 8982PSI

Third and latest design:

The glass is bonded to a titanium ring using PU resin. The titanium prevents radial displacement of the dome and eliminates tensile hoop stress.

This design was successfully pressure tested to 1600m. Measurement of axial displacement of the glass matches FEA.

Camera front end cross section



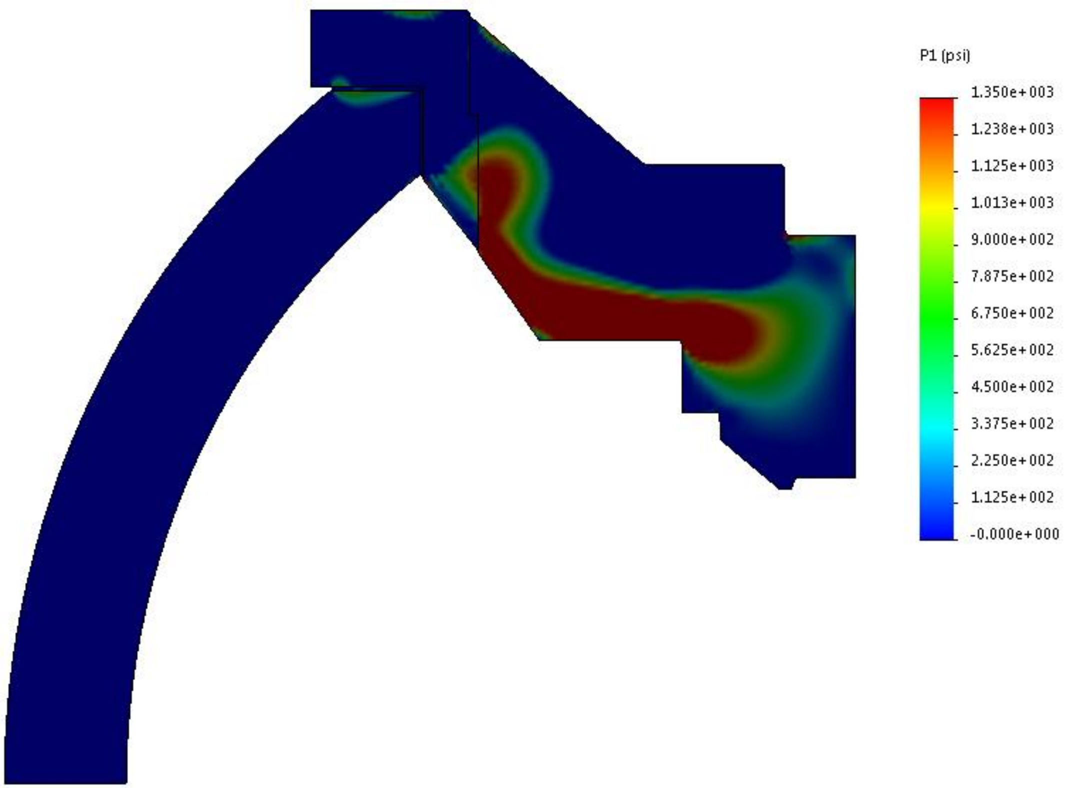
Glass bonded to titanium: FEA results

- Initial temperature: 30°C
- Pressure applied: 2200 PSI (1500m depth)
- Temperature applied: 4°C
- Thickness of resin buffer: 0.030”

- Max radial displacement of glass: -0.0005”
- Max axial displacement of glass: 0.007”

- Contact force of perimeter: 44000 lb
- Mean contact pressure on perimeter: 2328 PSI
- Contact force on seat: 170000 lb
- Mean contact pressure on seat: 10300 PSI

- Stress is compressive in most of the glass.
- Max principal stress in glass: Approximately 800 PSI.
- There is also a risk of higher tensile stress (2000PSI) developing at the upper edge of the outer surface. The titanium moves away from the resin and pulls on it. This could be remedied by applying mold release on the glass or titanium to prevent bonding.



Proof testing

- PhD Students at UC Berkeley calculated proof test values. Their work was based on the literature suggested by the facturographer (George Quinn). They suggested 3.3:1 test pressure ratio. Quinn looked at their report and thought it was in error.

- Proof testing pressure should be determined to insure 95% confidence that the dome will not fail during its life cycle (5 years of operation, 60 dives, 100 hours under pressure).