Testimony of
[redacted] P.E., DFE

Summary of preliminary findings regarding OceanGate and loss the Titan Submersible

Kemper Engineering Services
Baton Rouge, LA

Task

At the request of the US Coast Guard, provide additional expertise in the investigation of OceanGate with respect to the Titan Submersible and subsequent loss of life in the areas of:

- Engineering design and design management practices
- Organizational impact on engineering design, safety, and reliability
- Relevant codes & standards, with focus on ASME PVHO-1
- Verification, Validation, and Uncertainty Quantification (VVUQ)
- Acrylic windows
- Pressure vessels under external pressure
- Carbon fiber composites
- Computer modeling and simulations
- Failure analysis and root cause analysis

Timeline

Feb. 2016	KES and OceanGate present at Marine Technology Society (MTS) Manned Underwater Vehicles (MUV)/Underwater Intervention (New Orleans, La.) Also met socially through MTS social events.
Dec. 2017	[redacted] requested a pro-bono analysis of a non-standard acrylic window for CYCLOPS II.
Jan. 2018	Transmitted window report to OceanGate via Kohnen. Results indicated failure likely after multiple full-depth dives
Mar. 2018	[redacted] agrees to sign letter to OceanGate
June 2023	OceanGate's TITAN (formerly CYCLOPS II) lost at sea
Sep. 2023	KES submitted report to NTSB regarding TITAN window
Mar. 2024	KES agrees to volunteer to assist in USCG MBI

[redacted] P.E., DFE

Positions: Principal Engineer, KES (since company formation Dec. 2006)

Engineering Researcher, University of Louisiana at Lafayette (since Feb. 2024)

Lieutenant Colonel, U.S. Army Corps of Engineers (Retired June 2020)

Education: Xaverian High School, Brooklyn NY

Louisiana State University, Baton Rouge LA (BS Mechanical Engineering)

U.S. Army (Army Reserves Engineer officer career through Command and General Staff College)

Experience: Engineering since 1992, Self-employed as a consulting engineer since 1997 when not on military duty

Began working on saturation diving systems, diving support in 1998. Consulted for multiple diving, subsea, and marine companies, incl. complex hyperbaric systems, like the **Acergy DSV EAGLE** (dove Russia's KURSK) as well as submarine (PVHO) window manufacturers (Blanson Acrylic (UK), Hydrospace (USA)), consulted for Woods Hole Oceanographic Institute, OceanX, and US Navy Deep Submergence re. small subs; consulted on **Perry PC 1601**, potential work with Reynolds Marion's ***HyperSub***.

Joined ASME **Pressure Vessels for Human Occupancy** (PVHO) Codes & Stds committee in 2008
Chair, Viewports Subcommittee (2016); Chair, Design By Analysis for Glassy Polymers Task Group (2019)

Joined ASME **Verification, Validation, and Uncertainty Quantification** (VVUQ) Codes & Stds in 2022
Subcommittees: Solid Mechanics (VVUQ 10); AI and Machine Learning (VVUQ 70)

Bart Kemper, P.E., DFE

Licenses and Recognitions:

Professional Engineer (LA, TX, WA, others); Queensland, Australia

Board Certified Forensic Engineer (US through NAFE/CESB; International through IBFES)

Fellow, American Soc. of Mechanical Engineers (ASME); National Soc. of Professional Engineers (NSPE)

Editor-in-Chief, *Journal of the National Academy of Forensic Engineers* *(NAFE)*

Member: ASME, NSPE, MTS, NAFE, IEEE Oceanic Eng. Soc. (OES), Society of Naval Architects and Marine Engineers (SNAME), Am. Academy of Forensic Science (AAFS), Am. Assoc. for the Adv. of Science (AAAS)

Select Papers, Presentations:

[redacted] "Review of Life Limitations For Acrylic Windows In Pressure Vessels." ASME Int'l Mechanical Engineering Congress and Expo., Nov. 2023

[redacted] "Application of VVUQ Concepts to ASME Codes and Standards for Pressure Vessels," ASME Verification, Validation, and Uncertainty Quantification (VVUQ) Conference, May 2023

[redacted] "Developing 'Design By Analysis' methodology for windows for PVHOs." *ASCE-ASME Journal of Risk and Uncertainty in Engineering Systems* (Vol 6, Issue 3), Sept. 2020

[redacted] "Heat Retention and Structural Integrity of Glassy Polymer Windows." UI 2018

[redacted] "Jurisdictional Acceptance of Non-ASME Pressure Vessels for Human Occupancy". Joint ASME/USCG Workshop on Marine Technology & Standards, 2013

Total of 12 peer-reviewed papers, 20 conference papers or presentations on PVHOs, pressure vessels, subsea, or VVUQ

Bart Kemper does not represent ASME, MTS, NSPE, USCG, DHS, nor any organization other than KES. All opinions are his.

Kemper Engineering team

Lead Investigator [redacted] P.E., DFE, Principal Engineer KES, Researcher Univ. of La. Lafayette (PVHO)

KES Staff [redacted] Company President (PVHO)

Prof. Emeritus A.J. McPhate (LSU), P.E., Senior Consultant

[redacted] Staff Engineer

[redacted] mechanical engineering student intern (LSU)

[redacted] chemical engineering student intern (BRCC)

[redacted] IT support

Consultants [redacted] E., Mechanical/Biomedical Engineer (PVHO)

[redacted] Former saturation diver and diving operations manager (PVHO)

[redacted] P.E., PhD, Mechanical/Aerospace Engineer

[redacted] P.E., DFE, Mech/Aerospace Engineer, formerly of NASA, incl. COLUMBIA investigation

[redacted] P.E., DFE, Electrical Engineer

[redacted] Aerospace Engineer/Simulation Expert

PVHO = Member of ASME PVHO or subcommittees; DFE = Diplomate of Forensic Engineering (US Board Certification)

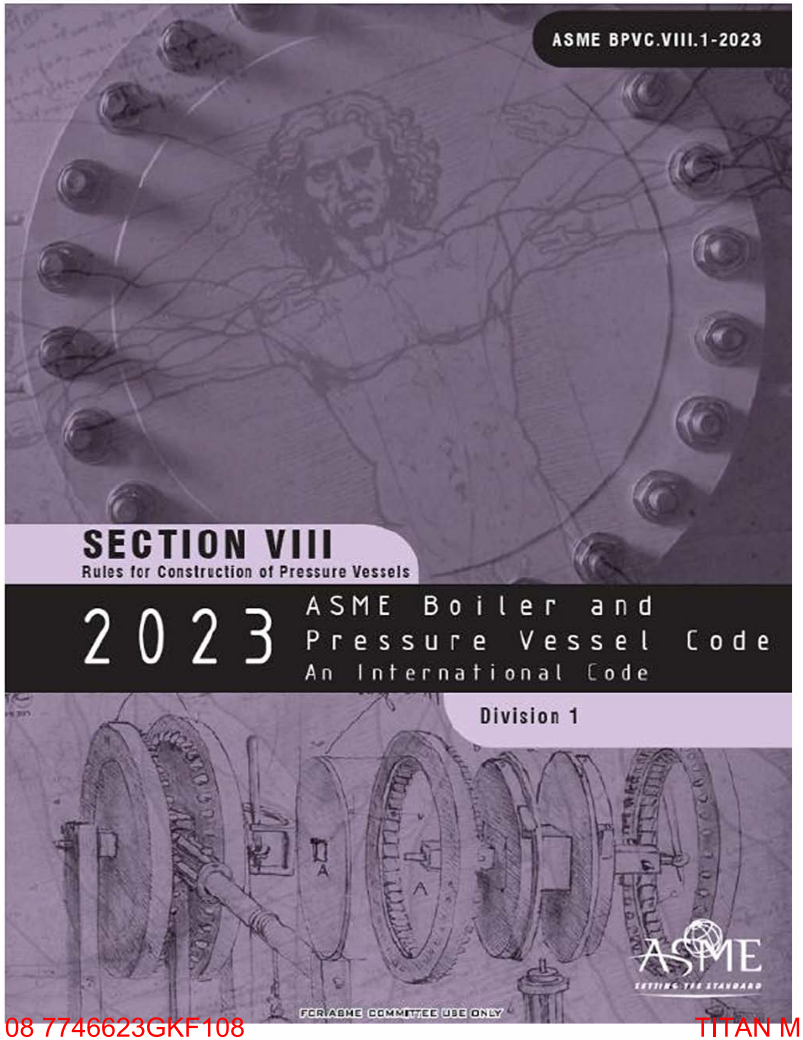
Kemper Engineering team

- **Brief by exception** (focus on implosion, consideration of previous briefings)
- Role of engineering codes & standards and the jurisdictional authorities
- Innovation and the need to use VVUQ methods
- Initial results regarding:
  - Window design
  - Pressure hull (carbon fiber composite, titanium heads, glued joint)
  - Testing & Simulations (as part of VVUQ)
  - Operations and safety procedures
- **Observations**: Failure to have an integrated life cycle systems engineering approach; Failure to have a Responsible-In-Charge professional

Engineering codes & standards (C&S)

- **“Written in blood”** – C&S often in direct response to safety need
- Intended to capture **best practices** as specific calculations, data sets, and procedures so others don’t have to “reinvent the wheel”.
- Developed by **volunteers** with a cross-section of interests, vetted for knowledge
- Vetted by the **American National Standards Institute** (ANSI)
- New technology is adopted based on **data, studies, and practice**
- **Cannot enforce**; is published and jurisdictions decide to use it
- Is a **system of codes**, where other requirements are cited by codes, e.g. ASME pressure vessel code cites piping, bolting and testing codes as well as cites ASCE-7 for wind and other environmental loads, etc. ASME PVHO-1 relies on other codes for detailed structural design other than windows.

ASME Pressure Vessel Codes



ASME BPVC.VIII.1-2023

SECTION VIII
Rules for Construction of Pressure Vessels

2023 ASME Boiler and Pressure Vessel Code
An International Code

Division 1

ASME
SETTING THE STANDARD

FOR ASME COMMITTEE USE ONLY

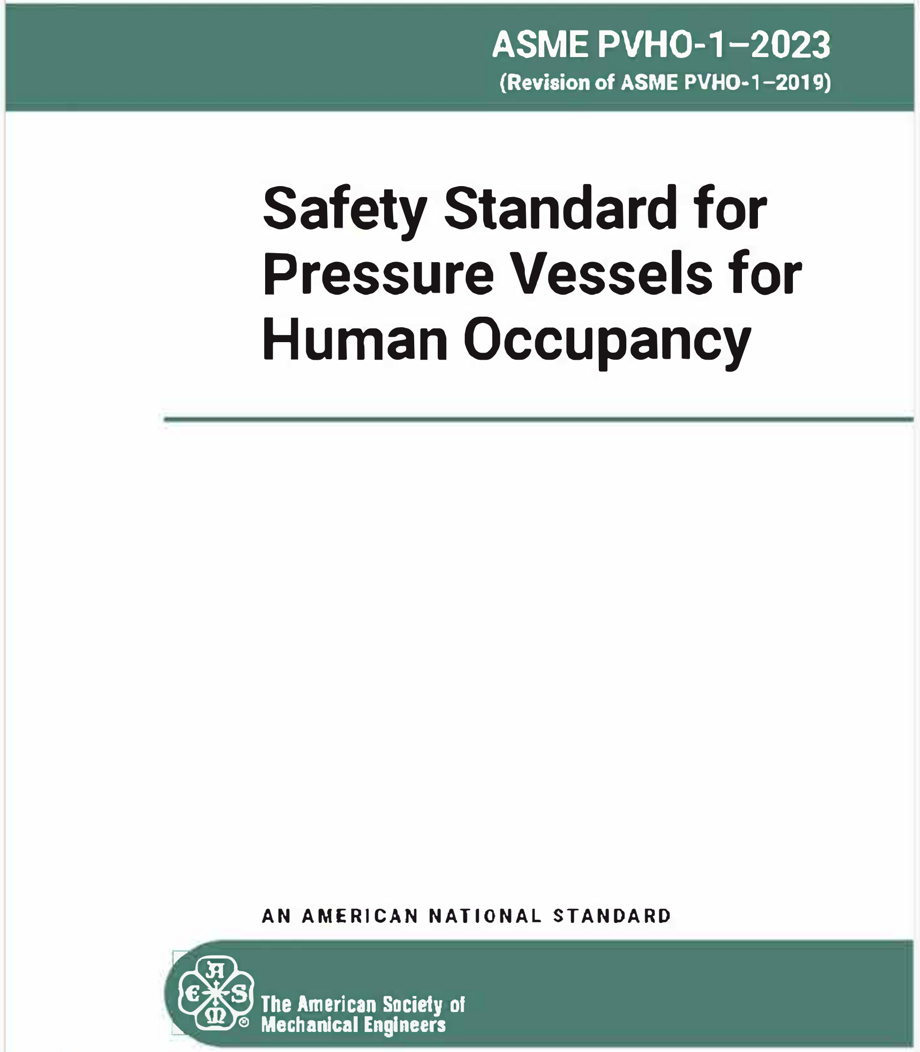
The ASME Boiler and Pressure vessel code is design **system with 13 sections in 34 binders** that includes material properties indexed by alloy and temperature as well as design rules for boilers, nuclear applications, heat exchangers, and pressure vessels. It includes rules for “design by analysis” (Section VIII, Div. 2, Part 5).

There are separate groups of codes for piping (B31) and other related items.

API 579/ASME FFS1 provides for “fitness for service” to determine **reliable useful life** of in-service items.

ASME PVHO-1 is a design and safety code. ASME PVHO-2 is post-construction. It was **originally focused on windows** and has expanded to look at **PVHOs as a system**, with emphasis on life support and safety. It does not address the pressure vessel design in depth but instead defers to other codes (ASME Section VIII, Class societies).

ASME PVHO-1 specific documents



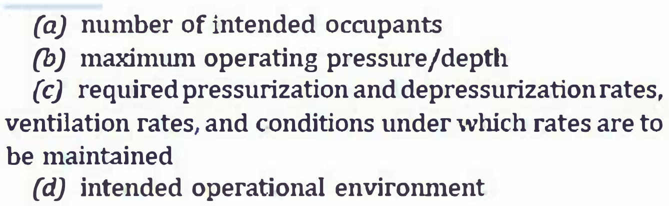
ASME PVHO-1–2023
(Revision of ASME PVHO-1–2019)

Safety Standard for Pressure Vessels for Human Occupancy

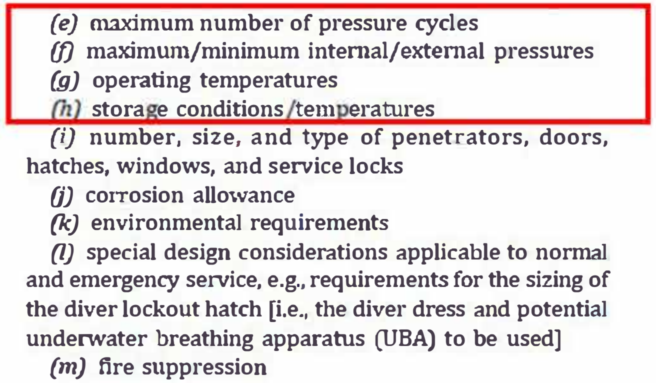
AN AMERICAN NATIONAL STANDARD

The American Society of Mechanical Engineers

User's Design Specification
(Section 7: Submarines)



(a) number of intended occupants
(b) maximum operating pressure/depth
(c) required pressurization and depressurization rates, ventilation rates, and conditions under which rates are to be maintained
(d) intended operational environment



(e) maximum number of pressure cycles
(f) maximum/minimum internal/external pressures
(g) operating temperatures
(h) storage conditions/temperatures
(i) number, size, and type of penetrators, doors, hatches, windows, and service locks
(j) corrosion allowance
(k) environmental requirements
(l) special design considerations applicable to normal and emergency service, e.g., requirements for the sizing of the diver lockout hatch [i.e., the diver dress and potential underwater breathing apparatus (UBA) to be used]
(m) fire suppression

Manufacturer's Data Report
Form GR-1. Shows the design meets the User's Design Specification. Requires Professional Engineer to review and sign

Risk Analysis
Retained document to be transmitted with item

Retained documents
List of items that are inspectable by jurisdiction, also to be retained and transmitted with the PVHO, including window certifications (VP1 through VP4). These documents would be available for review if OceanGate followed the PVHO-1

Acrylics windows?

During TRIESTE dives, glass and synthetic sapphire allowed leaks as pressure increased. Ceramic cracks often often unstable, give little reaction time.

Acrylics are a plastic and molds to the conical seat. Cracks tend to be more stable, allows reaction time.

Acrylic is weaker than metals and more temperature sensitive. Dr. Stachiw made conservative assumptions so windows do not fail before other components. **No design failures to date.**

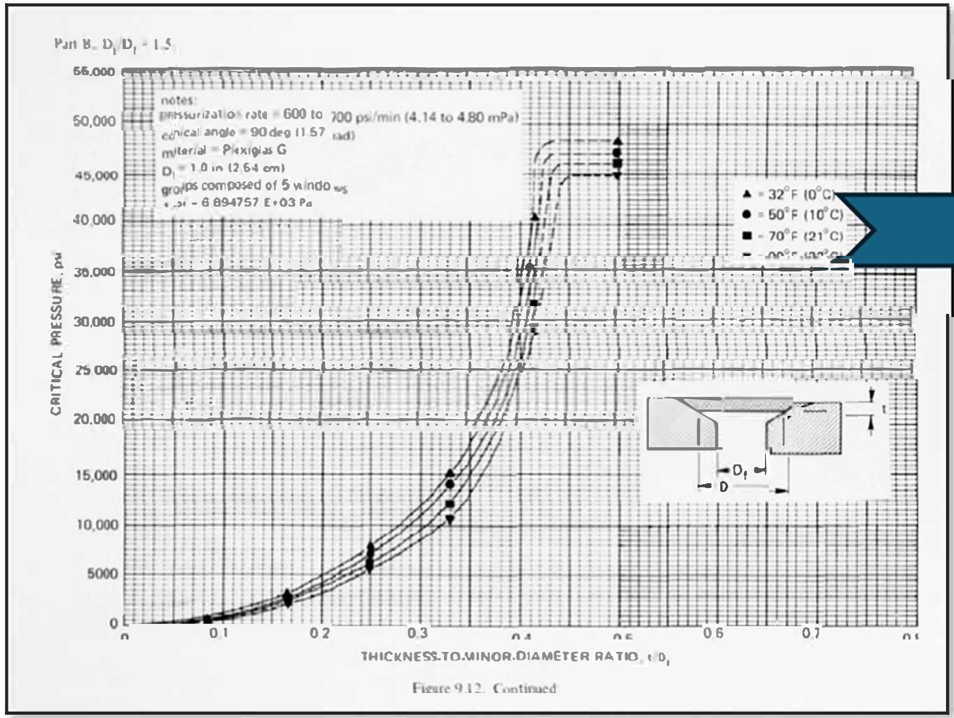
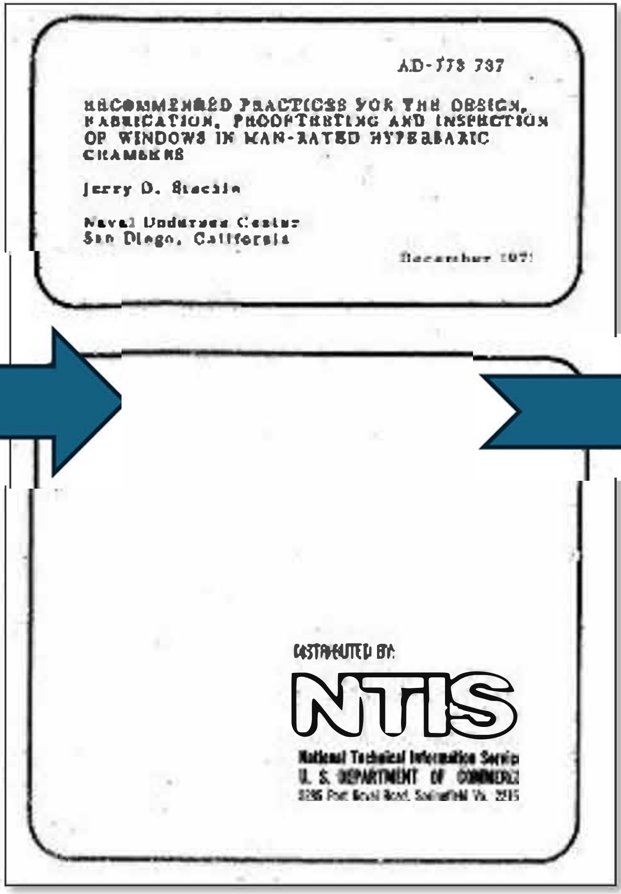


Figure 9.12. Continued



AD-773 737

RECOMMENDED PRACTICES FOR THE DESIGN, FABRICATION, PROOFTESTING AND INSPECTION OF WINDOWS IN MAN-RATED HYPERBARIC CHAMBERS

Jerry D. Stachiw

Naval Undersea Center
San Diego, California

December 197[illegible]

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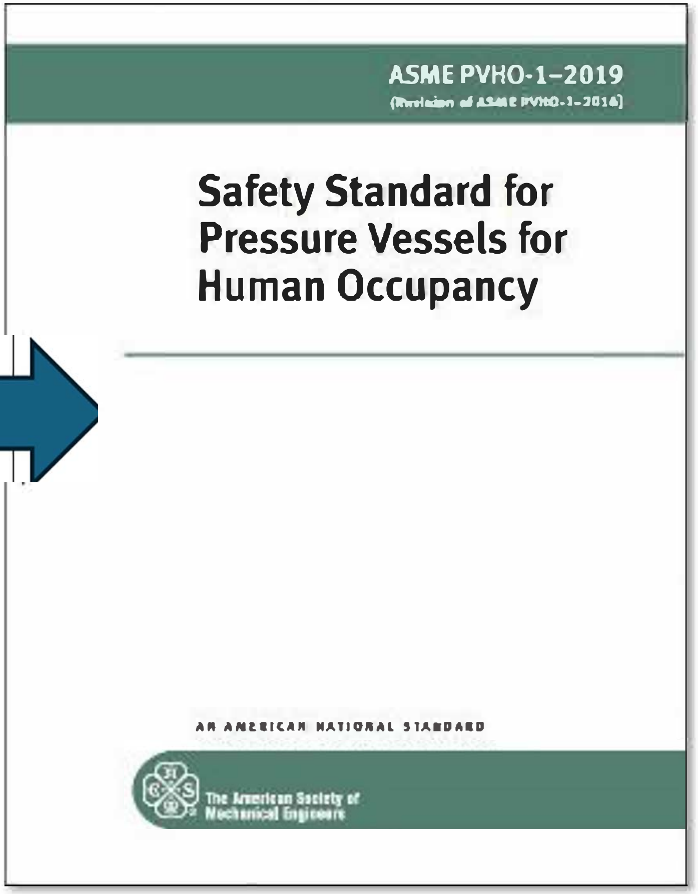


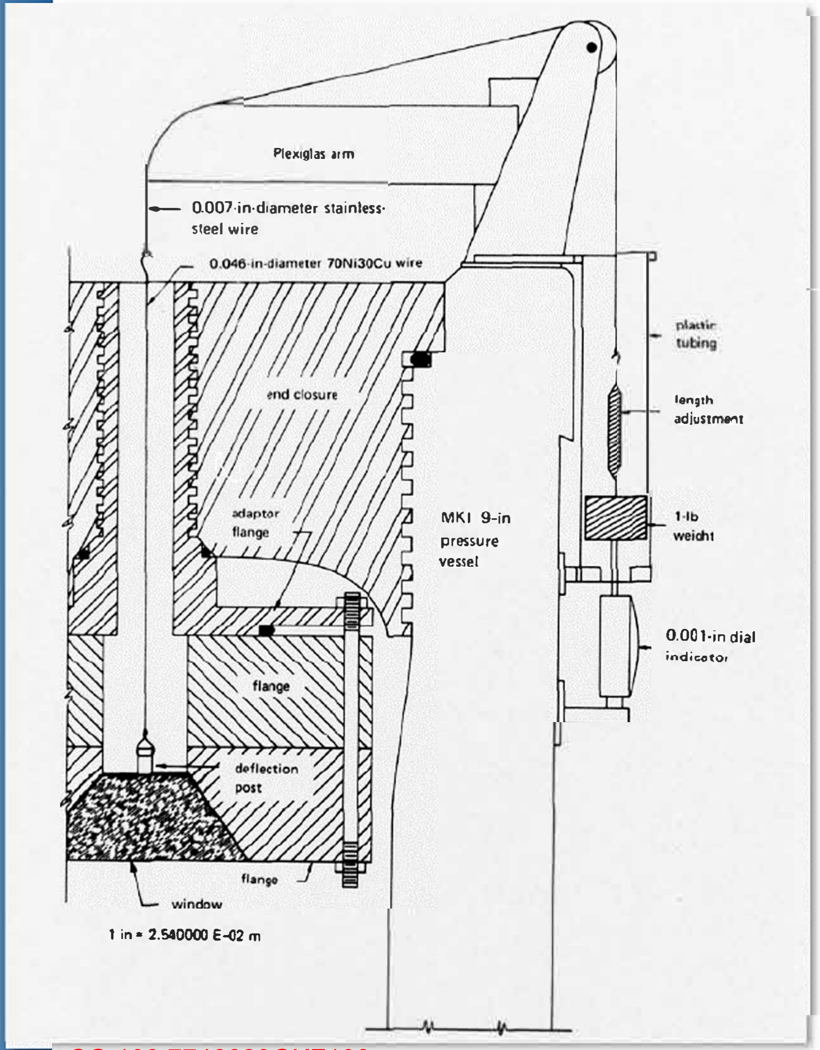
ASME PVHO-1–2019
(Revision of ASME PVHO-1–2016)

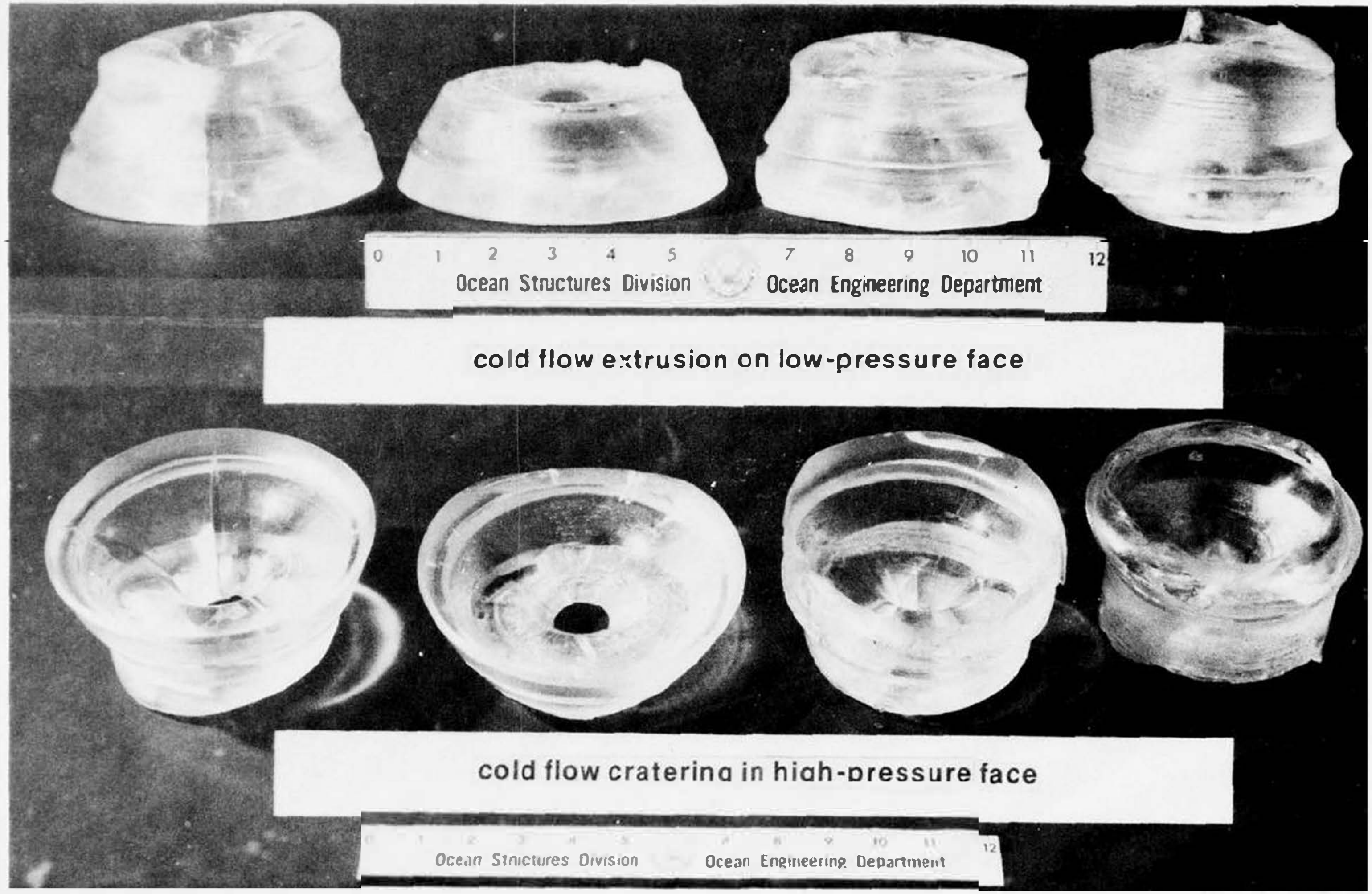
Safety Standard for Pressure Vessels for Human Occupancy

AN AMERICAN NATIONAL STANDARD

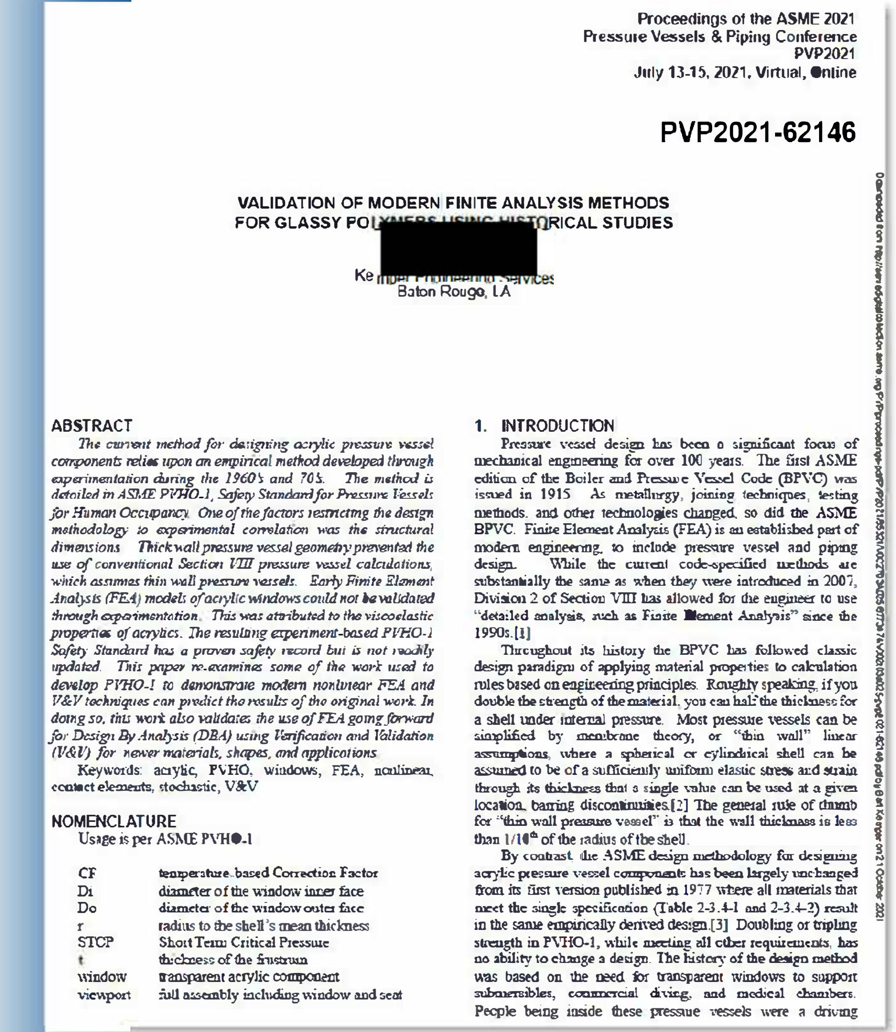
The American Society of Mechanical Engineers

Acrylics windows?





Acrylics windows?



Proceedings of the ASME 2021
Pressure Vessels & Piping Conference
PVP2021
July 13-15, 2021, Virtual, Online

PVP2021-62146

VALIDATION OF MODERN FINITE ANALYSIS METHODS
FOR GLASSY POL[redacted]ORICAL STUDIES

Ke
Baton Rouge, LA

ABSTRACT

The current method for designing acrylic pressure vessel components relies upon an empirical method developed through experimentation during the 1960's and 70's. The method is detailed in ASME PVHO-1, Safety Standard for Pressure Vessels for Human Occupancy. One of the factors restricting the design methodology to experimental correlation was the structural dimensions. Thick wall pressure vessel geometry prevented the use of conventional Section VIII pressure vessel calculations, which assumes thin wall pressure vessels. Early Finite Element Analysis (FEA) models of acrylic windows could not be validated through experimentation. This was atributed to the viscoelastic properties of acrylics. The resulting experiment-based PVHO-1 Safety Standard has a proven safety record but is not readily updated. This paper re-examines some of the work used to develop PVHO-1 to demonstrate modern nonlinear FEA and V&V techniques can predict the results of the original work. In doing so, this work also validates the use of FEA going forward for Design By Analysis (DBA) using Verification and Validation (V&V) for newer materials, shapes, and applications.

Keywords: acrylic, PVHO, windows, FEA, nonlinear, contact elements, stochastic, V&V

NOMENCLATURE

Usage is per ASME PVHO-1

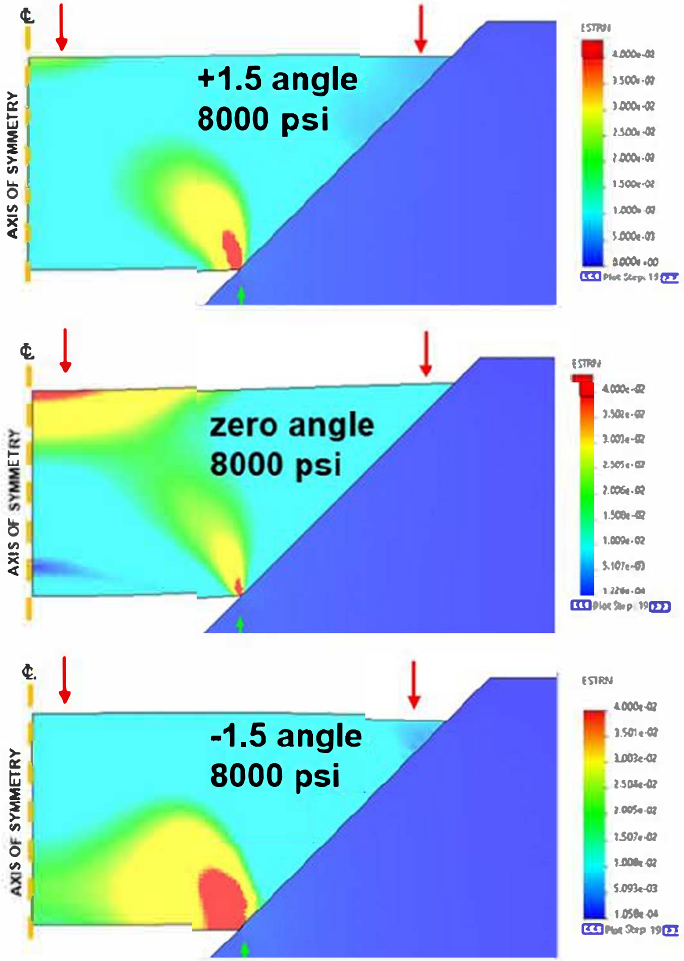
CF	temperature-based Correction Factor
Di	diameter of the window inner face
Do	diameter of the window outer face
r	radius to the shell's mean thickness
STCP	Short Term Critical Pressure
t	thickness of the frustrum
window	transparent acrylic component
viewport	full assembly including window and seat

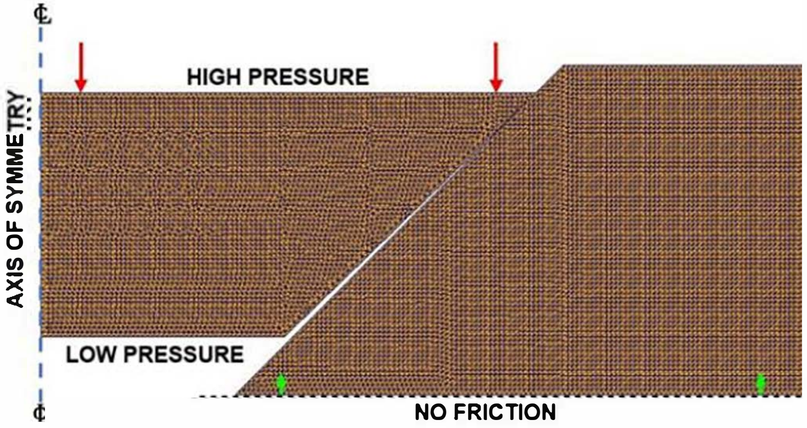
1. INTRODUCTION

Pressure vessel design has been a significant focus of mechanical engineering for over 100 years. The first ASME edition of the Boiler and Pressure Vessel Code (BPVC) was issued in 1915. As metallurgy, joining techniques, testing methods, and other technologies changed, so did the ASME BPVC. Finite Element Analysis (FEA) is an established part of modern engineering, to include pressure vessel and piping design. While the current code-specified methods are substantially the same as when they were introduced in 2007, Division 2 of Section VIII has allowed for the engineer to use "detailed analysis, such as Finite Element Analysis" since the 1990s.[1]

Throughout its history the BPVC has followed classic design paradigm of applying material properties to calculation rules based on engineering principles. Roughly speaking, if you double the strength of the material, you can half the thickness for a shell under internal pressure. Most pressure vessels can be simplified by membrane theory, or "thin wall" linear assumptions, where a spherical or cylindrical shell can be assumed to be of a sufficiently uniform elastic stress and strain through its thickness that a single value can be used at a given location, barring discontinuities.[2] The general rule of thumb for "thin wall pressure vessel" is that the wall thickness is less than 1/10th of the radius of the shell.

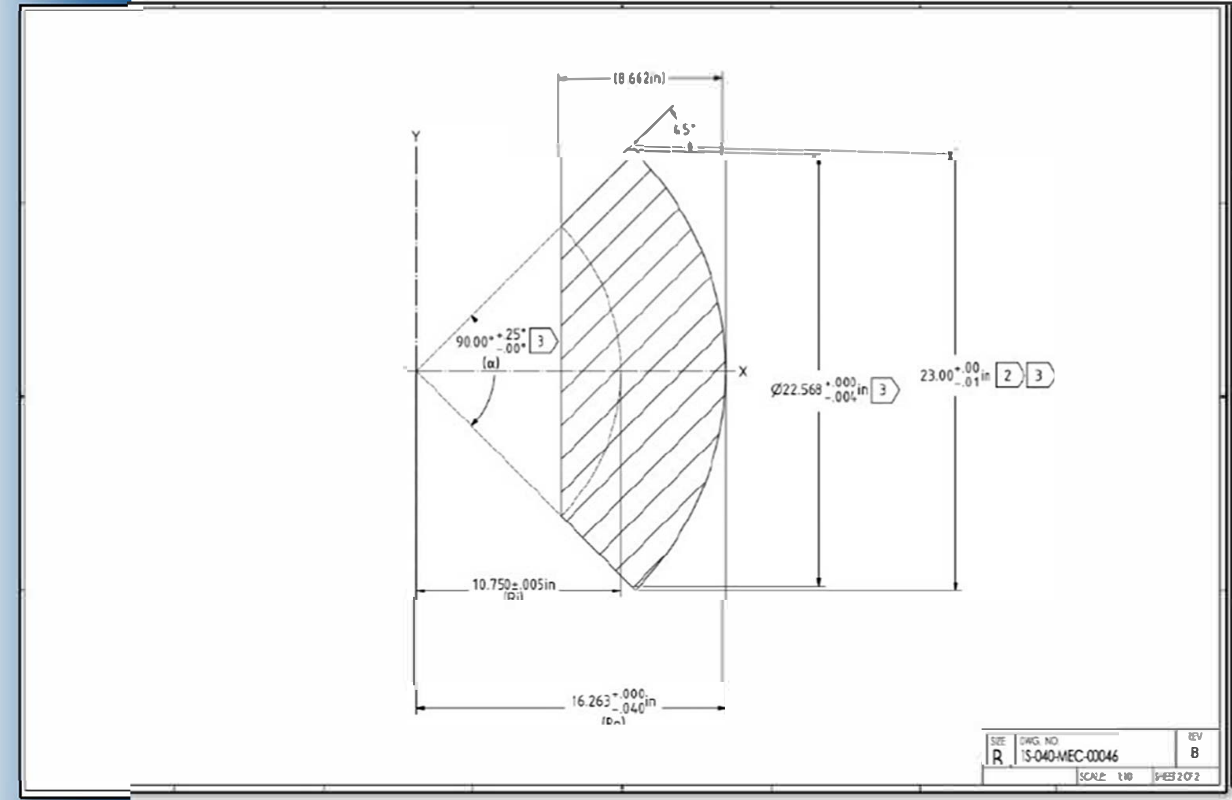
By contrast, the ASME design methodology for designing acrylic pressure vessel components has been largely unchanged from its first version published in 1977 where all materials that meet the single specification (Table 2-3.4-1 and 2-3.4-2) result in the same empirically derived design.[3] Doubling or tripling strength in PVHO-1, while meeting all other requirements, has no ability to change a design. The history of the design method was based on the need for transparent windows to support submersibles, commercial diving, and medical chambers. People being inside these pressure vessels were a driving



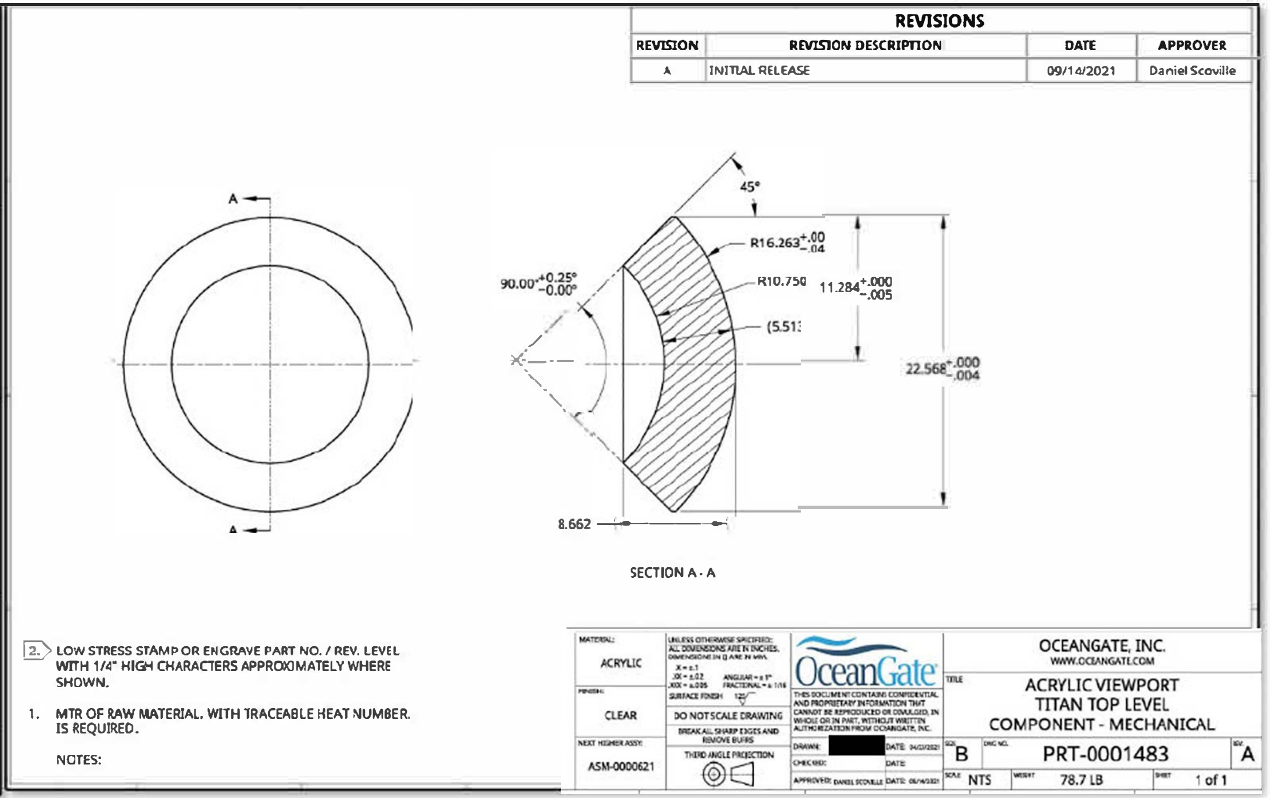


New analyses **modeled the window seat and used nonlinear material and contact elements** to allow the window to slide. Used higher order elements with greater mesh density. Examined the role of angular mismatch (less-than-ideal fit). Found significant correlation to historical data, **validates FEA method for acrylic windows.**

OceanGate Window



Original OceanGate drawing provided in 2017



REVISIONS

REVISION	REVISION DESCRIPTION	DATE	APPROVER
A	INITIAL RELEASE	09/14/2021	Daniel Scoville

SECTION A - A

2. LOW STRESS STAMP OR ENGRAVE PART NO. / REV. LEVEL WITH 1/4" HIGH CHARACTERS APPROXIMATELY WHERE SHOWN.

1. MTR OF RAW MATERIAL, WITH TRACEABLE HEAT NUMBER. IS REQUIRED.

NOTES:

MATERIAL: ACRYLIC
FINISH: CLEAR
NEXT HIGHER ASSY: ASM-0000621
UNLESS OTHERWISE SPECIFIED: ALL DIMENSIONS ARE IN INCHES. DIMENSIONS IN [] ARE IN MM.
X = ±.1
.XX = ±.02
.XXX = ±.005
ANGULAR = ±1°
FRACTIONAL = ± 1/16
SURFACE FINISH 125
DO NOT SCALE DRAWING
BREAK ALL SHARP EDGES AND REMOVE BURRS
THIRD ANGLE PROJECTION
OceanGate
THIS DOCUMENT CONTAINS CONFIDENTIAL AND PROPRIETARY INFORMATION THAT CANNOT BE REPRODUCED OR DIVULGED, IN WHOLE OR IN PART, WITHOUT WRITTEN AUTHORIZATION FROM OCEANGATE, INC.
DRAWN: [redacted] DATE: 04/22/2021
CHECKED: DATE:
APPROVED: DANIEL SCOVILLE DATE: 09/14/2021
OCEANGATE, INC.
WWW.OCEANGATE.COM
TITLE ACRYLIC VIEWPORT TITAN TOP LEVEL COMPONENT - MECHANICAL
SIZE B DWG NO. PRT-0001483 REV A
SCALE NTS WEIGHT 78.7 LB SHEET 1 of 1

OG_USCG_004796_TITAN MBI_ Acrylic Viewport Top Level Component.pdf

OceanGate Window Report (2018)

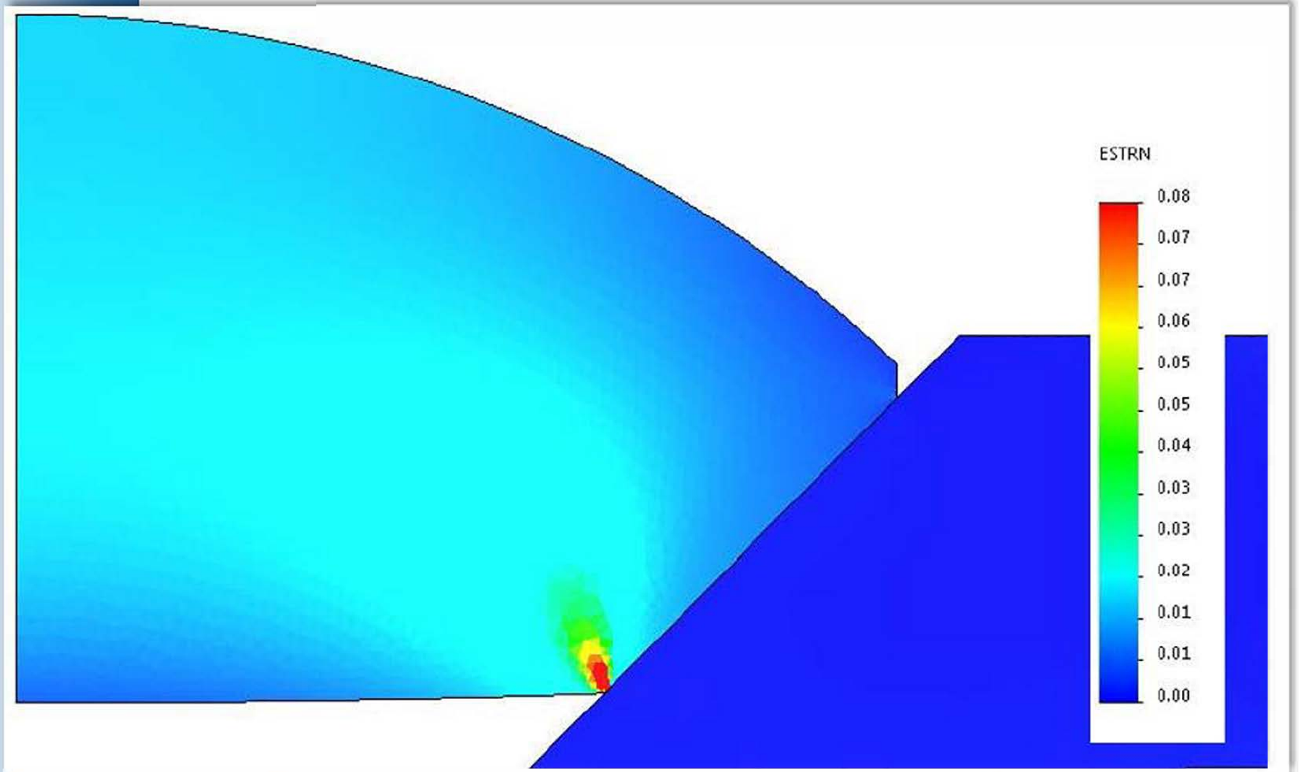


Fig. C5. Nonlinear 5800 psi strain. Strains above 0.06 (red-to-yellow) is of concern. This is consistent with cyclic failure. These results are do not account for heat transfer, dive rates, service life, or creep effects, so any conclusion regarding suitability is preliminary. The intent is to provide a preliminary comparison of the Cyclops design to a flat top design.

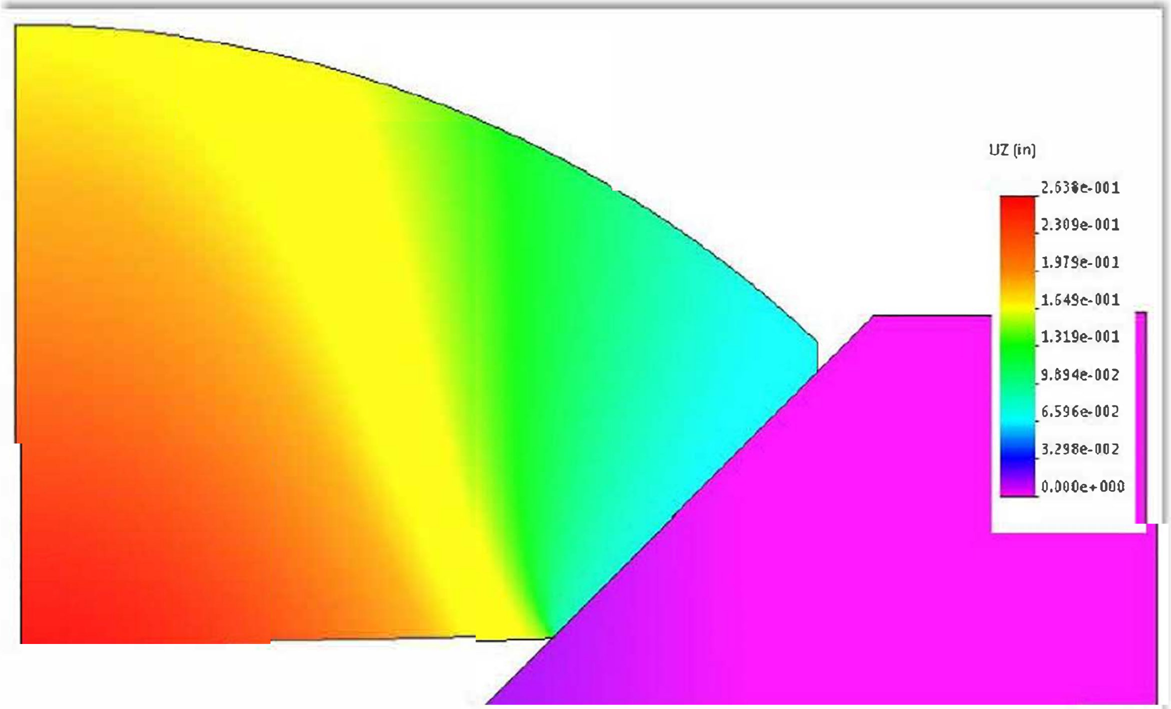
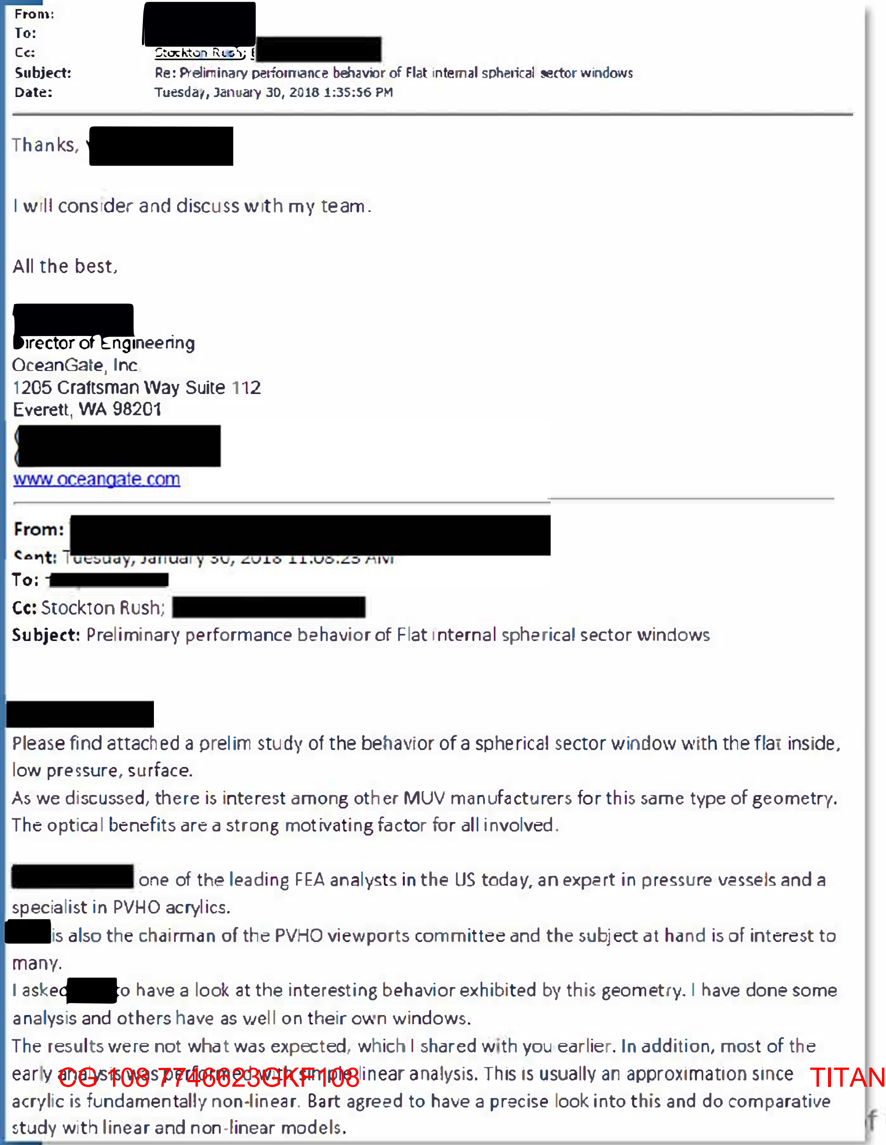


Fig. C6. Nonlinear 5800 psi displacement. Maximum downwards displacement is 0.263 inches.

In news reports, the observed inwards deflection is ¾ of an inch, or 3x the idealized model. It is unknown whether it's measured from the window edge (easier but more severe) or the window's center (per FEA).

OceanGate Window Report (2018)



From: [redacted]
To:
Cc: Stockton Rush; [redacted]
Subject: Re: Preliminary performance behavior of Flat internal spherical sector windows
Date: Tuesday, January 30, 2018 1:35:56 PM

Thanks, [redacted]

I will consider and discuss with my team.

All the best,

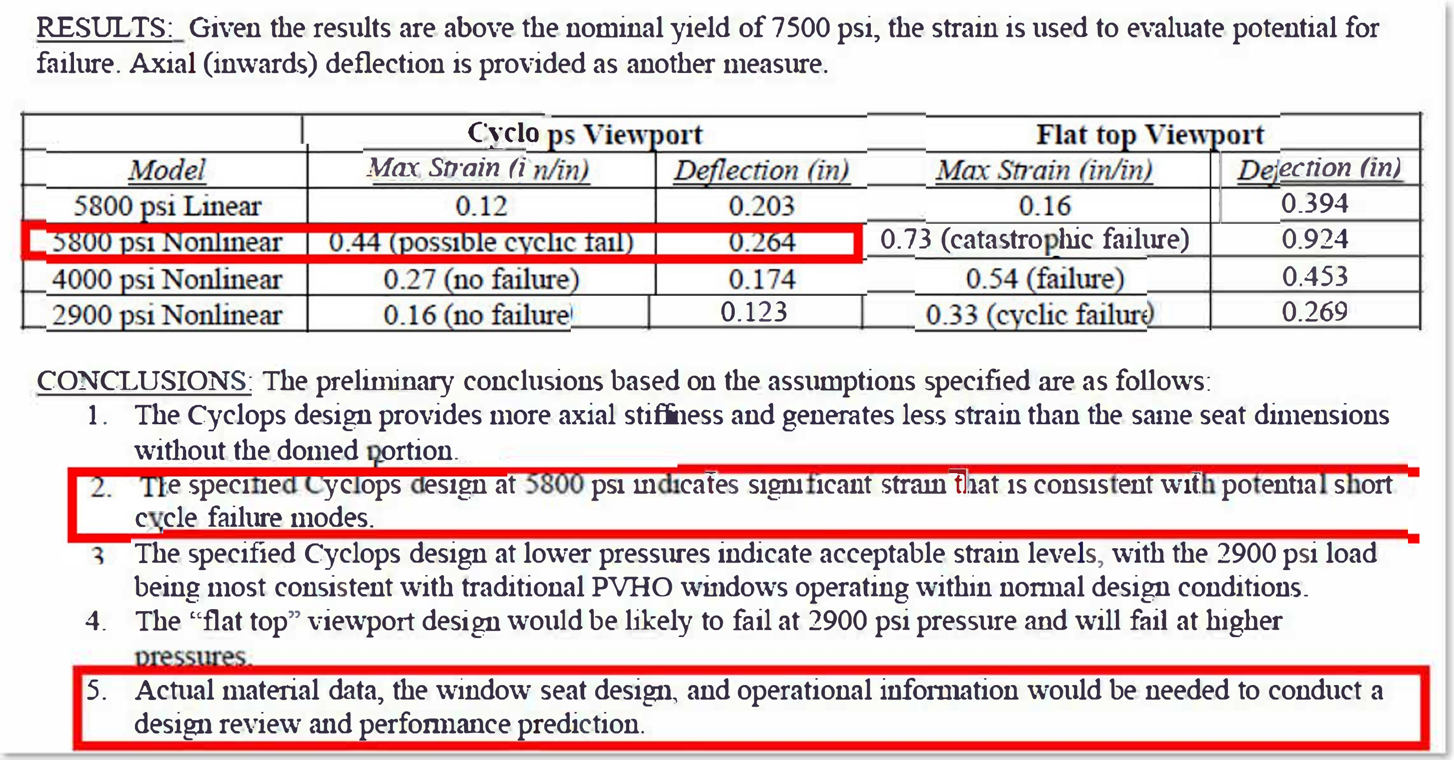
[redacted]
Director of Engineering
OceanGate, Inc
1205 Craftsman Way Suite 112
Everett, WA 98201
[redacted]
www.oceangate.com

From: [redacted]
Sent: Tuesday, January 30, 2018 11:08:25 AM
To: [redacted]
Cc: Stockton Rush; [redacted]
Subject: Preliminary performance behavior of Flat internal spherical sector windows

[redacted]

Please find attached a prelim study of the behavior of a spherical sector window with the flat inside, low pressure, surface.
As we discussed, there is interest among other MUV manufacturers for this same type of geometry.
The optical benefits are a strong motivating factor for all involved.

[redacted] one of the leading FEA analysts in the US today, an expert in pressure vessels and a specialist in PVHO acrylics.
[redacted] is also the chairman of the PVHO viewports committee and the subject at hand is of interest to many.
I asked [redacted] to have a look at the interesting behavior exhibited by this geometry. I have done some analysis and others have as well on their own windows.
The results were not what was expected, which I shared with you earlier. In addition, most of the early analysis was performed with simple linear analysis. This is usually an approximation since acrylic is fundamentally non-linear. Bart agreed to have a precise look into this and do comparative study with linear and non-linear models.



RESULTS: Given the results are above the nominal yield of 7500 psi, the strain is used to evaluate potential for failure. Axial (inwards) deflection is provided as another measure.

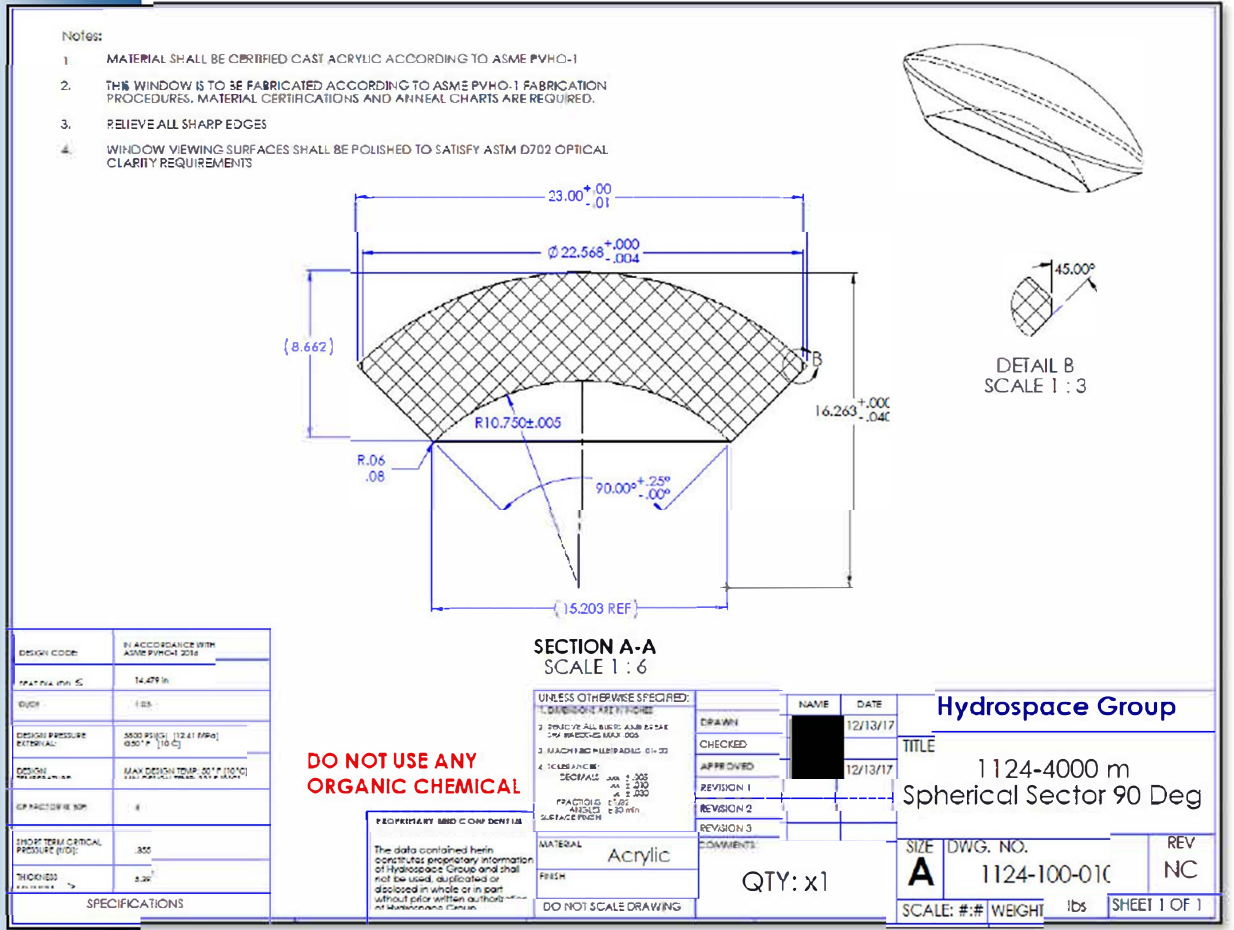
	Cyclops Viewport		Flat top Viewport	
Model	*Max Strain (in/in)*	*Deflection (in)*	*Max Strain (in/in)*	*Deflection (in)*
5800 psi Linear	0.12	0.203	0.16	0.394
5800 psi Nonlinear	0.44 (possible cyclic fail)	0.264	0.73 (catastrophic failure)	0.924
4000 psi Nonlinear	0.27 (no failure)	0.174	0.54 (failure)	0.453
2900 psi Nonlinear	0.16 (no failure)	0.123	0.33 (cyclic failure)	0.269

CONCLUSIONS: The preliminary conclusions based on the assumptions specified are as follows:

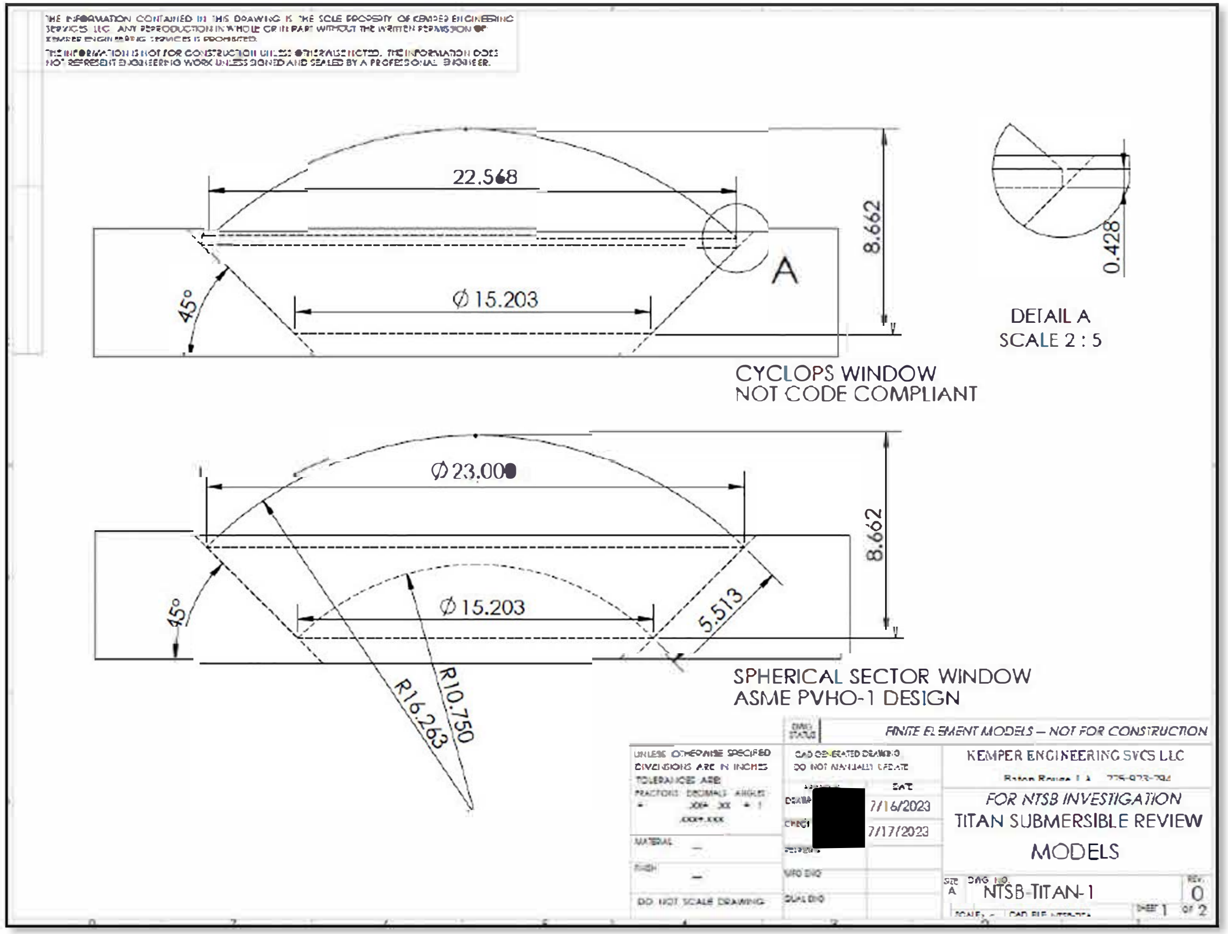
1. The Cyclops design provides more axial stiffness and generates less strain than the same seat dimensions without the domed portion.
2. The specified Cyclops design at 5800 psi indicates significant strain that is consistent with potential short cycle failure modes.
3. The specified Cyclops design at lower pressures indicate acceptable strain levels, with the 2900 psi load being most consistent with traditional PVHO windows operating within normal design conditions.
4. The "flat top" viewport design would be likely to fail at 2900 psi pressure and will fail at higher pressures.
5. Actual material data, the window seat design, and operational information would be needed to conduct a design review and performance prediction.

[redacted] asked KES for an analysis of the OceanGate design. The results indicated it was possible the window would fail in cyclic failure. We did not have the head design to analyze the mechanical response of the pressure boundary to the dive profile. **The intent was to start a discussion to agree upon a safer, more reliable solution.**

OceanGate Window



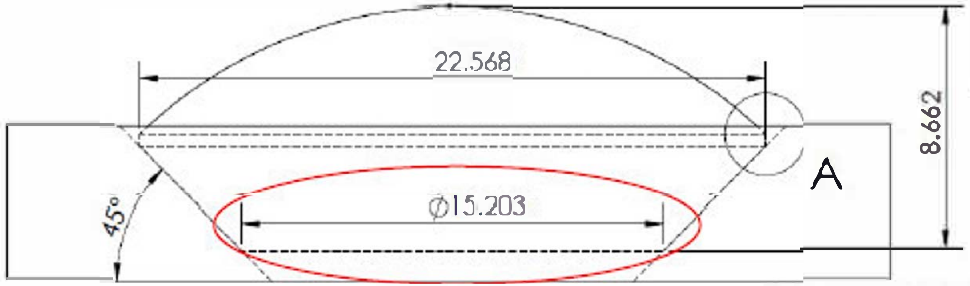
Hydrospace spherical sector drawing provided in 2017



KES drawings of OceanGate design and the ASME PVHO-1 spherical sector equivalent

ASME PVHO-1 Window Design

Units are US Customary to be consistent with OceanGate. Spherical sector window in the same footprint as the Cyclops II (Titan) window design, but uses less acrylic.



CYCLOPS WINDOW
NOT CODE COMPLIANT

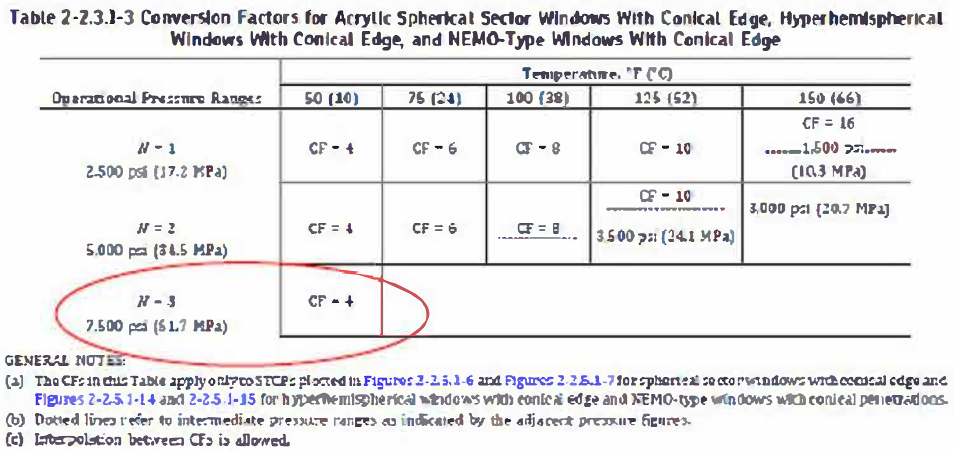


Table 2-2.3.1-3 Conversion Factors for Acrylic Spherical Sector Windows With Conical Edge, Hyperhemispherical Windows With Conical Edge, and NEMO-Type Windows With Conical Edge

	Temperature, °F (°C)				
Operational Pressure Ranges	50 (10)	75 (24)	100 (38)	125 (52)	150 (66)
N = 1 2,500 psi (17.2 MPa)	CF = 4	CF = 6	CF = 8	CF = 10	CF = 16 1,500 psi (10.3 MPa)
N = 2 5,000 psi (34.5 MPa)	CF = 4	CF = 6	CF = 8	CF = 10 3,500 psi (24.1 MPa)	3,000 psi (20.7 MPa)
N = 3 7,500 psi (51.7 MPa)	CF = 4				

GENERAL NOTES:

(a) The CFs in this Table apply only to STCPs plotted in Figures 2-2.5.1-6 and Figures 2-2.5.1-7 for spherical sector windows with conical edge and Figures 2-2.5.1-14 and 2-2.5.1-15 for hyperhemispherical windows with conical edge and NEMO-type windows with conical penetrations.

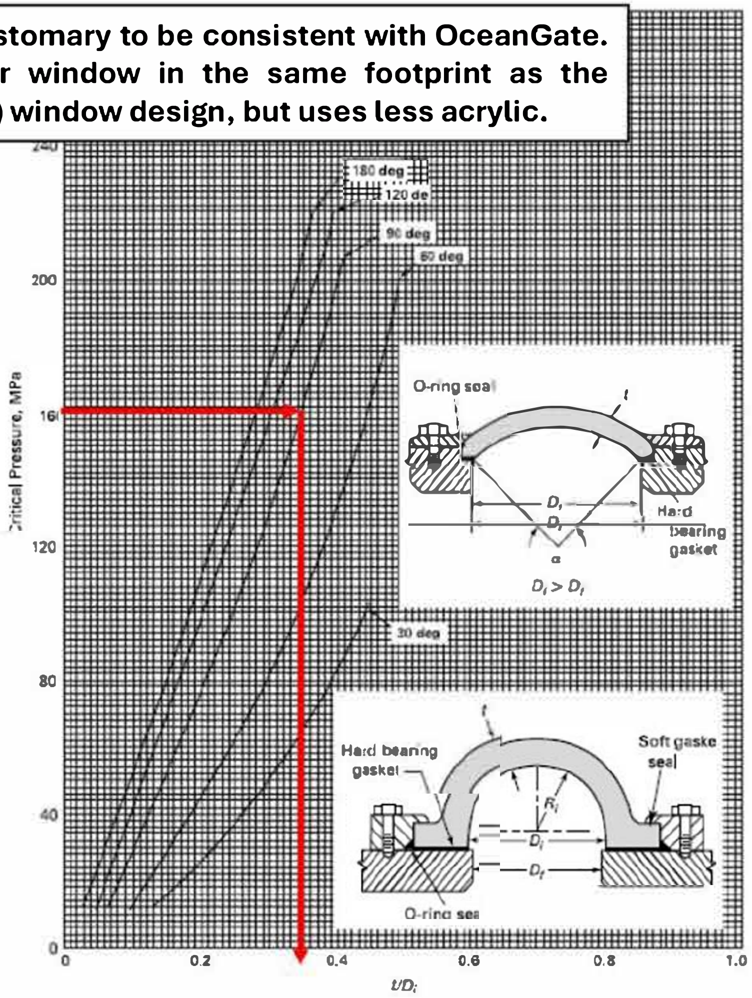
(b) Dotted lines refer to intermediate pressure ranges as indicated by the adjacent pressure figures.

(c) Interpolation between CFs is allowed.

Design pressure = 5800 psi

CF for 5800 psi = 4

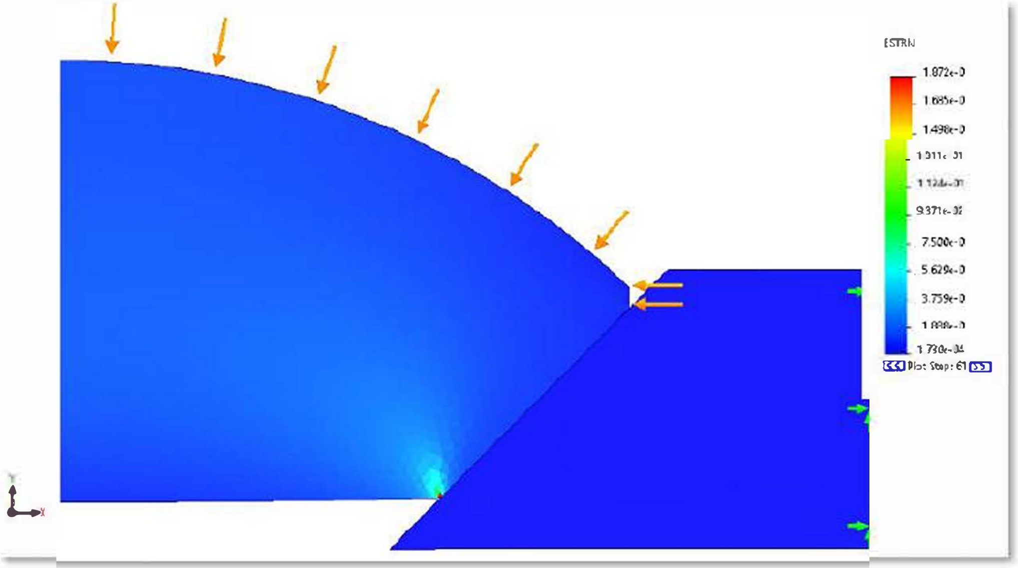
STCP = 4 * 5800 psi = 23,200 psi (160 MPa) (Use 160 MPa due to graph units)

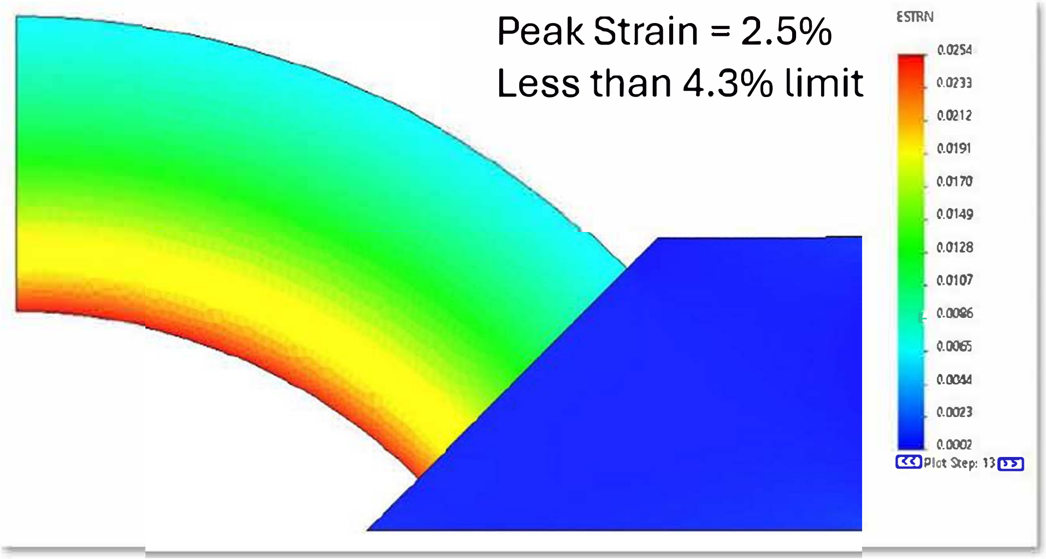


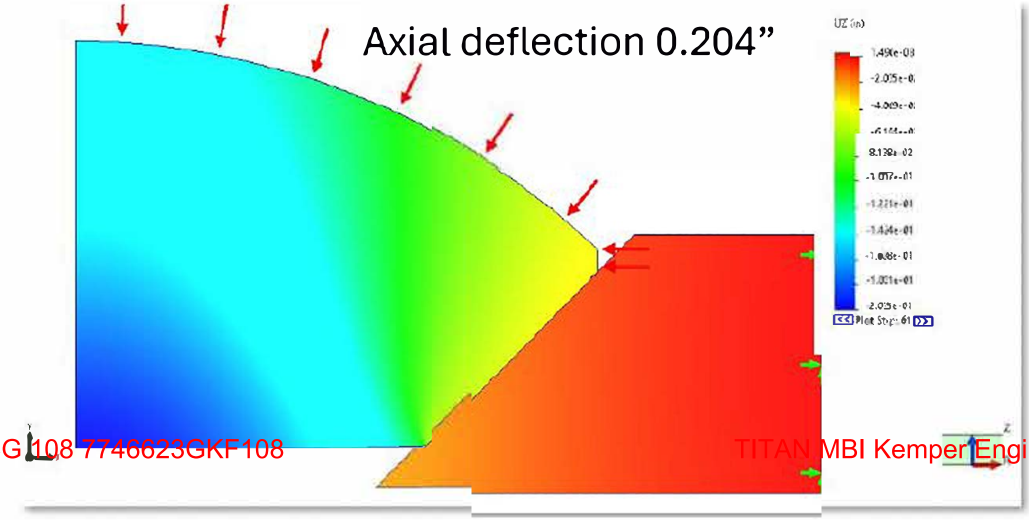
t/Di = 0.36

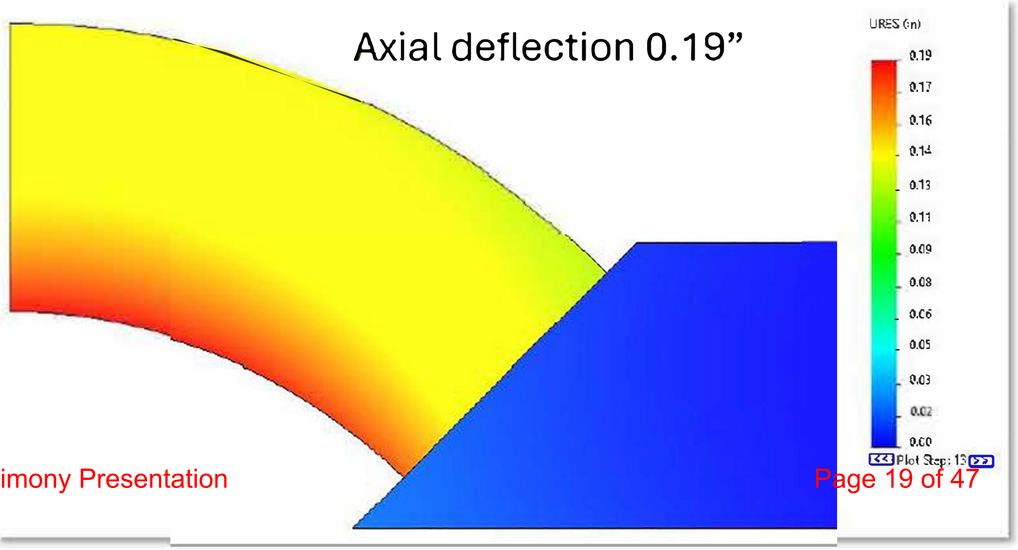
t = Di * (t/Di) = 15.2 * 0.36 = 5.47 inches, rounded up to 5.5 inches

Post incident analysis (full depth)

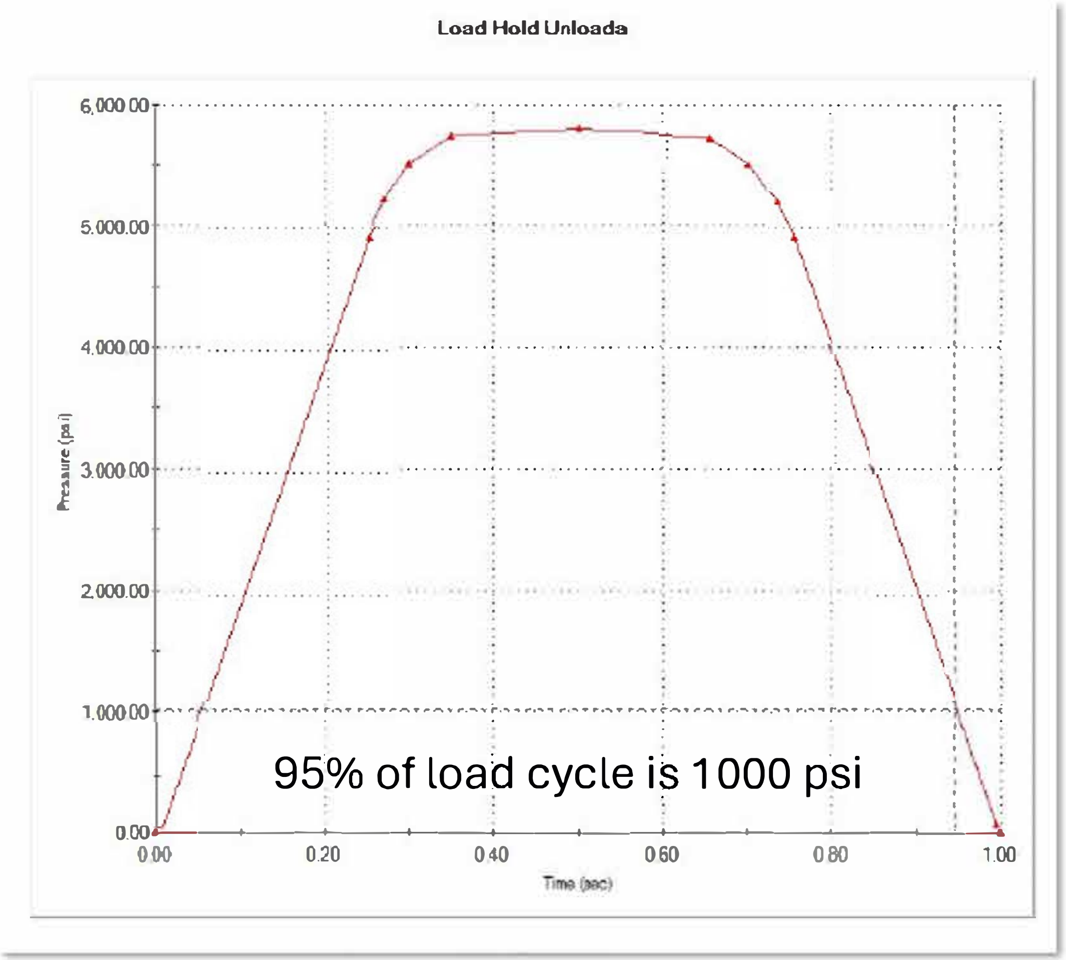



Peak Strain = 2.5%
Less than 4.3% limit

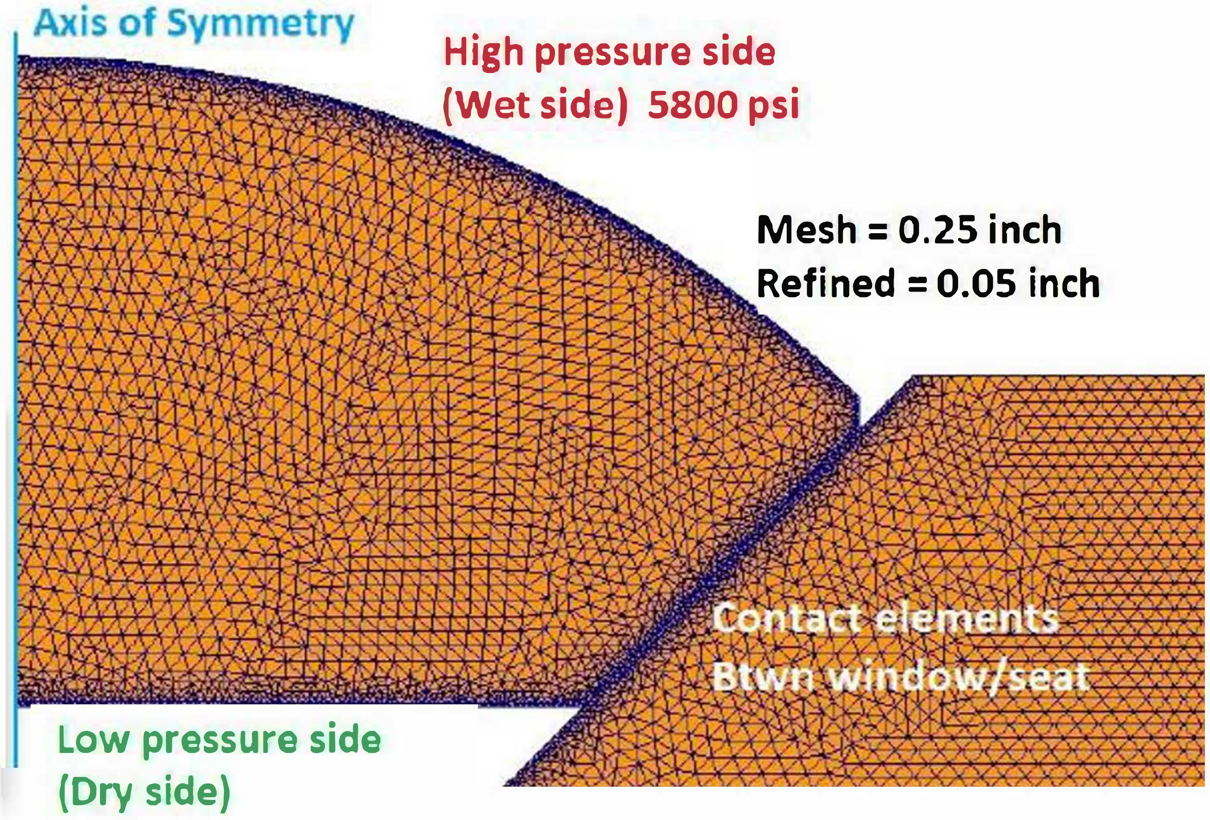

Axial deflection 0.204”


Axial deflection 0.19”

Post-incident analysis (1 cycle, no creep)



95% of load cycle is 1000 psi



Axis of Symmetry

High pressure side
(Wet side) 5800 psi

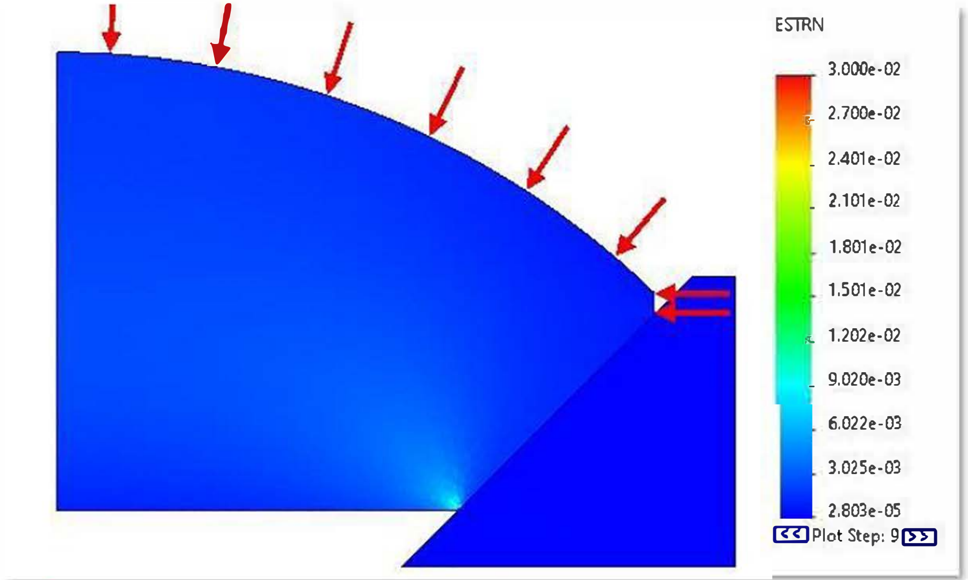
Mesh = 0.25 inch
Refined = 0.05 inch

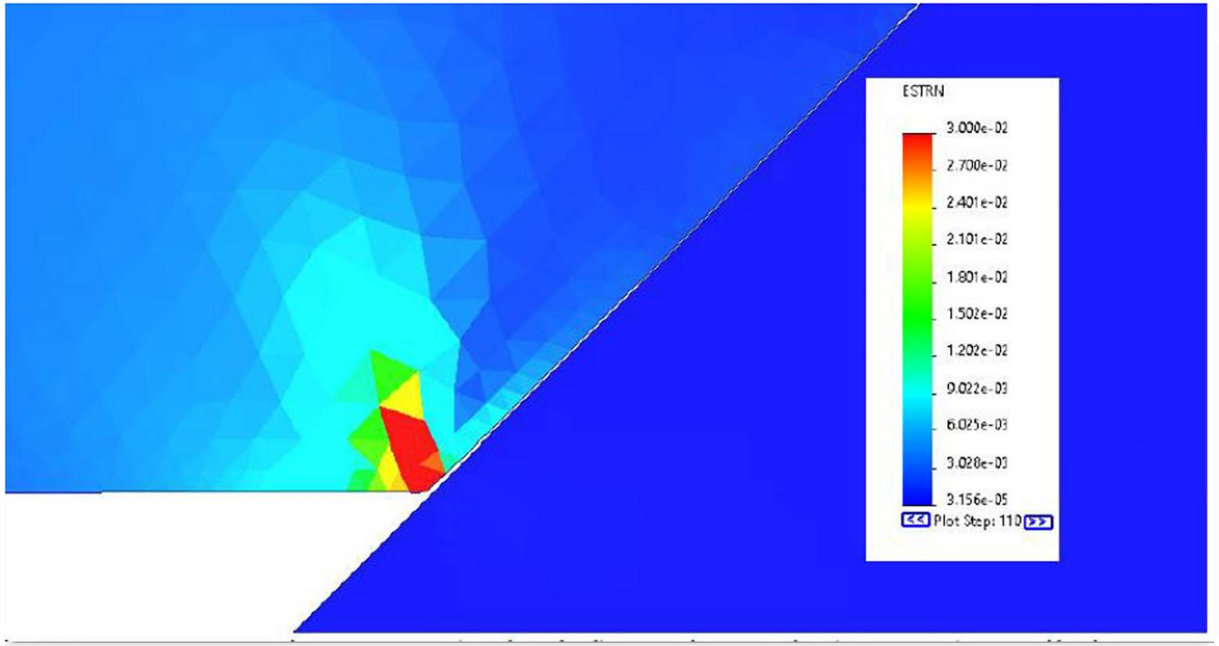
Contact elements
Btwn window/seat

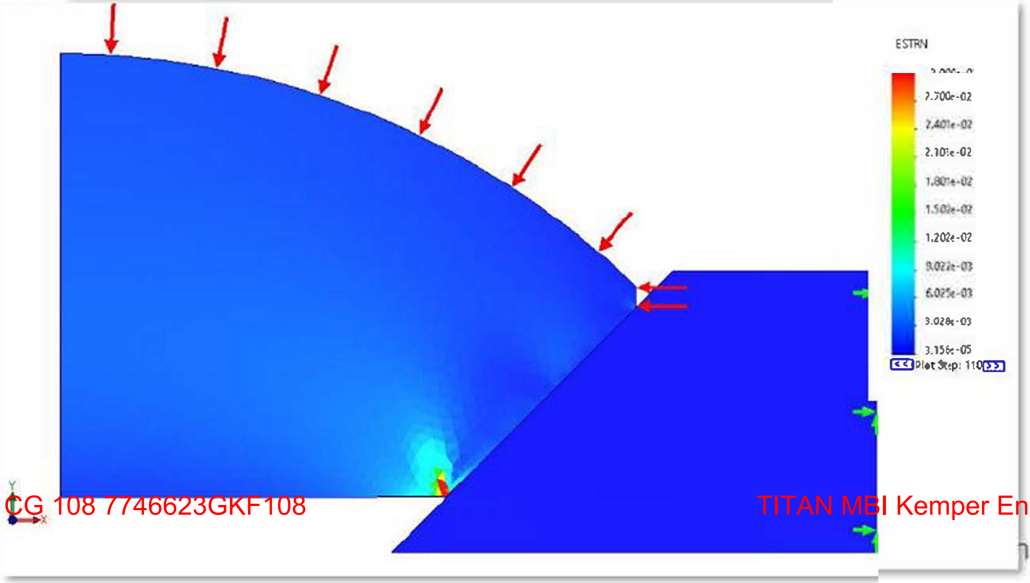
Low pressure side
(Dry side)

Later analysis used a cyclic loading/unloading to assess the potential for progressive creep due to plastic deformation.

Post-incident analysis (1 cycle)

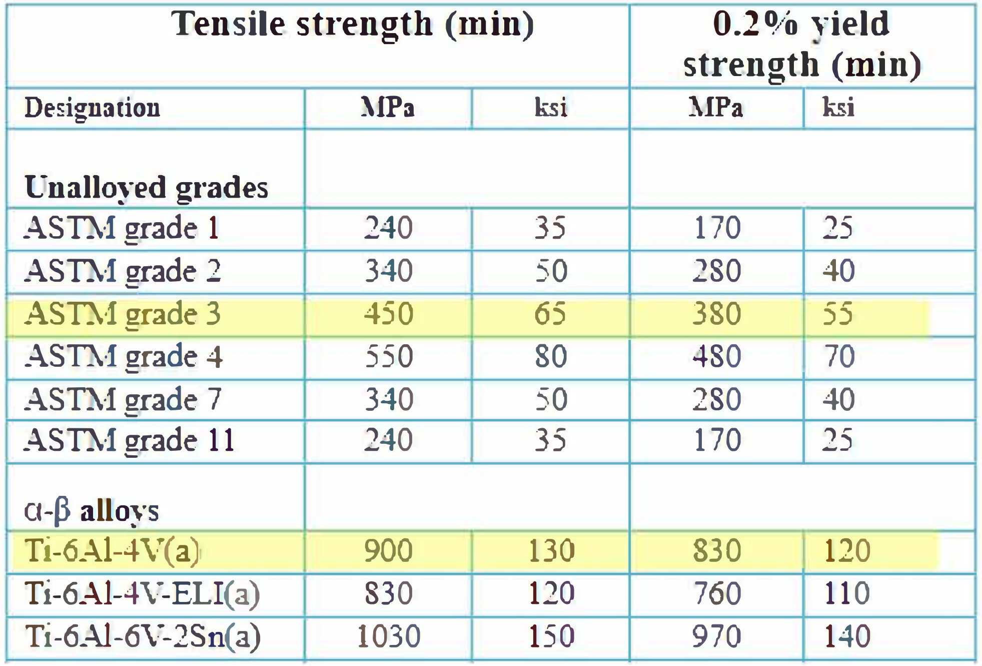




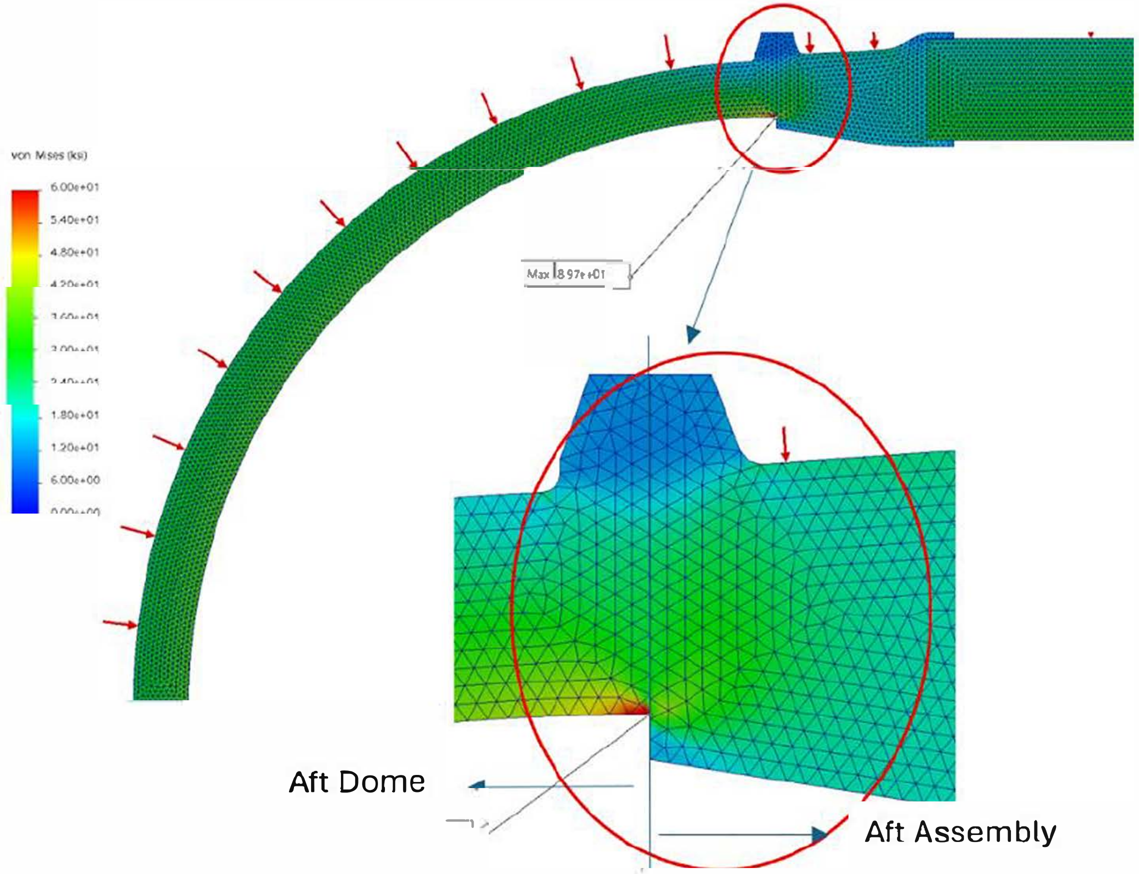


Upper left shows the full range of strain at 1000 psi loading is 1% strain (0.023" center). At the same 1000 psi in unloading, the 16% peak strain (0.032" center), then both are shown at 3% strain. The upper region remains elastic. There is significant residual strain and deflection after one cycle. **Each cycle creates more deflection, strain at low pressure face**. No creep (cold flow) modeled.

Titanium domes & ring



Tensile strength (min)			0.2% yield strength (min)	
Designation	MPa	ksi	MPa	ksi
Unalloyed grades				
ASTM grade 1	240	35	170	25
ASTM grade 2	340	50	280	40
ASTM grade 3	450	65	380	55
ASTM grade 4	550	80	480	70
ASTM grade 7	340	50	280	40
ASTM grade 11	240	35	170	25
α-β alloys				
Ti-6Al-4V(a)	900	130	830	120
Ti-6Al-4V-ELI(a)	830	120	760	110
Ti-6Al-6V-2Sn(a)	1030	150	970	140



Grade 5 (Ti 6Al-4V) to Grade 3

Yield 120 ksi ➔ 55 ksi= 45% of Gr5

Tensile 130 ksi ➔ 65 ksi = 50% of Gr5

https://www.dierk-raabe.com/titanium-alloys/mechanical-properties-of-titanium/

Initial linear results indicates **no danger of collapse,** detailed analysis will be in the report.

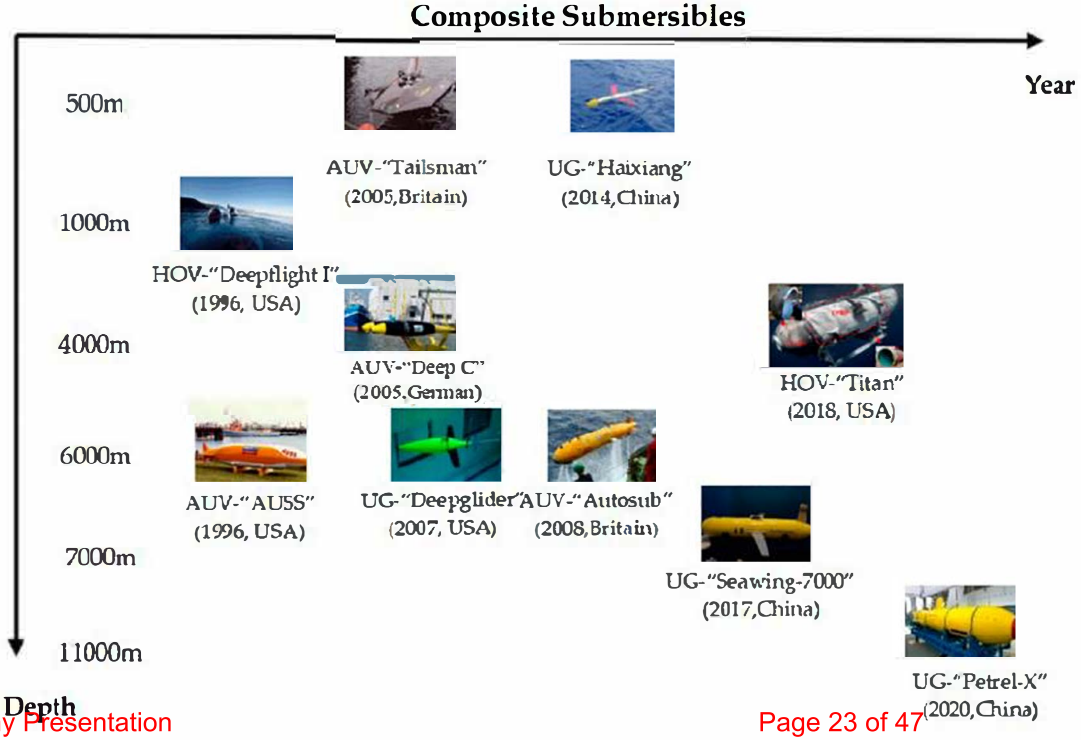
Insufficient data to evaluate the glued joints.

Why carbon fiber

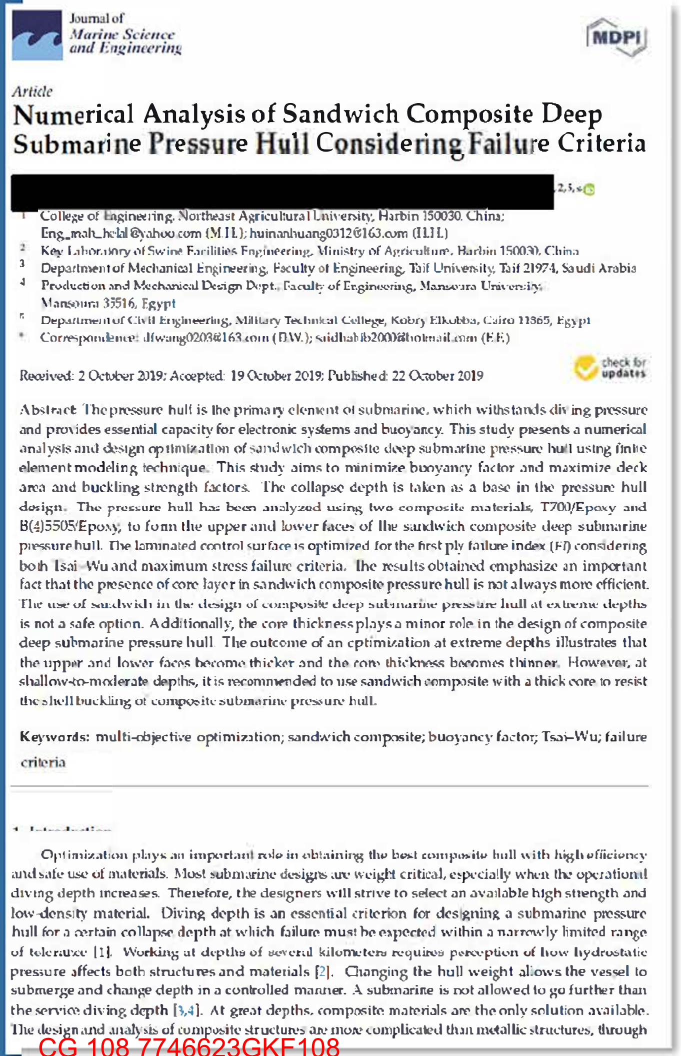
- Long lead time, high costs with alloy steel or titanium
- Normalized for tensile load, carbon fiber-reinforced composites are 70 percent lighter than steel and 40 percent lighter than aluminum. **Key for submersibles.**
- Ultimate tensile strength: steel 400 - 690 MPa, carbon fiber 1,200-2,410 MPa, *depending on fiber orientation*.

- Long-term corrosion resistance to chemical and temperature environments.
- Generally less expensive material and fabrication, depending on labor, novelty.
- Long lead time with alloy steel or titanium, constrains production
- Compressive strength and shear resistance is problematic with carbon fiber.

(Fig ref 36)



Examples of carbon fiber



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Article

Numerical Analysis of Sandwich Composite Deep Submarine Pressure Hull Considering Failure Criteria

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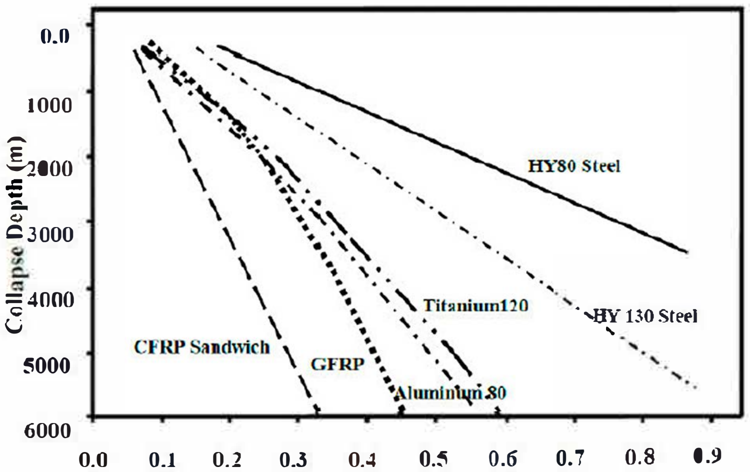
Received: 2 October 2019; Accepted: 19 October 2019; Published: 22 October 2019

Abstract: The pressure hull is the primary element of submarine, which withstands diving pressure and provides essential capacity for electronic systems and buoyancy. This study presents a numerical analysis and design optimization of sandwich composite deep submarine pressure hull using finite element modeling technique. This study aims to minimize buoyancy factor and maximize deck area and buckling strength factors. The collapse depth is taken as a base in the pressure hull design. The pressure hull has been analyzed using two composite materials, T700/Epoxy and B(4)5505/Epoxy, to form the upper and lower faces of the sandwich composite deep submarine pressure hull. The laminated control surface is optimized for the first ply failure index (*FI*) considering both Tsai–Wu and maximum stress failure criteria. The results obtained emphasize an important fact that the presence of core layer in sandwich composite pressure hull is not always more efficient. The use of sandwich in the design of composite deep submarine pressure hull at extreme depths is not a safe option. Additionally, the core thickness plays a minor role in the design of composite deep submarine pressure hull. The outcome of an optimization at extreme depths illustrates that the upper and lower faces become thicker and the core thickness becomes thinner. However, at shallow-to-moderate depths, it is recommended to use sandwich composite with a thick core to resist the shell buckling of composite submarine pressure hull.

Keywords: multi-objective optimization; sandwich composite; buoyancy factor; Tsai–Wu; failure criteria

1. Introduction

Optimization plays an important role in obtaining the best composite hull with high efficiency and safe use of materials. Most submarine designs are weight critical, especially when the operational diving depth increases. Therefore, the designers will strive to select an available high strength and low-density material. Diving depth is an essential criterion for designing a submarine pressure hull for a certain collapse depth at which failure must be expected within a narrowly limited range of tolerance [1]. Working at depths of several kilometers requires perception of how hydrostatic pressure affects both structures and materials [2]. Changing the hull weight allows the vessel to submerge and change depth in a controlled manner. A submarine is not allowed to go further than the service diving depth [3,4]. At great depths, composite materials are the only solution available. The design and analysis of composite structures are more complicated than metallic structures, through



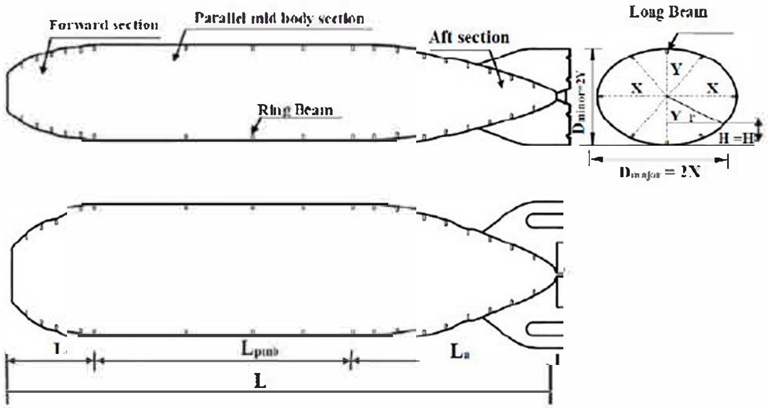
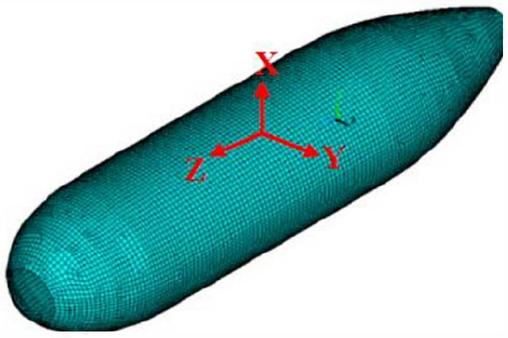
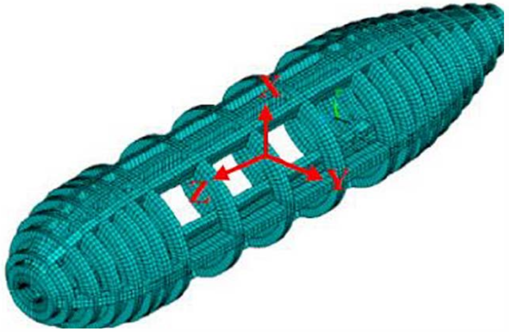


Figure 1. Parameterization of non-circular submersible pressure hulls.



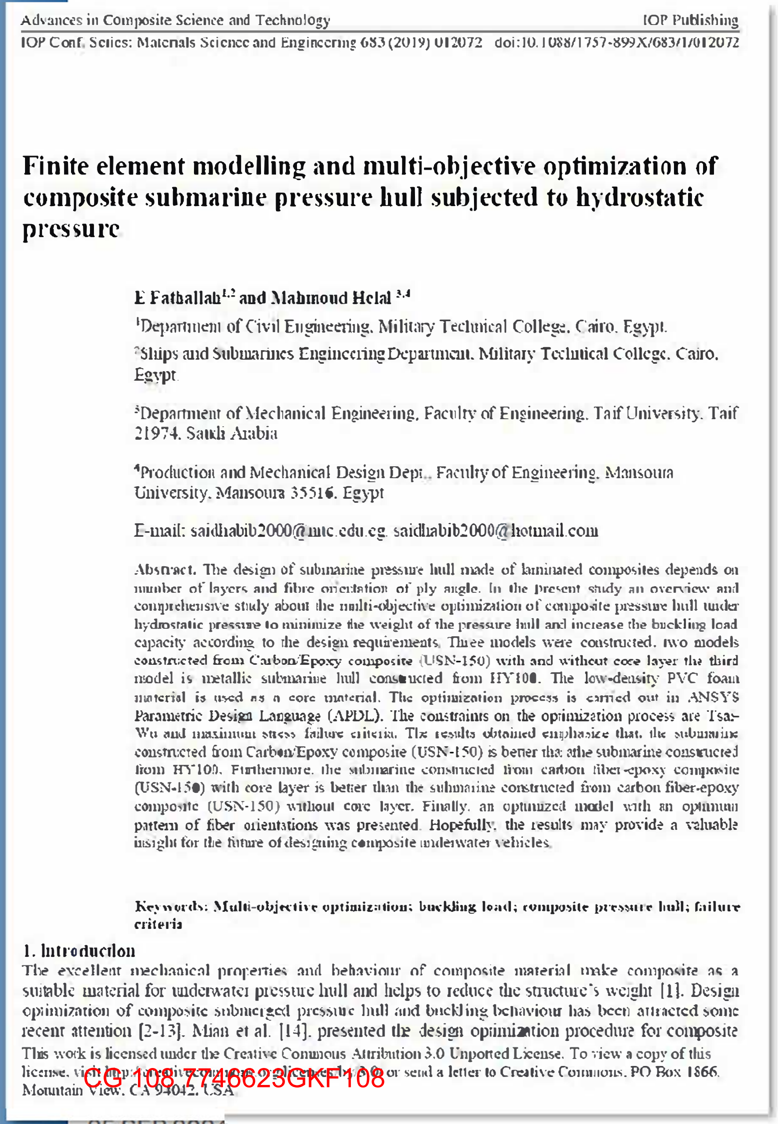
a) Shell element (SHELL281).



b) Beam element (BEAM189).

Figure 4. Mesh of global model.

Examples of carbon fiber



Advances in Composite Science and Technology IOP Publishing
IOP Conf. Series: Materials Science and Engineering 683 (2019) 012072 doi:10.1088/1757-899X/683/1/012072

Finite element modelling and multi-objective optimization of composite submarine pressure hull subjected to hydrostatic pressure

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Abstract. The design of submarine pressure hull made of laminated composites depends on number of layers and fibre orientation of ply angle. In the present study an overview and comprehensive study about the multi-objective optimization of composite pressure hull under hydrostatic pressure to minimize the weight of the pressure hull and increase the buckling load capacity according to the design requirements. Three models were constructed, two models constructed from Carbon/Epoxy composite (USN-150) with and without core layer the third model is metallic submarine hull constructed from HY100. The low-density PVC foam material is used as a core material. The optimization process is carried out in ANSYS Parametric Design Language (APDL). The constraints on the optimization process are Tsai-Wu and maximum stress failure criteria. The results obtained emphasize that, the submarine constructed from Carbon/Epoxy composite (USN-150) is better than the submarine constructed from HY100. Furthermore, the submarine constructed from carbon fiber-epoxy composite (USN-150) with core layer is better than the submarine constructed from carbon fiber-epoxy composite (USN-150) without core layer. Finally, an optimized model with an optimum pattern of fiber orientations was presented. Hopefully, the results may provide a valuable insight for the future of designing composite underwater vehicles.

Keywords: Multi-objective optimization; buckling load; composite pressure hull; failure criteria

1. Introduction

The excellent mechanical properties and behaviour of composite material make composite as a suitable material for underwater pressure hull and helps to reduce the structure's weight [1]. Design optimization of composite submerged pressure hull and buckling behaviour has been attracted some recent attention [2-13]. Mian et al. [14]. presented the design optimization procedure for composite

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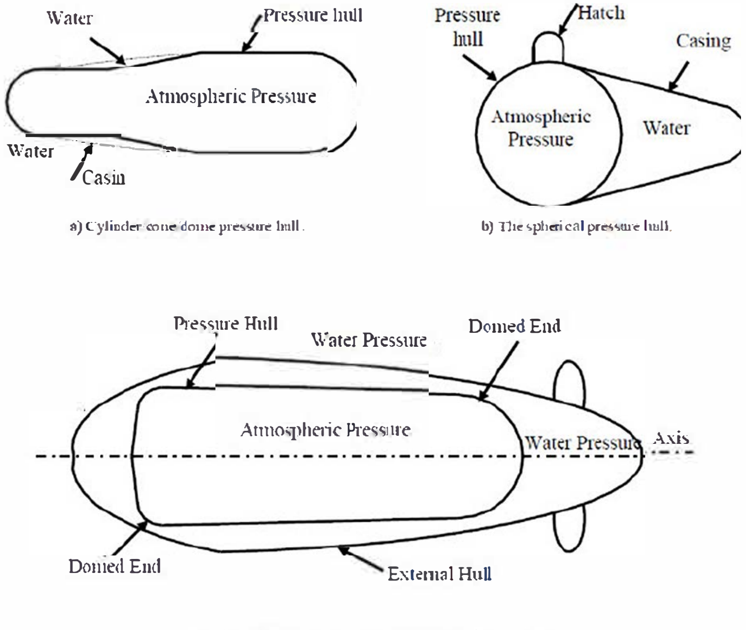
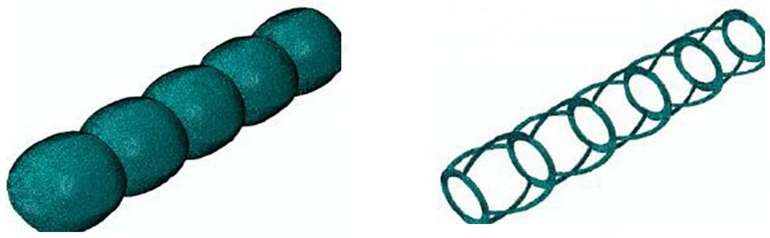


a) Cylinder cone dome pressure hull. b) The spherical pressure hull.

c) Ring-stiffened circular cylinder, blocked by end caps.

Figure 1 Possible forms of submerged vehicles.



a) Shell element. b) Beam Element.

Figure 3 Finite element modeling of the cross-elliptical submersible pressure hulls.

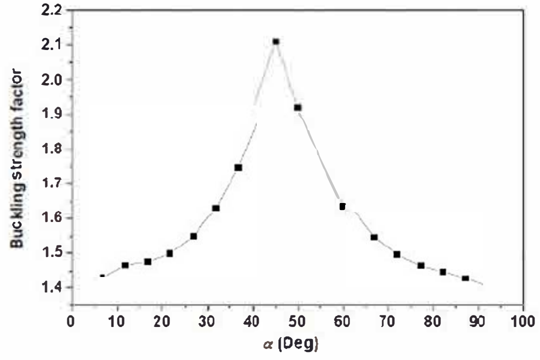
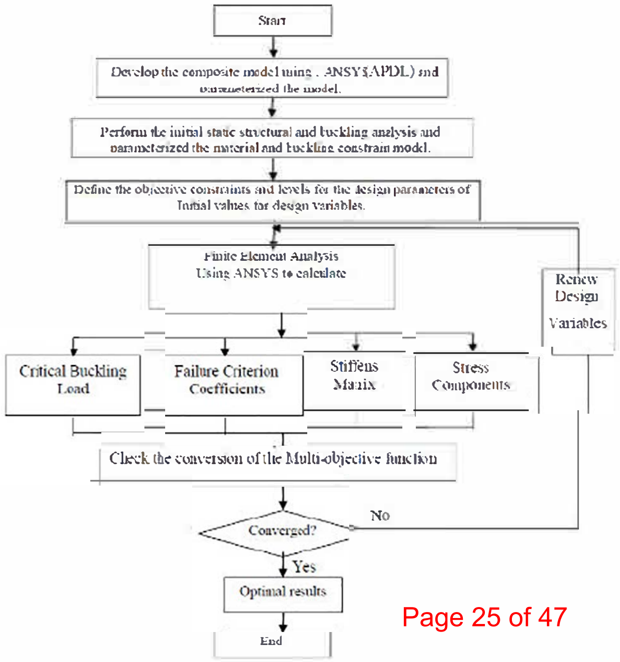


Figure 5 Effect of buckling strength factor by fiber orientation α



Example of Factors to Consider

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Review

A Review on Structural Failure of Composite Pressure Hulls in Deep Sea

[redacted]

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Abstract: With the increasing application and study of lightweight and high strength fiber reinforced polymer composites in ocean industry, the structural failure problem of composite pressure hulls has attracted great attention from many researchers in China and globally. Analysis of the structural failure mechanisms is foundational to the design of deep-sea composite pressure hulls, since nowadays the design rules of pressurized vessels is mostly formulated according to their failure modes. Hence this paper aims to review the research on the structural failure of composite pressure hulls in deep sea settings. First of all, the applied research status on composite material in marine equipment is analyzed, including inspection modalities for composite pressure hulls. The review then focuses on the three main failure modes, namely overall buckling, material failure and snap buckling of the deep-sea composite pressure hulls. The study identifies further problems of composite pressure hulls to be solved through the application of the deep sea equipment research, aiming to provide a reference for the study of mechanical behavior, ultimate strength computation, and design of thick composite pressure hulls for deep sea equipment.

Keywords: marine equipment; composite material; pressure hull; structural failure

check for updates

Citation: Li, Y.; Yu, C.; Wang, W.; Li, H.; Jiang, X. A Review on Structural Failure of Composite Pressure Hulls in Deep Sea. *J. Mar. Sci. Eng.* **2022**, *10*, 1456. https://doi.org/10.3390/jmse10101456

Academic Editor: José António Correia

Received: 12 September 2022
Accepted: 3 October 2022
Published: 9 October 2022

1. Introduction

In recent years, the research frontier of equipment technology has developed rapidly in the deep sea field. Various types of high-tech deep-sea equipment have been widely used in marine scientific research, resource exploration, military security and other fields. The composite pressure hull of a submersible is the main provider of buoyancy, and its weight accounts for about 1/4~1/2 of the total weight of the underwater vehicle. Perhaps the most critical consideration for the pressure hull of a submersible is the weight to displacement ratio. The structural material of a pressure hull is one of the main factors that determines its weight-displacement ratio and structural bearing capacity, which is also related to the progressiveness and reliability of submersibles. High strength steel is the most widely used material in general submersibles, and in recent years, titanium alloys with higher specific strength have been gradually adopted. However, deep submersibles have higher material requirements for specific strength and stiffness. Considering all the factors, fiber reinforced polymer (FRP) has greater advantages. In addition to high specific strength and specific stiffness characteristics, composite materials also have a comprehensive range of properties such as the excellent designability, seawater corrosion resistance, acoustic stealth and fatigue resistance. Therefore, composite materials are ideal structural materials for deep sea pressure hulls, and have been widely studied by scholars [1–5]. They have been used in deep-sea submersibles worldwide, such as Autonomous Underwater Vehicles (AUV) [6,7], Underwater Glider (UG) [8–11] and Human Occupied Vehicle (HOV) [12–14]; see Figure 1. Composite materials also have great potential in marine equipment, such as composite

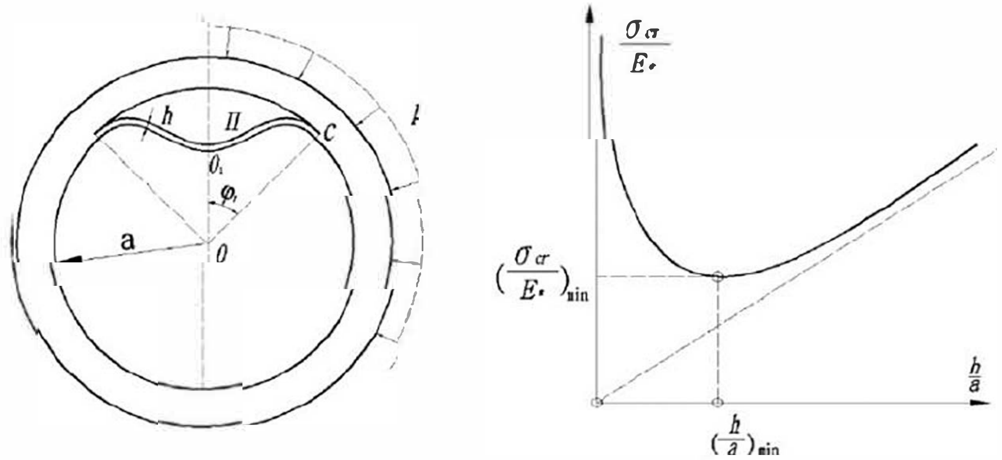


Figure 9. Delamination buckling failure mode and critical load of deep sea thick shell

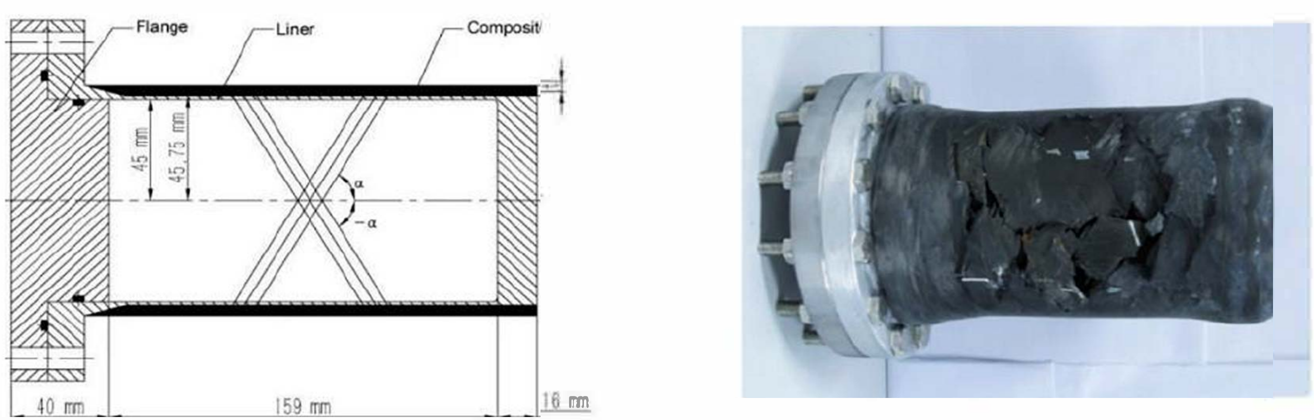
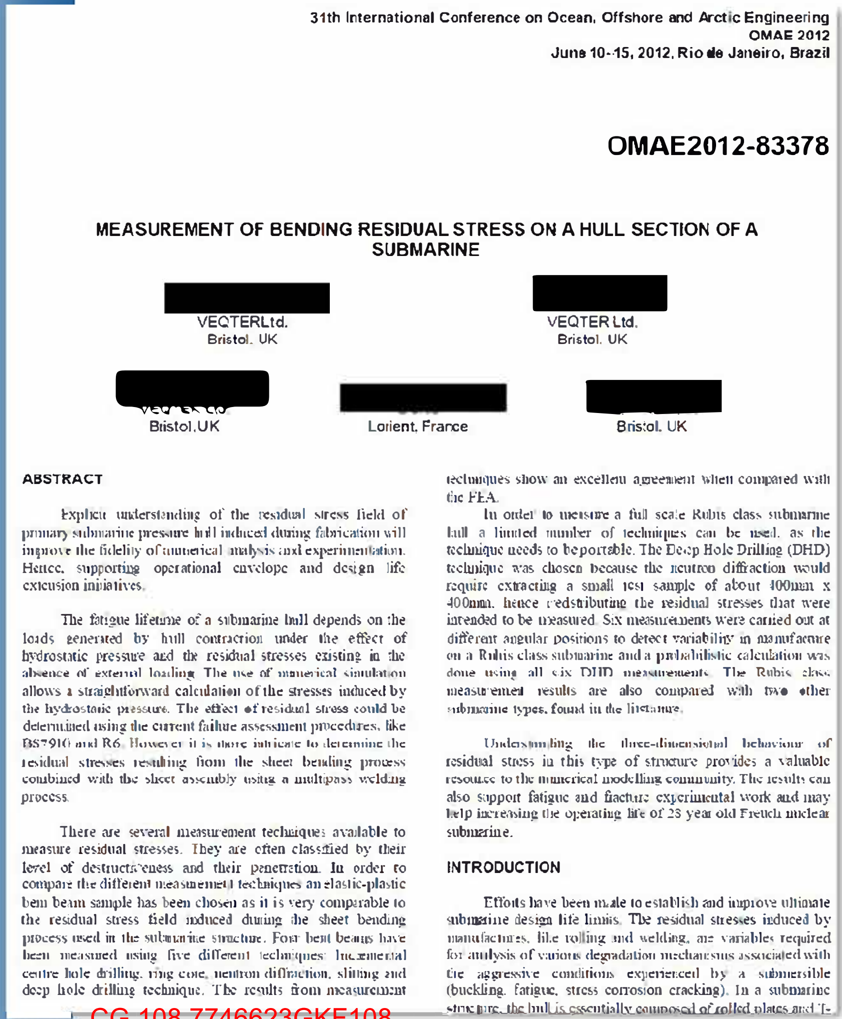


Figure 3. Hydrostatic damage test of carbon fiber composite pressure hull with aluminum liner

Examples of Factors to Consider



31th International Conference on Ocean, Offshore and Arctic Engineering
OMAE 2012
June 10-15, 2012, Rio de Janeiro, Brazil

OMAE2012-83378

MEASUREMENT OF BENDING RESIDUAL STRESS ON A HULL SECTION OF A SUBMARINE

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ABSTRACT

Explicit understanding of the residual stress field of primary submarine pressure hull induced during fabrication will improve the fidelity of numerical analysis and experimentation. Hence, supporting operational envelope and design life extension initiatives.

The fatigue lifetime of a submarine hull depends on the loads generated by hull contraction under the effect of hydrostatic pressure and the residual stresses existing in the absence of external loading. The use of numerical simulation allows a straightforward calculation of the stresses induced by the hydrostatic pressure. The effect of residual stress could be determined using the current failure assessment procedures, like BS7910 and R6. However it is more intricate to determine the residual stresses resulting from the sheet bending process combined with the sheet assembly using a multipass welding process.

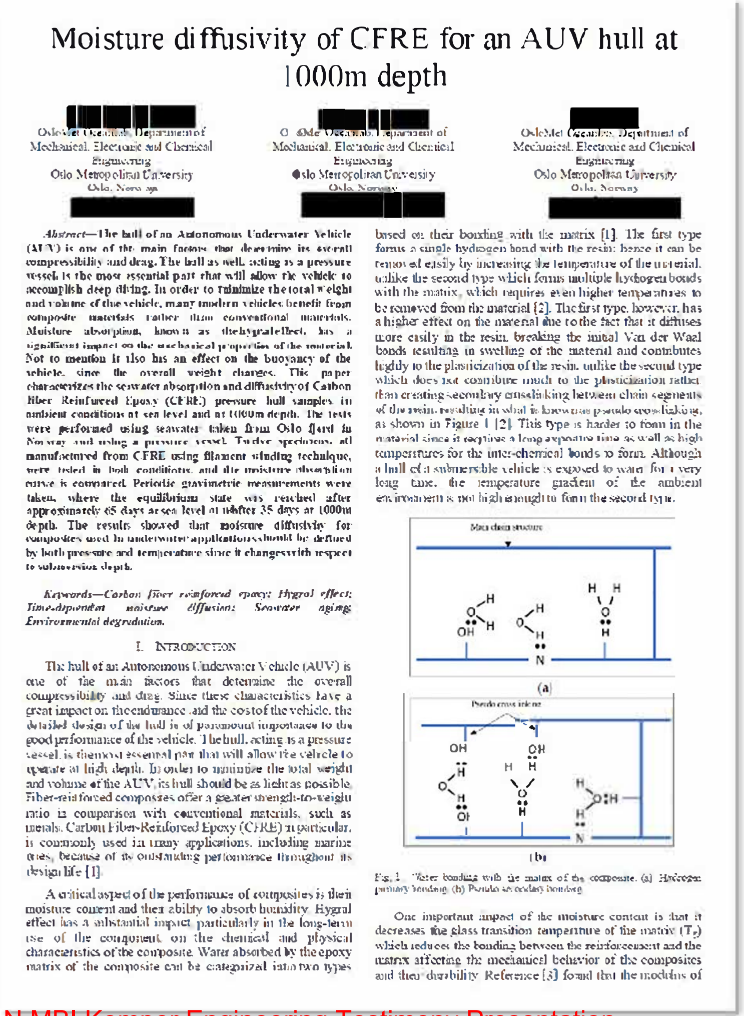
There are several measurement techniques available to measure residual stresses. They are often classified by their level of destructiveness and their penetration. In order to compare the different measurement techniques an elastic-plastic bent beam sample has been chosen as it is very comparable to the residual stress field induced during the sheet bending process used in the submarine structure. Four bent beams have been measured using five different techniques: Incremental centre hole drilling, ring core, neutron diffraction, slitting and deep hole drilling technique. The results from measurement techniques show an excellent agreement when compared with the FEA.

In order to measure a full scale Rubis class submarine hull a limited number of techniques can be used, as the technique needs to be portable. The Deep Hole Drilling (DHD) technique was chosen because the neutron diffraction would require extracting a small test sample of about 400mm x 400mm, hence redistributing the residual stresses that were intended to be measured. Six measurements were carried out at different angular positions to detect variability in manufacture on a Rubis class submarine and a probabilistic calculation was done using all six DHD measurements. The Rubis class measurement results are also compared with two other submarine types, found in the literature.

Understanding the three-dimensional behaviour of residual stress in this type of structure provides a valuable resource to the numerical modelling community. The results can also support fatigue and fracture experimental work and may help increasing the operating life of 28 year old French nuclear submarine.

INTRODUCTION

Efforts have been made to establish and improve ultimate submarine design life limits. The residual stresses induced by manufactures, like rolling and welding, are variables required for analysis of various degradation mechanisms associated with the aggressive conditions experienced by a submersible (buckling, fatigue, stress corrosion cracking). In a submarine structure, the hull is essentially composed of rolled plates and T-



Moisture diffusivity of CFRE for an AUV hull at 1000m depth

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Abstract—The hull of an Autonomous Underwater Vehicle (AUV) is one of the main factors that determine its overall compressibility and drag. The hull as well, acting as a pressure vessel, is the most essential part that will allow the vehicle to accomplish deep diving. In order to minimize the total weight and volume of the vehicle, many modern vehicles benefit from composite materials rather than conventional materials. Moisture absorption, known as the hygral effect, has a significant impact on the mechanical properties of the material. Not to mention it also has an effect on the buoyancy of the vehicle, since the overall weight changes. This paper characterizes the seawater absorption and diffusivity of Carbon fiber Reinforced Epoxy (CFRE) pressure hull samples in ambient conditions at sea level and at 1000m depth. The tests were performed using seawater taken from Oslo fjord in Norway and using a pressure vessel. Twelve specimens, all manufactured from CFRE using filament winding technique, were tested in both conditions, and the moisture absorption curve is compared. Periodic gravimetric measurements were taken, where the equilibrium state was reached after approximately 65 days at sea level or after 35 days at 1000m depth. The results showed that moisture diffusivity for composites used in underwater applications should be defined by both pressure and temperature since it changes with respect to submersion depth.

Keywords—Carbon fiber reinforced epoxy; Hygral effect; Time-dependent moisture diffusion; Seawater aging; Environmental degradation.

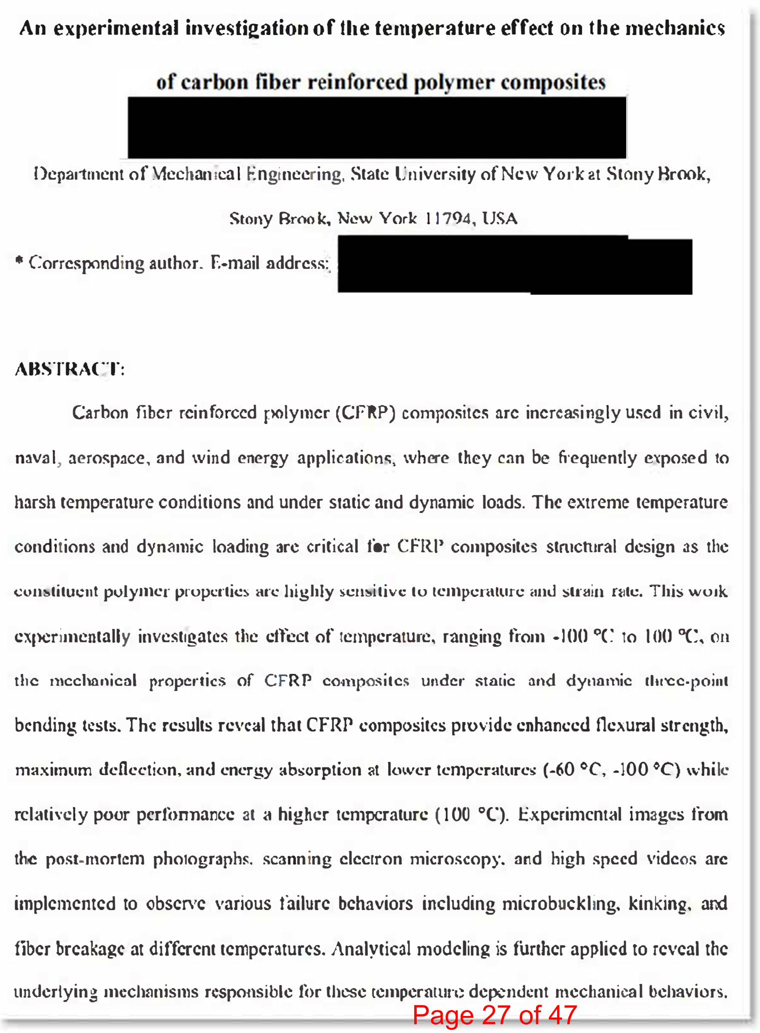
I. INTRODUCTION

The hull of an Autonomous Underwater Vehicle (AUV) is one of the main factors that determine the overall compressibility and drag. Since these characteristics have a great impact on the endurance and the cost of the vehicle, the detailed design of the hull is of paramount importance to the good performance of the vehicle. The hull, acting as a pressure vessel, is the most essential part that will allow the vehicle to operate at high depth. In order to minimize the total weight and volume of the AUV, its hull should be as light as possible. Fiber-reinforced composites offer a greater strength-to-weight ratio in comparison with conventional materials, such as metals. Carbon Fiber-Reinforced Epoxy (CFRE) in particular, is commonly used in many applications, including marine ones, because of its outstanding performance throughout its design life [1].

A critical aspect of the performance of composites is their moisture content and their ability to absorb humidity. Hygral effect has a substantial impact particularly in the long-term use of the component on the chemical and physical characteristics of the composite. Water absorbed by the epoxy matrix of the composite can be categorized into two types based on their bonding with the matrix [1]. The first type forms a single hydrogen bond with the resin; hence it can be removed easily by increasing the temperature of the material, unlike the second type which forms multiple hydrogen bonds with the matrix, which requires even higher temperatures to be removed from the material [2]. The first type, however, has a higher effect on the material due to the fact that it diffuses more easily in the resin, breaking the initial Van der Waal bonds resulting in swelling of the material and contributes highly to the plasticization of the resin, unlike the second type which does not contribute much to the plasticization rather than creating secondary crosslinking between chain segments of the resin, resulting in what is known as pseudo cross-linking, as shown in Figure 1 [2]. This type is harder to form in the material since it requires a long exposure time as well as high temperatures for the inter-chemical bonds to form. Although a hull of a submersible vehicle is exposed to water for a very long time, the temperature gradient of the ambient environment is not high enough to form the second type.

Fig. 1. Water bonding with the matrix of the composite. (a) Hydrogen primary bonding (b) Pseudo secondary bonding

One important impact of the moisture content is that it decreases the glass transition temperature of the matrix (T_g) which reduces the bonding between the reinforcement and the matrix affecting the mechanical behavior of the composites and their durability. Reference [3] found that the modulus of



An experimental investigation of the temperature effect on the mechanics of carbon fiber reinforced polymer composites

[redacted]

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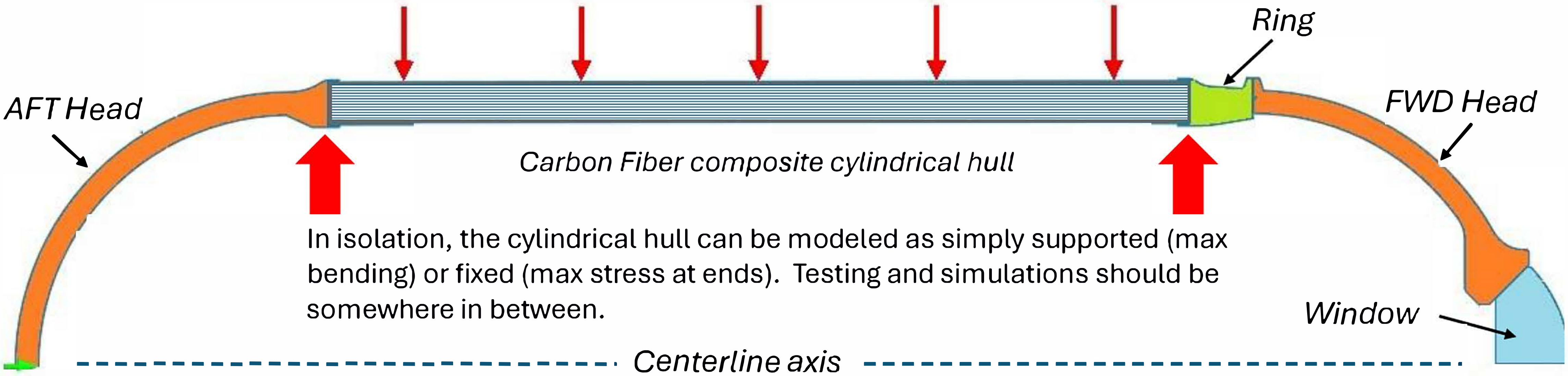
* Corresponding author. E-mail address: [redacted]

ABSTRACT:

Carbon fiber reinforced polymer (CFRP) composites are increasingly used in civil, naval, aerospace, and wind energy applications, where they can be frequently exposed to harsh temperature conditions and under static and dynamic loads. The extreme temperature conditions and dynamic loading are critical for CFRP composites structural design as the constituent polymer properties are highly sensitive to temperature and strain rate. This work experimentally investigates the effect of temperature, ranging from -100 °C to 100 °C, on the mechanical properties of CFRP composites under static and dynamic three-point bending tests. The results reveal that CFRP composites provide enhanced flexural strength, maximum deflection, and energy absorption at lower temperatures (-60 °C, -100 °C) while relatively poor performance at a higher temperature (100 °C). Experimental images from the post-mortem photographs, scanning electron microscopy, and high speed videos are implemented to observe various failure behaviors including microbuckling, kinking, and fiber breakage at different temperatures. Analytical modeling is further applied to reveal the underlying mechanisms responsible for these temperature dependent mechanical behaviors.

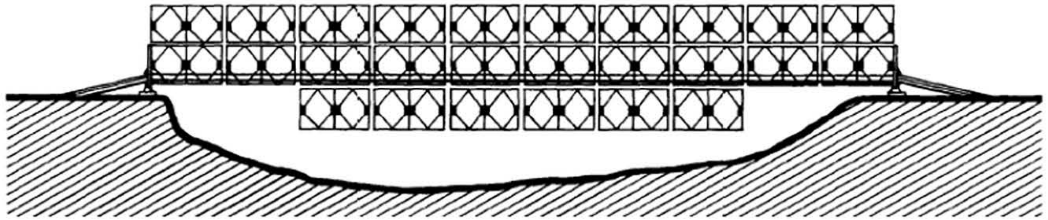
Carbon Fiber – Load bearing members

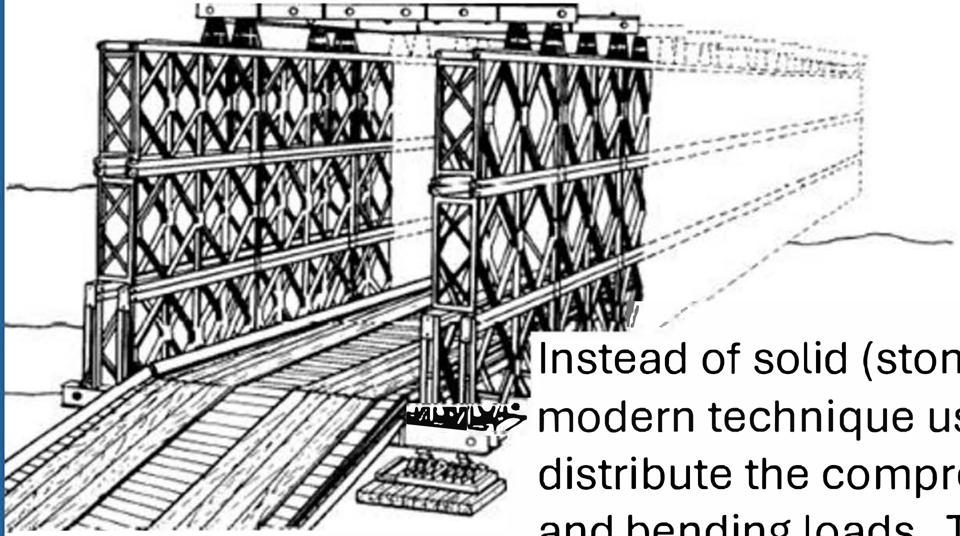


In isolation, the cylindrical hull can be modeled as simply supported (max bending) or fixed (max stress at ends). Testing and simulations should be somewhere in between.

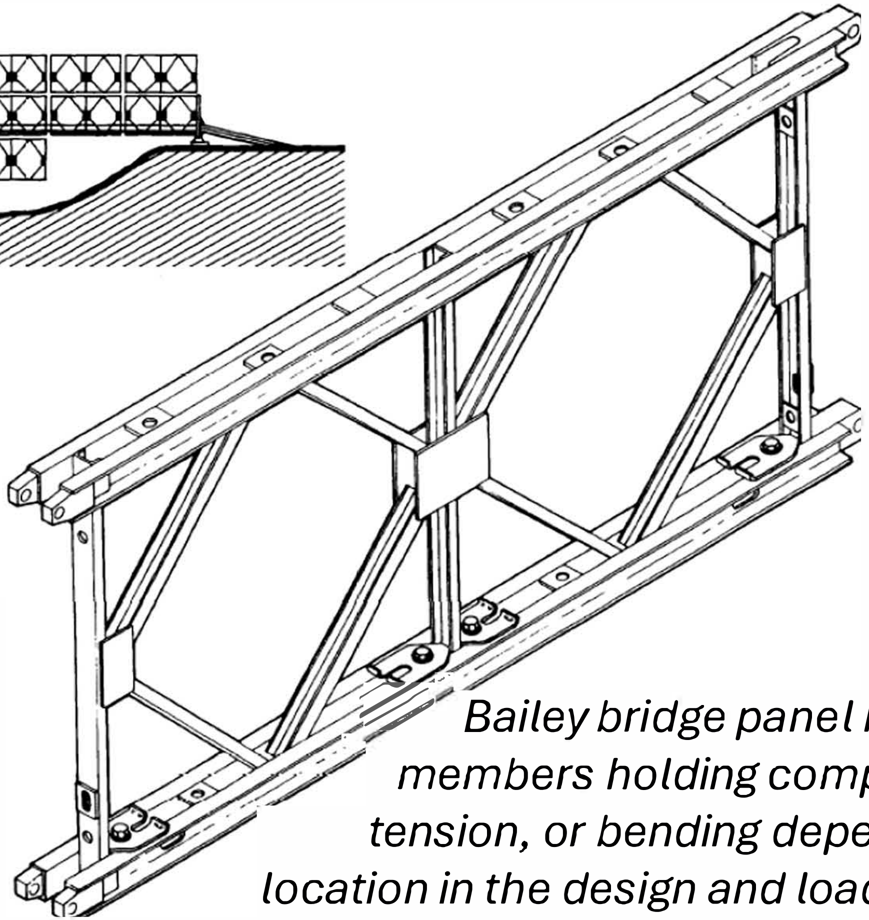
Carbon fiber is an emerging technology with respect to external pressure applications. As a code, it is covered in Section X, ASME Boiler and Pressure Vessel Code. External pressure is limited to vacuum (100 kPA, 15 psig). Outside of code limits, it requires detailed engineering and testing (ie, VVUQ).

Carbon Fiber – Load bearing members





Instead of solid (stone) bridges, a modern technique uses trusses to distribute the compressive, tensile, and bending loads. Their working volume is mostly air. Individual members can fail without dropping the bridge.



Bailey bridge panel has truss members holding compression, tension, or bending depending on location in the design and load applied

Carbon Fiber – Load bearing members

Single panel wide, Single panel high

*Double panel wide,
Single panel high*

*Double panel wide,
Double panel high*

How many panels thick and high depends on the intended load and span length. It's not a question of the strength of the panel, **it's the design for the application.**

Carbon Fiber – Load bearing members

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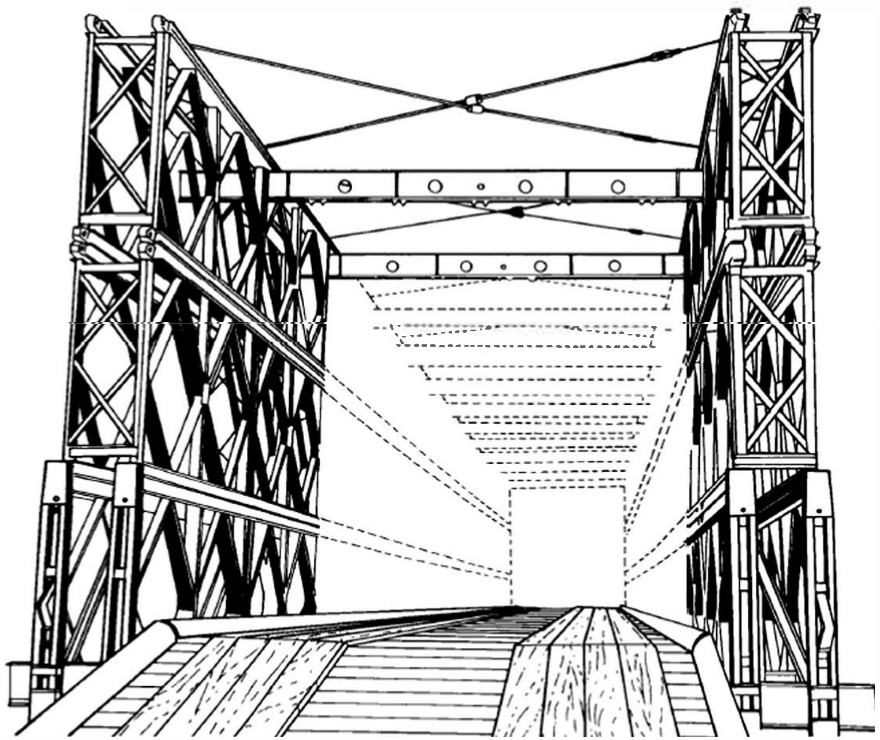
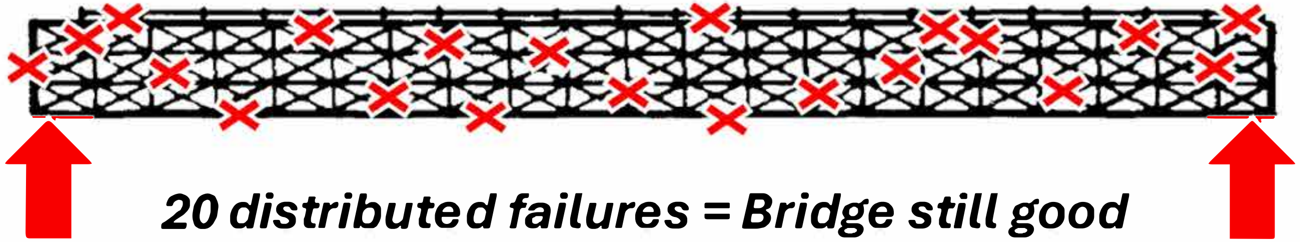


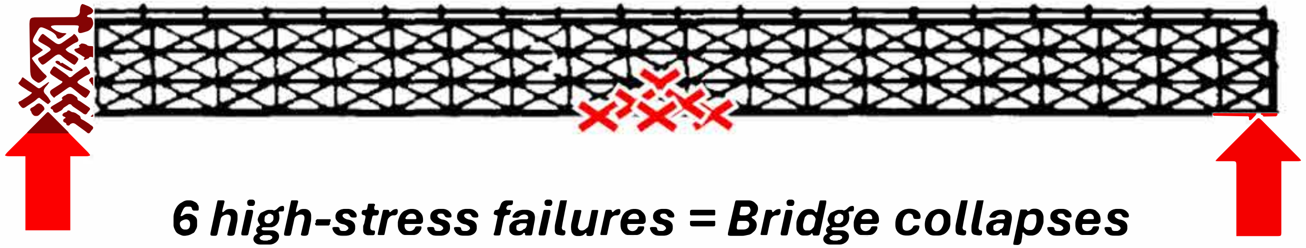
Random distribution for failed members



20 distributed failures = Bridge still good

Shear Failure

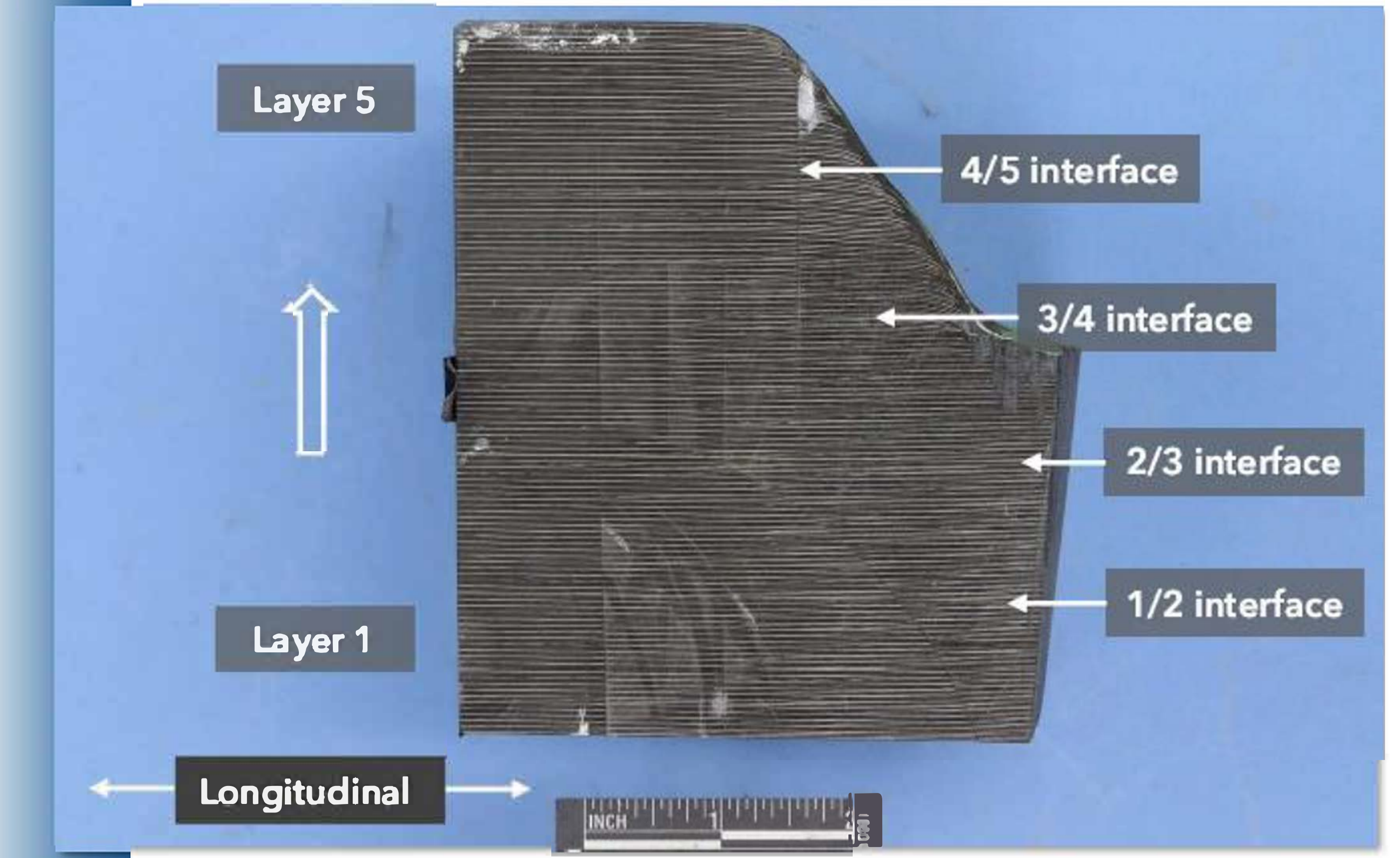
Bending Failure



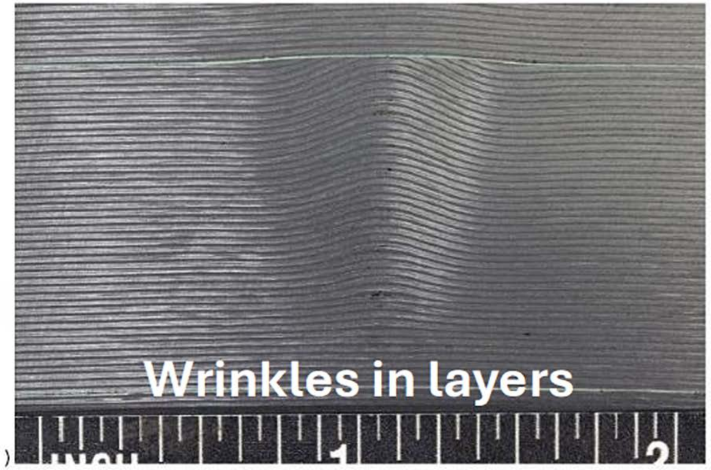
6 high-stress failures = Bridge collapses

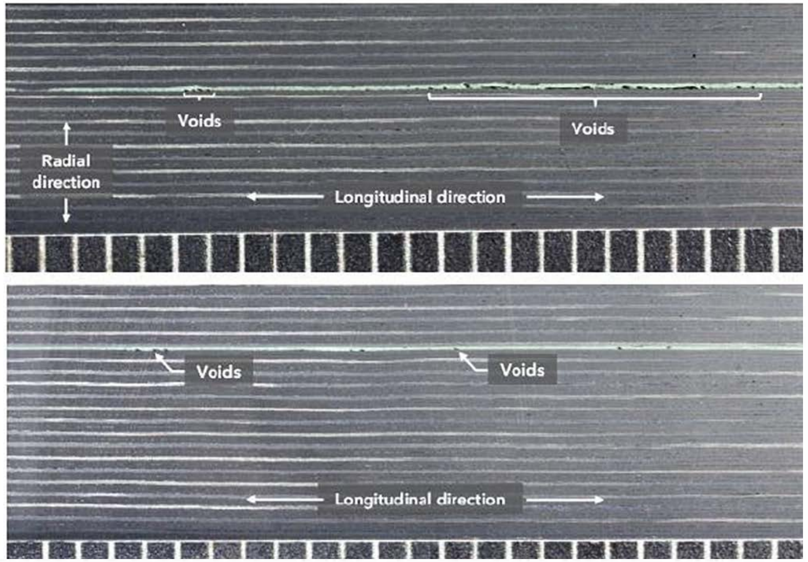
In order to count the number of “breaks” to predict failure, ***where*** the breaks will occur under load and ***how many*** are needed to create failure must be determined. Usage tends to bias “breaks” to locations consistent with structural failure modes.

Carbon Fiber design, issues (see NTSB)



Cross section of the full thickness



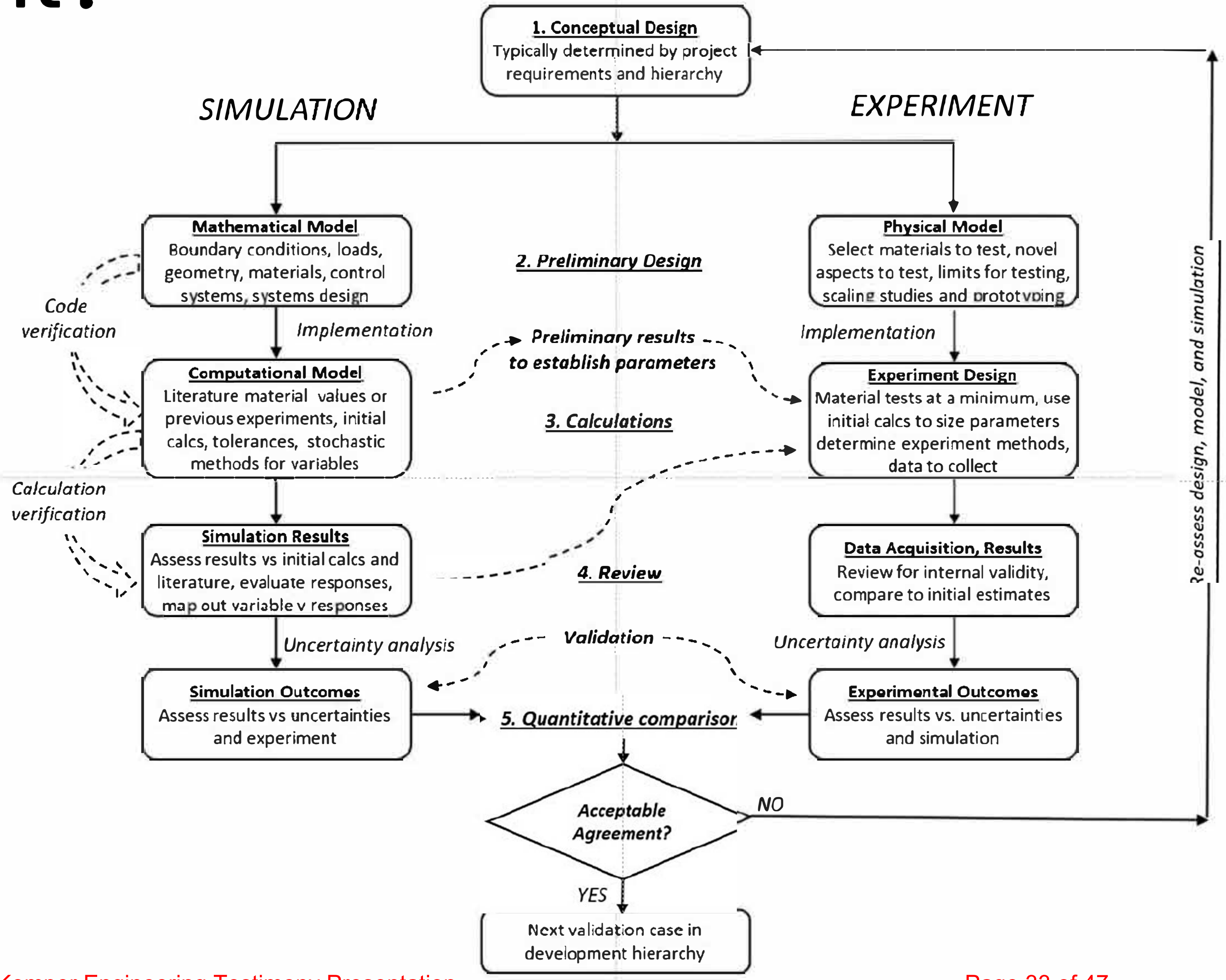


VVUQ: What is it?

Verification, Validation, and Uncertainty Quantification (VVUQ) is a systematic approach applied to any industry engaged in **innovation with significant risk to people** or financial risk in order to **reduce and manage risk in a transparent, accountable manner**.

A significant aspect is **documenting requirements, predictions, measured results, and methods** in order to maximize the potential to be informed by the process while doing what no one has done before. **No "cookbook."**

Example: **FDA requires VVUQ** for software in medical devices as well if simulations are used in design.



1. Conceptual Design
Typically determined by project requirements and hierarchy

SIMULATION

EXPERIMENT

Mathematical Model
Boundary conditions, loads, geometry, materials, control systems, systems design

2. Preliminary Design

Physical Model
Select materials to test, novel aspects to test, limits for testing, scaling studies and prototyping

Code verification

Implementation

Preliminary results to establish parameters

Implementation

Computational Model
Literature material values or previous experiments, initial calcs, tolerances, stochastic methods for variables

3. Calculations

Experiment Design
Material tests at a minimum, use initial calcs to size parameters determine experiment methods, data to collect

Calculation verification

Simulation Results
Assess results vs initial calcs and literature, evaluate responses, map out variable v responses

4. Review

Data Acquisition, Results
Review for internal validity, compare to initial estimates

Uncertainty analysis

Validation

Uncertainty analysis

Simulation Outcomes
Assess results vs uncertainties and experiment

5. Quantitative comparison

Experimental Outcomes
Assess results vs. uncertainties and simulation

Acceptable Agreement?

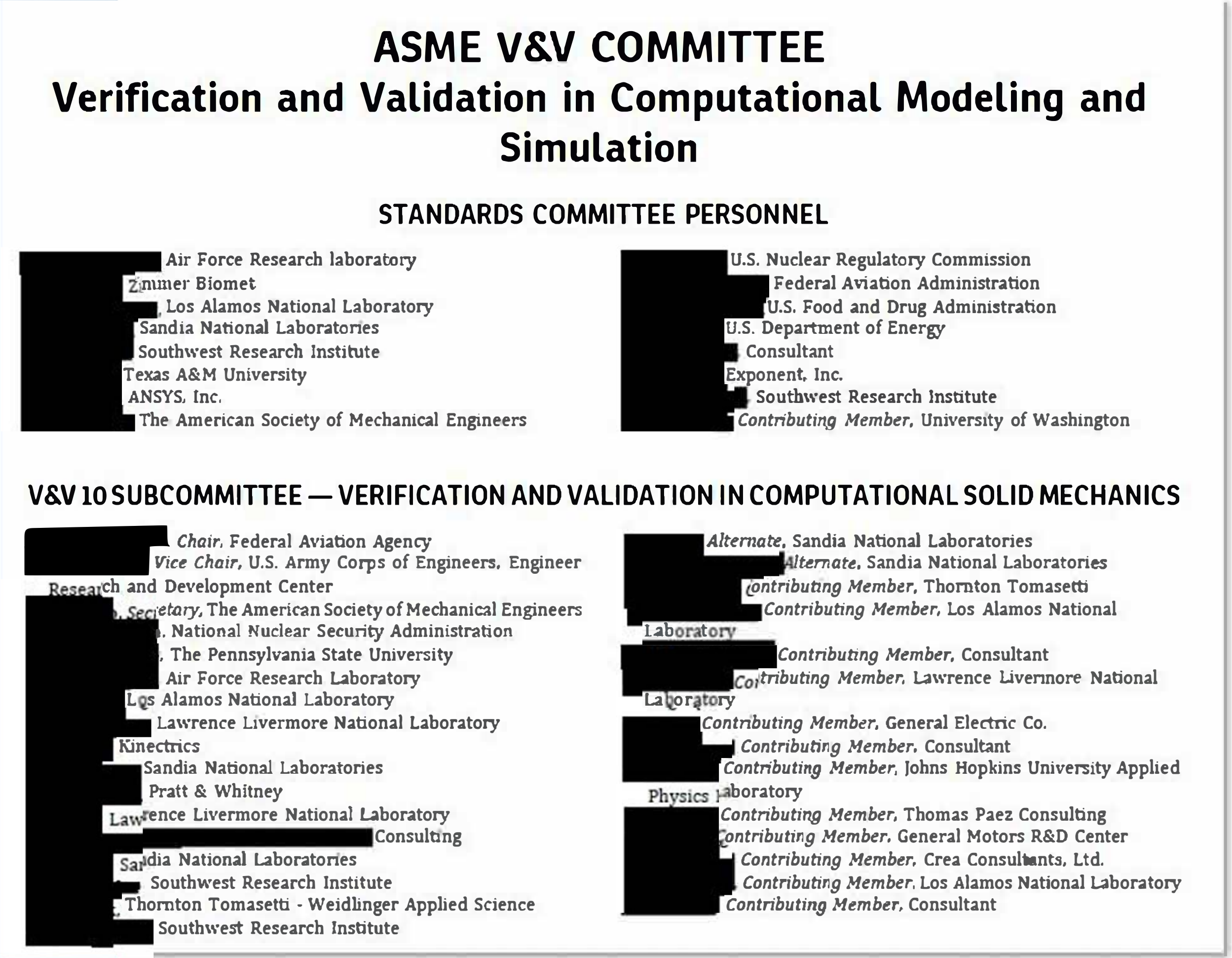
NO

Re-assess design, model, and simulation

YES

Next validation case in development hierarchy

ASME VVUQ Codes & Stds



ASME V&V COMMITTEE
Verification and Validation in Computational Modeling and Simulation

STANDARDS COMMITTEE PERSONNEL

[redacted] Air Force Research laboratory
[redacted] Zimmer Biomet
[redacted], Los Alamos National Laboratory
[redacted] Sandia National Laboratories
[redacted] Southwest Research Institute
[redacted] Texas A&M University
[redacted] ANSYS, Inc.
[redacted] The American Society of Mechanical Engineers

[redacted] U.S. Nuclear Regulatory Commission
[redacted] Federal Aviation Administration
[redacted] U.S. Food and Drug Administration
[redacted] U.S. Department of Energy
[redacted] Consultant
[redacted] Exponent, Inc.
[redacted] Southwest Research Institute
[redacted] *Contributing Member*, University of Washington

V&V 10 SUBCOMMITTEE — VERIFICATION AND VALIDATION IN COMPUTATIONAL SOLID MECHANICS

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[redacted] *Vice Chair*, U.S. Army Corps of Engineers, Engineer Research and Development Center
[redacted], *Secretary*, The American Society of Mechanical Engineers
[redacted], National Nuclear Security Administration
[redacted], The Pennsylvania State University
[redacted] Air Force Research Laboratory
[redacted] Los Alamos National Laboratory
[redacted] Lawrence Livermore National Laboratory
[redacted] Kinectrics
[redacted] Sandia National Laboratories
[redacted] Pratt & Whitney
[redacted] Lawrence Livermore National Laboratory
[redacted] Consulting
[redacted] Sandia National Laboratories
[redacted] Southwest Research Institute
[redacted] Thornton Tomasetti - Weidlinger Applied Science
[redacted] Southwest Research Institute

[redacted] *Alternate*, Sandia National Laboratories
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[redacted] *Contributing Member*, Thornton Tomasetti
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[redacted] *Contributing Member*, Los Alamos National Laboratory
[redacted] *Contributing Member*, Consultant

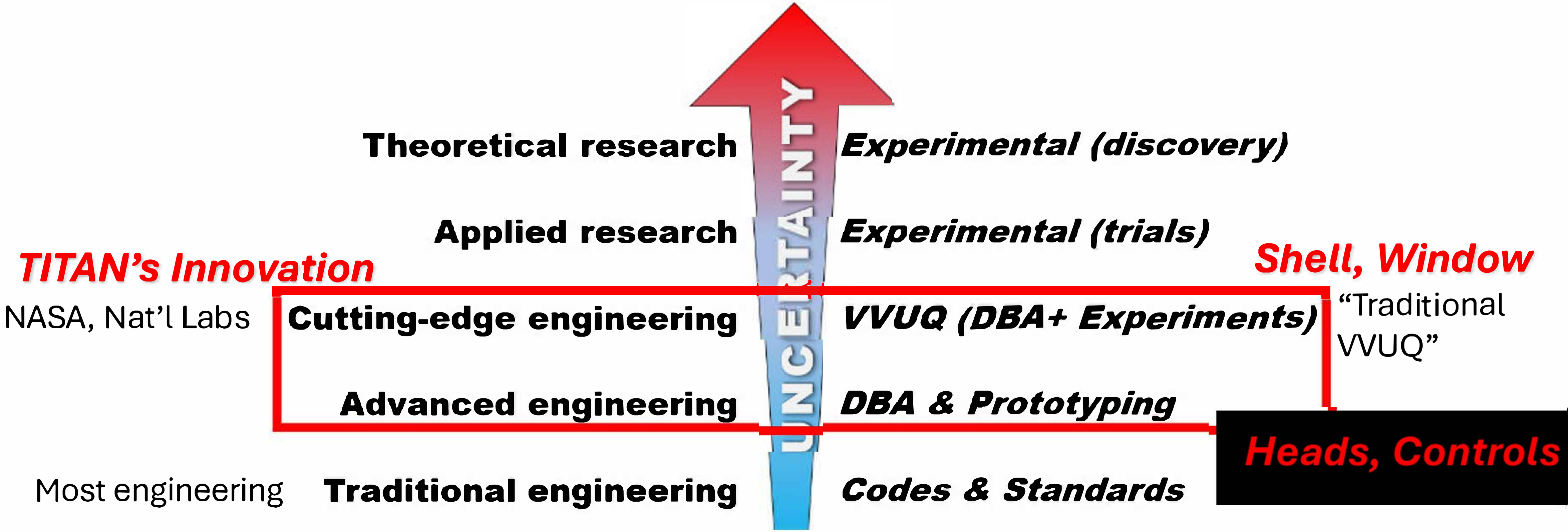
VVUQ started in the 1970's. The VVUQ standards began early 2000's. They are not "cookbooks," but peer reviewed guidance based on best practices to instruct users in developing their own VVUQ program.

ASME members: National laboratories, Biomedical, Aerospace, Power, Automotive, Defense – VVUQ is used in any industry.

VVUQ 10 – Computational Solid Mechanics
VVUQ 20– Comp. Fluid Dynamics & Heat Transfer
VVUQ 30 – Nuclear System Thermal Fluids Behavior
VVUQ 40 – Comp. Modeling of Medical Devices
VVUQ 50 – Advanced Manufacturing
VVUQ 60 – Energy Systems
VVUQ 70 – AI and Machine Learning
VVUQ 80 – Pharmaceutical Products *(new)*
VVUQ 90 – Airframe Structures *(new)*

Other VVUQ methods such as NASA, DoD, etc.

VVUQ Spectrum



Theoretical research | ***Experimental (discovery)***

Applied research | ***Experimental (trials)***

NASA, Nat'l Labs | **Cutting-edge engineering** | ***VVUQ (DBA+ Experiments)*** | "Traditional VVUQ"

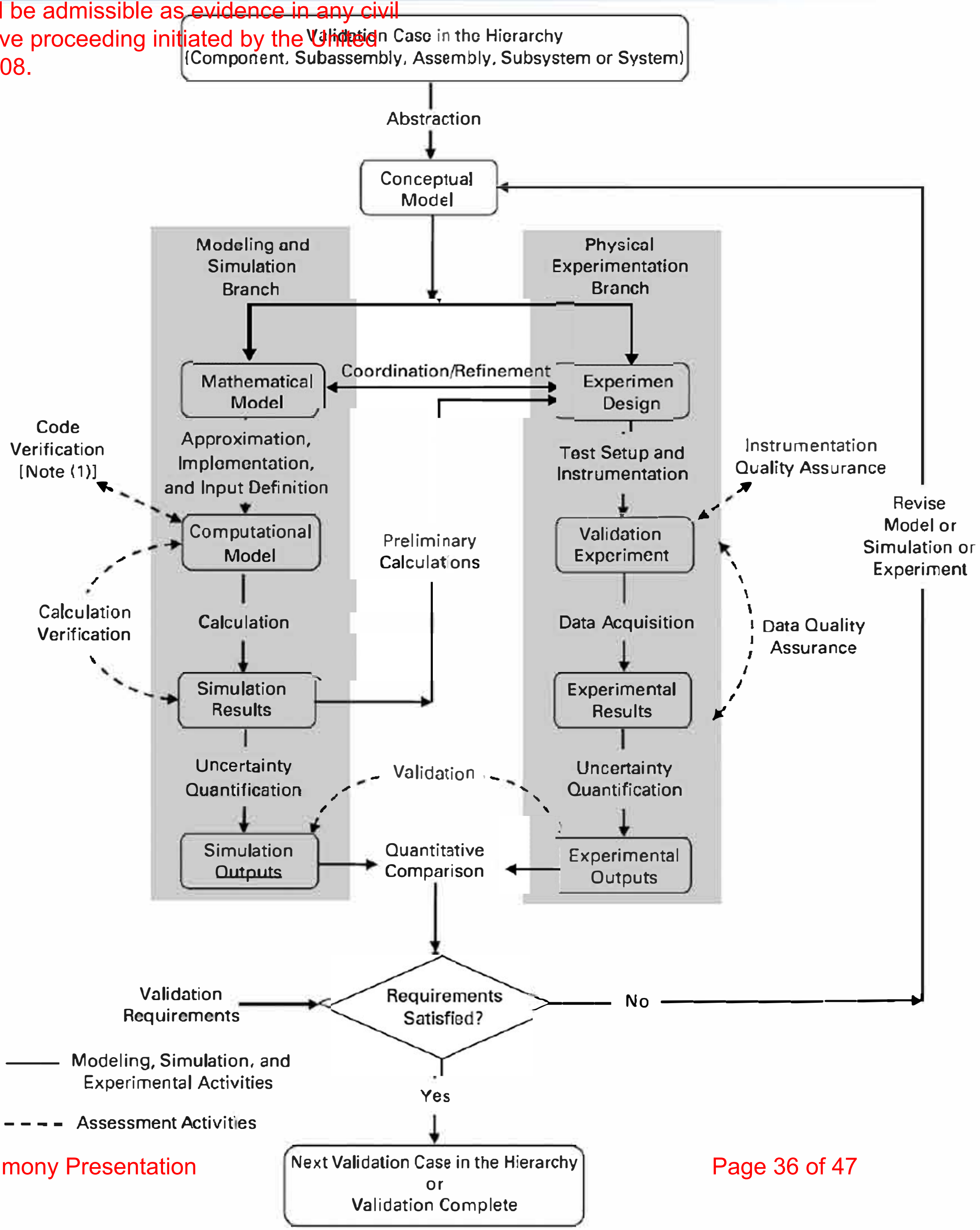
Advanced engineering | ***DBA & Prototyping***

Most engineering | **Traditional engineering** | ***Codes & Standards***

As a field becomes more established and codifed, the VVUQ is "baked in" the codes & standards.
"Cookbook engineering" has **design margins to reduce uncertainty and risk.**

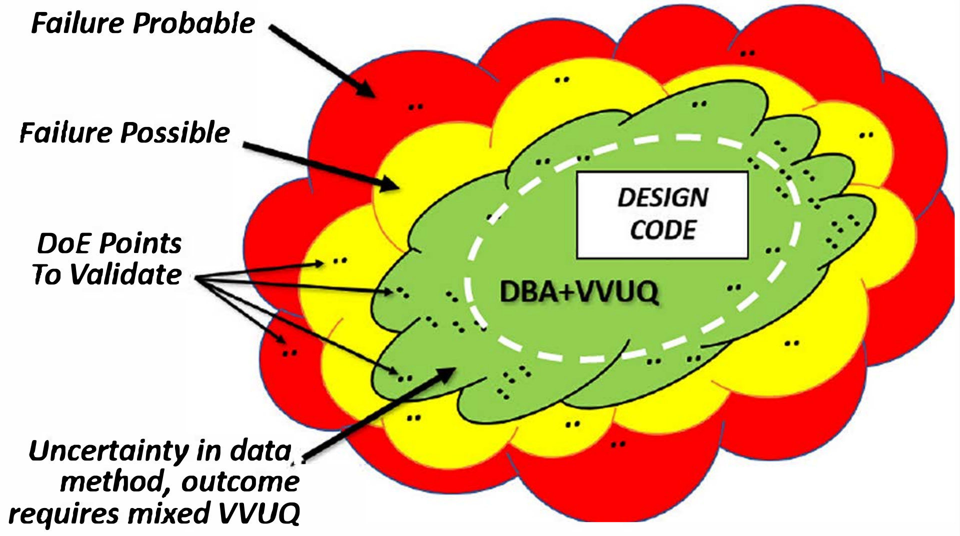
Traditional VVUQ

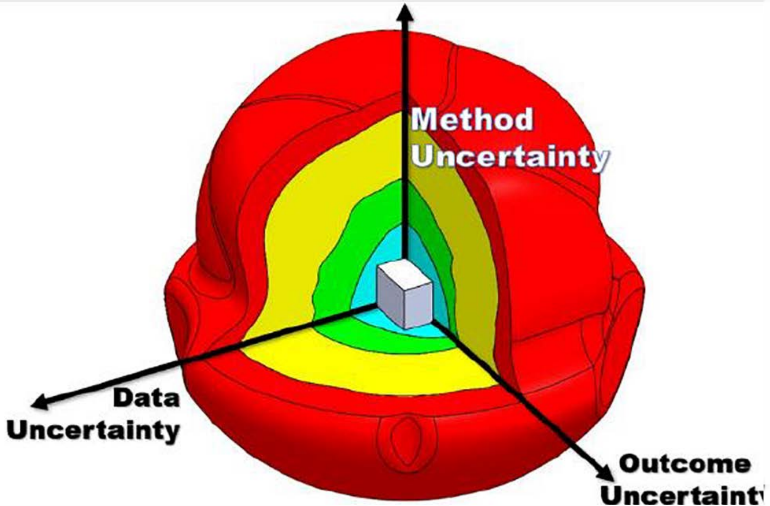
- Nondeterministic (multiple design factors over range of values)
- Requires methodical use of experiments, theory, and simulation
- ID's irreducible v reducible uncertainty
- **Intended to inform managers of risks in project to assess time/resources**
- Often for **high consequence** or **extraordinary novelty**
- Detailed, time consuming, and therefore **expensive**. Normal engineering is "cookbook" and much less expensive.



“Engineer”/Deterministic VVUQ

- **<u>No longer in the “cookbook” box.</u>**
- Still in a well-understood domain or the consequences are not “zero failure,” as supported by FMEA, so not “traditional VVUQ”
- Many codes and groups have “design by analysis”, including ASME and USCG
- Often solving for single set of design factors
- More VVUQ the less the code provides:
  - guidance for using computer models/sims
  - guidance on material data
  - guidance on load combination
  - guidance for failure criteria





VVUQ and the Pressure Vessel Code

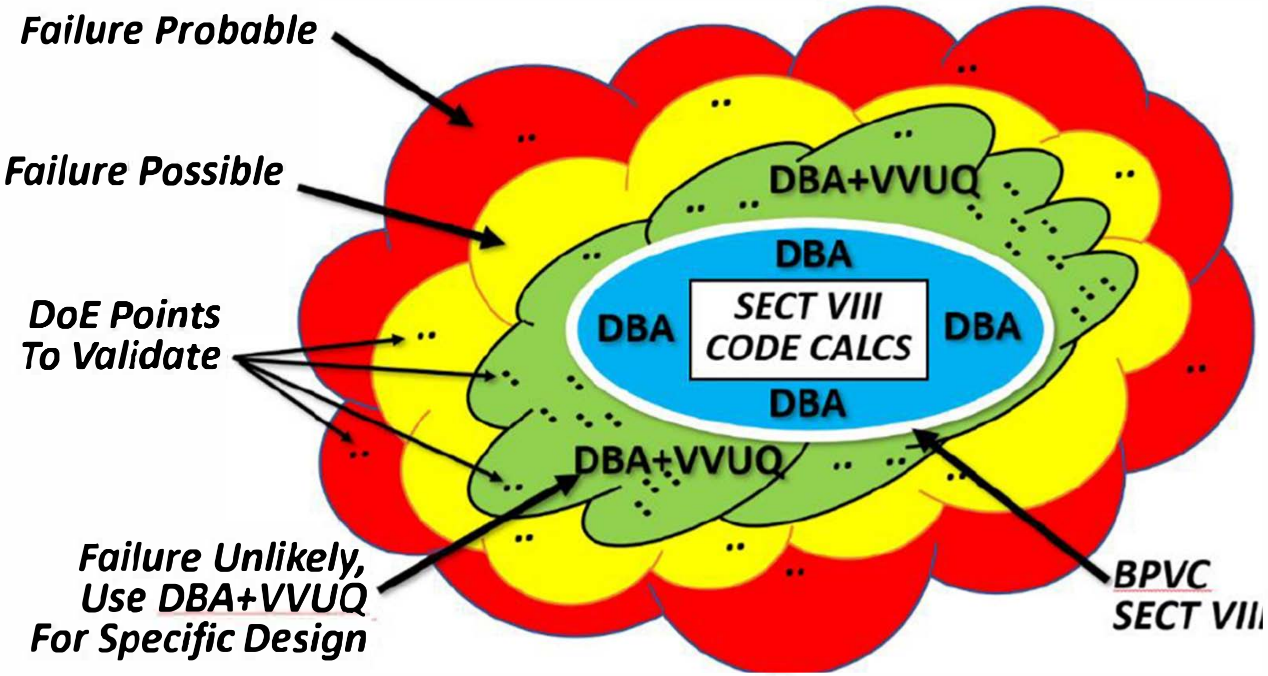
"Cookbook" design is well-defined <u>white box</u>

ASME BPVC Section VIII, Div. 2, Part 5, "*Design By Analysis*" (also API 579/ASME FFS1)

<u>**Complete system specified** (blue oval)</u>

- Types of FEM analyses permitted
- Material properties, curves at temp
- Service condition (corrosion, fatigue, etc.)
- Load combinations for different analyses
- NDT, system testing requirements

<u>**Within limits of Part 5**</u>, solution verification is still highly prudent and is good practice, not required. Outside the limits, VVUQ is needed.



VVUQ drives windows DBA method

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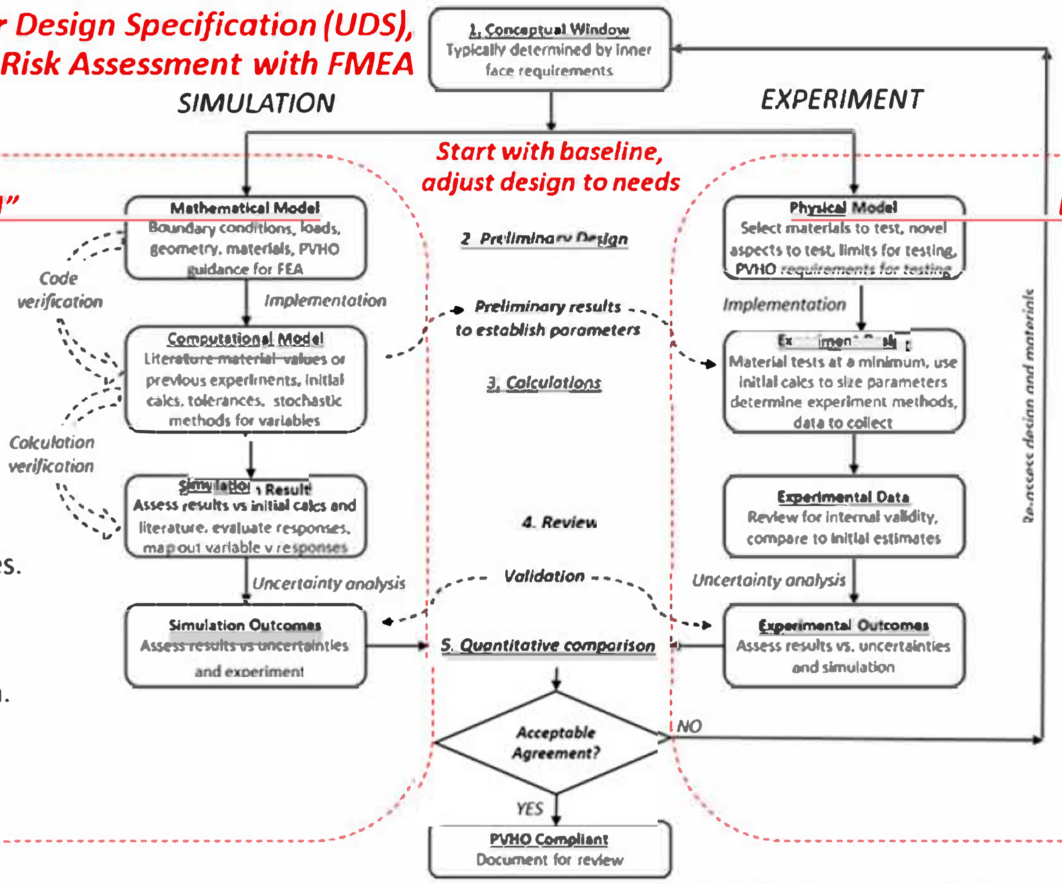
Design By Analysis Framework using PVHO-1, BPVC, V&V10

***START:* User Design Specification (UDS), Risk Assessment with FMEA**



SIMULATION → "Section VIII"

- Based on UDS, select methods, initial design, literature values for materials, conduct initial FMEA
- Analyze initial design, including implicit nonlinear Finite Element Analysis for stochastic range of variables. Diagnose critical areas.
- Update analysis using experimental material data. Inputs for Design of Experiment (DoE) for prototype.

EXPERIMENT → "Section II"

- Based on UDS, select initial materials and test methods. Establish QA and tolerances.
- Analyze materials stochastically for use in implicit nonlinear FEA. Develop DoE for prototype test
- Update testing based on analysis results, implement for key variables, locations, conditions.

***END:* Self contained, fully documented design with calculations, validation tests, and data. Suitable for Authorized Inspector review**

Why OceanGate needed VVUQ

- **High risk** to people and organization regarding first mode failures
- **New materials** (specific composition of carbon fiber composite)
- **Novel applications** (non-spherical external pressure shell, novel window)
- **Novel systems integration** of new items and "off the shelf" items

DESIGN → PREDICT → TEST → MEASURE → ANALYZE → REDESIGN ⮌

☑ Minimize novelty within design cycle to minimize VVUQ needed

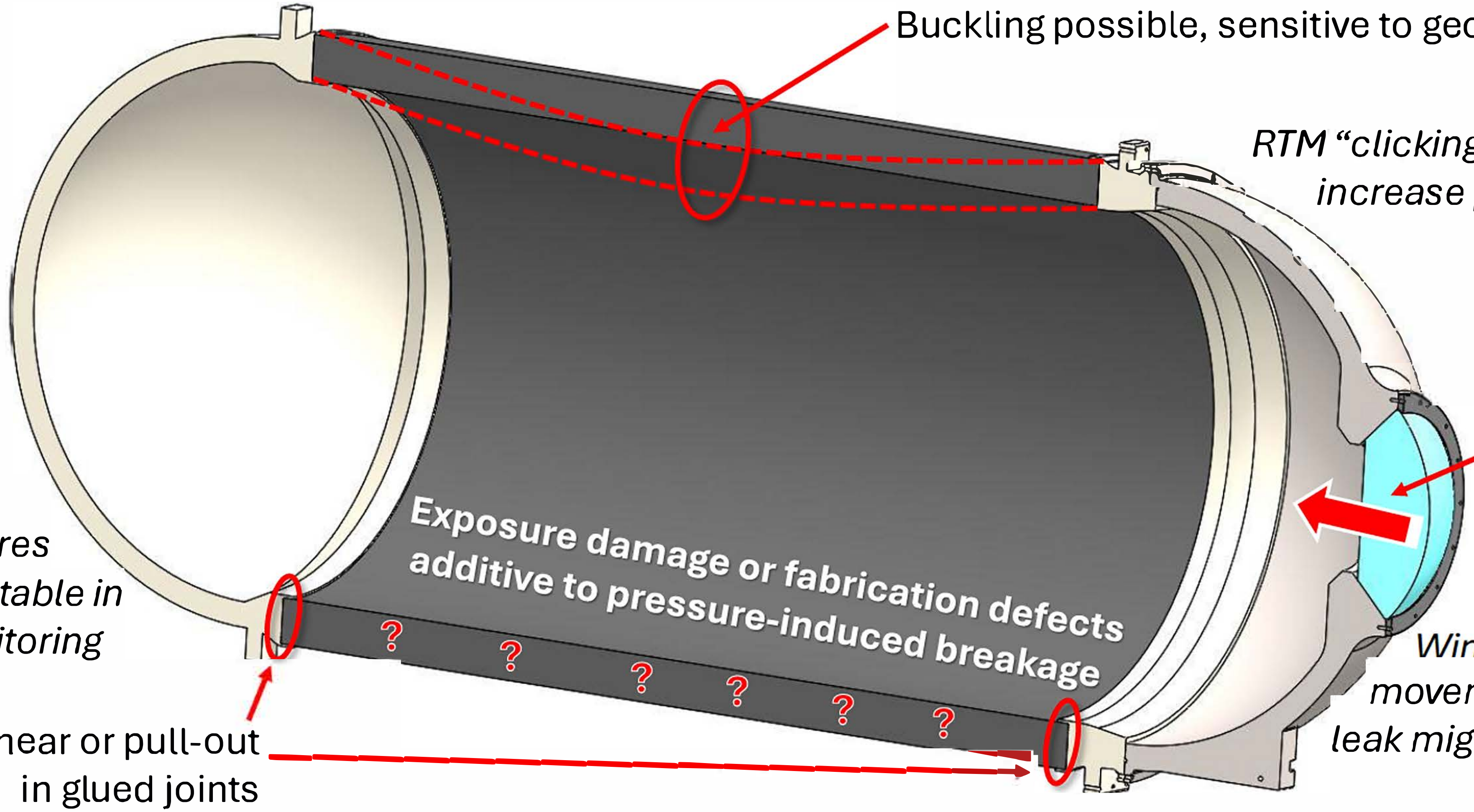
☑ Documentation to maximize value of simulations & tests

☑ VVUQ as "standard of care" for ethical engineering practice when codes and standards are not available to define what "what right looks like" from experience

Potential Implosion points

- ***Indeterminate*** at this time
- **Multiple unmitigated single-mode failure** paths, overlapping uncertainty based on data available precludes definitive causation
- ***Unknown design life***. It may have had a short life (no defect impact)
- Possible initiation cause(s):
    - Internal localized failure due to **cumulative carbon fiber breakage** (end of life without any shortcomings)
    - Internal localized failure of the composite hull **due to fabrication**
    - Internal localized failure due to **damage from exposure**
    - Shear failure along **glue line** between hull and ring
    - Failure of the **acrylic window**

Potential Implosion points



Bending in composite creates localized breakage (expected); ***Unknown design life***
Buckling possible, sensitive to geometry

RTM “clicking” rate likely to increase prior to failure

Window failure

Window excessive movement, crack, or leak might be detected

Exposure damage or fabrication defects additive to pressure-induced breakage

Glue failures not detectable in RTM monitoring

Shear or pull-out in glued joints

Kemper Engineering Recommendations

- Require **experimental craft** to be limited to an operator and assistant, with no passengers or cargo. Require inspections for suitability for the proposed usage, including tests.
- **Require min. level** of navigation aids, emergency gear, and a *standardized lifting point for extraction* via ROV attaching a cable
- **Require ASME PVHO-1** for all submarines made or operated in the US. Deviations allowed with justification (ie, “addressed by class.”)
- **Require a licensed Professional Engineer** to be part of the engineering team. They do not have to be the head engineer, but they provide code-required attestations such as the Manufacturer’s Data Report, review design calculations, etc.

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Testimony of
[redacted] P.E., DFE

Summary of preliminary findings regarding OceanGate and loss the Titan Submersible

Kemper Engineering Services
Baton Rouge, LA

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