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THESIS

TESTING AND ANALYSIS OF ADDITIVELY MANUFACTURED BUOYANT PARTS

by

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June 2023

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**TESTING AND ANALYSIS OF ADDITIVELY MANUFACTURED BUOYANT
PARTS**

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Submitted in partial fulfillment of the
requirements for the degree of

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ABSTRACT

The computational results of a non-linear buckling analysis and the experimental results for hydrostatic testing of hollow aluminum parts produced using liquid metal jet printing (LMJP) additive manufacturing (AM) are reported. Two variants of hollow metal parts were tested, a single enclosed piece meant to be attached externally to a payload and a two-piece design meant to house a payload. The designs were manufactured using LMJP and underwent hydrostatic testing to determine the viability of using LMJP to produce parts for the undersea warfare community. The designs underwent a non-linear buckling analysis to assess the adequacy of traditional structural engineering techniques for LMJP parts. The non-linear analysis-predicted areas of max stress matched the experimental areas of failure. The non-linear analysis accurately predicted the buckling shape. The predicted magnitude of failure was five to twenty times higher than experimentation. The suspected cause was that the LMJP process reduced the mechanical strength of the aluminum 4008 and introduced anisotropic properties that made it weaker than the standard aluminum alloy used in the finite element analysis. Measuring these properties and constructing a custom computational material to be used for analysis may eliminate the overestimation of failure. This work shows that current tools for structural analysis provide valid insight into the behavior of AM parts but that AM techniques introduce unique material challenges.

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List of Acronyms and Abbreviations

AM	additive manufacturing
ASTM	American Society for Testing and Materials
BJT	binder jetting
CAD	computer-aided design
COTS	commercial off-the-shelf
DAQ	data acquisition
DED	directed energy deposition
DLP	digital light processing
DOD	Department of Defense
FEA	finite element analysis
GUI	graphical user interface
LMJP	liquid metal jet printing
MEX	material extrusion
MJT	material jetting
NI	National Instruments
NPS	Naval Postgraduate School
NPT	National Pipe Thread
PBF	powder bed fusion
PSD	power spectral density

PT	pressure transducer
SHL	sheet lamination
SLD	surface layer duct
SLS	selective laser sintering
SL	stereolithography
SOP	standard operating procedure
SSP	sound speed profile
USN	U.S. Navy
UV	ultraviolet
UW	undersea warfare
VPP	vat photopolymerization

Nomenclature

α	Cone Angle from Centerline
δPT	Pressure Transducer Error
δQ	Total System Error
$\delta T B_{input}$	Block Input Error
$\delta T B_{noise}$	Block Noise Error
λ_i	Eigenvalues
ν	Poisson's Ratio
σ_{yield}	Yield Stress
A_i	Eigenvectors
A_o	Original Cross Sectional Area Under Tensile Stress
A_s	Stressed Cross Sectional Area
d_E	Diameter of Exposed Area
d_N	Nominal Diameter
d_p	Pitch Diameter
E_Y	Young's Modulus of Elasticity
F_{base}	Base Load
$F_{buckling}$	Buckling Load
$F_{failure}$	Force at Failure
$F_{pressure}$	Force Due to Applied Pressure
F_{ref}	Reference Load

F_{torq}	Force Due to Applied Torque
I	Lubrication Factor
K	Stiffness Matrix
k	Bolt Material and Size Constant
K_{σ}	Geometric Stiffness Matrix
L/D	Length to Depth Ratio
L_o	Stressed Length
Le	Engagement Length
n	Threads per Inch
p	Thread Pitch
P_{max}	Maximum Pressure
Re	Yield Strength
Rm	Ultimate Strength
T	Torque

CHAPTER 1:

Introduction

1.1 Motivation

The United States submarine force is a vital asset to national security, and its role in the U.S. military is rapidly expanding. With this expanding role comes new challenges and an opportunity to use exciting new technologies to solve them. This work explores one such technology and will show how additive manufacturing (AM) can expand the U.S. submarine force's capability.

1.1.1 Submarine Resilience

Submarine operations necessitate self-reliance. This coupled with complex machinery vital to survival and small, tight spaces make submarine onboard logistics extremely complex. As a result, almost all of the limited space onboard is used for repair parts [1]. AM can drastically increase the flexibility and efficiency of onboard storage by equipping the submarine with 3D printers and the raw materials required to produce parts, thus reducing the number of repair parts carried. Although AM technologies have not yet been vetted to consistently and reliably produce parts to U.S. Navy (USN) certification standards, this technology is ready for non-certified and consumable parts. This work will demonstrate AM use for the latter.

1.1.2 Sound Speed Profile

A submarine's primary sensory input underwater is acoustic and acoustic performance is heavily dependent on multiple environmental factors including ocean floor topography, sea life, and the speed of sound in water [2]. The speed of sound in water is influenced by three factors: temperature, salinity, and pressure (Fig. 1.1). Each of these factors is a function of depth and are combined to create a graph of the speed of sound vs. depth called the sound speed profile (SSP), shown in Fig. 1.2 [2].

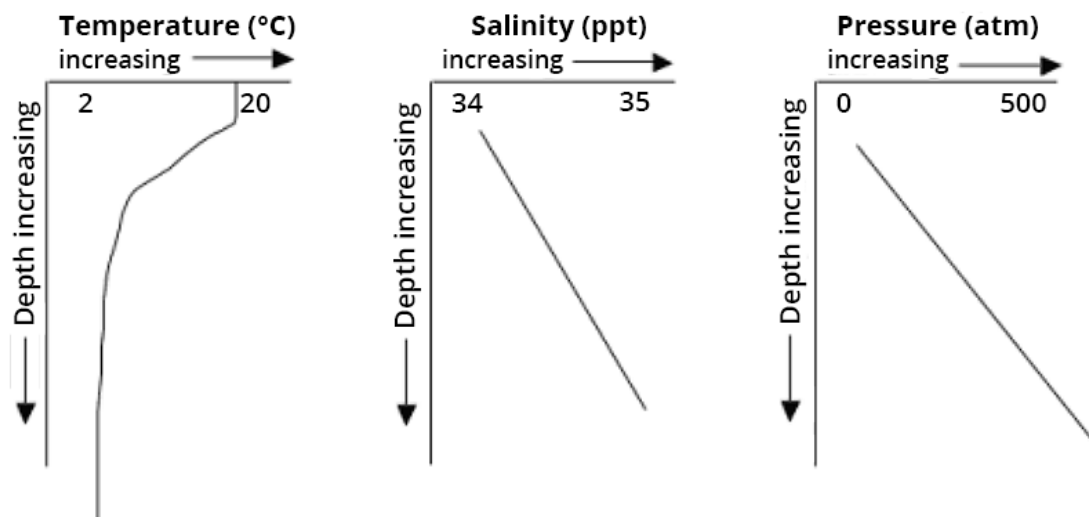


Figure 1.1. Effects of Temperature, Salinity, and Pressure on SSP.
Source: [3].

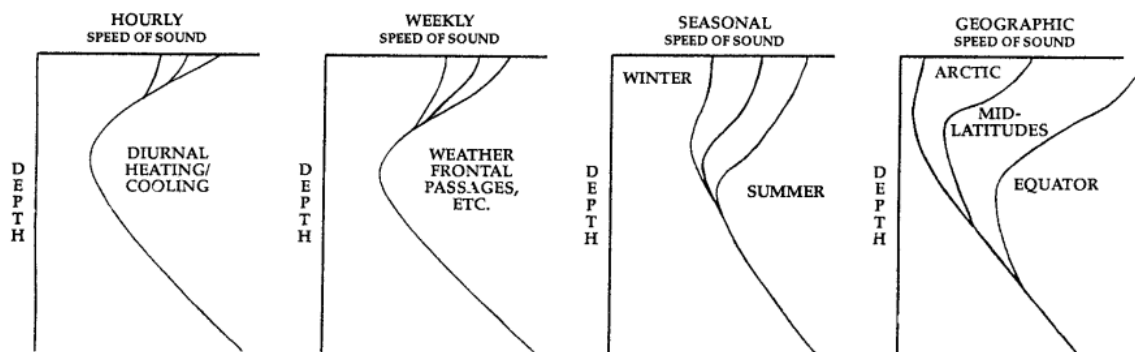


Figure 1.2. Examples of the SSP. Source: [2, Ch 7].

The SSP observed through the lens of Snell's law reveals several acoustic phenomena that make a submarine's ability to detect other vessels more difficult. One such phenomenon is the surface layer duct (SLD), shown in Fig. 1.3.

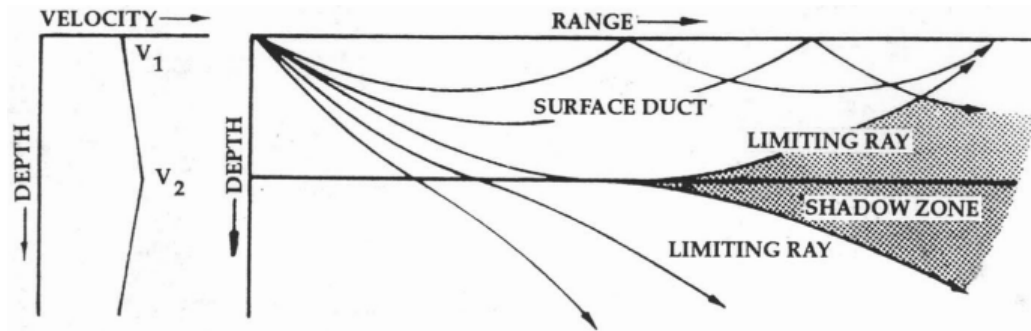


Figure 1.3. Surface Duct Propagation Path with Limiting Rays and Shadow Zone. Source: [2, Ch 7].

The SLD is formed when sound emitted from a source exceeds some angular threshold away from the ocean floor, causing it to be redirected toward the surface [2]. This sound reflects off the surface downward and is redirected back up towards the surface again. This traps the sound in the SLD until its energy dissipates. This phenomenon produces an area below the SLD called a shadow zone [2]. A submarine operating in the shadow zone of another vessel is at greater risk for collision because most or all sound emitted by the other vessel could be directed away from where the submarine is operating. The dangers presented by the shadow zone and other acoustic anomalies mandate the need for a device that could be deployed from a submarine to aid in safe navigation through these phenomenon. Since submarines can operate for months without a port call, and the conditions of their operating environments can vary drastically, a submarine would have to carry an impractical number of these devices. This creates an obstacle easily bound by AM.

1.2 Problem Statement

Liquid metal AM is a relatively new technology with potential that has not yet been fully realized. Liquid metal droplet printing can produce metal structures that cannot be

manufactured using traditional (powder-based) metal additive manufacturing. Specifically, liquid metal droplet printing can produce hollow metal parts without internal support structures or leaving unused build material inside. This research focuses on a non-linear buckling analysis of zero-post-processing designs for buoyant parts and the associated experimental results. This research aims to examine the suitability of AM buoyant parts to carry a payload for the undersea warfare (UW) community. Specifically, the objective of this research is to compare the results of a non-linear buckling analysis with experimental results in order to determine the viability of using traditional structural analysis methods for developing AM buoyant parts.

CHAPTER 2: Contextual Review

2.1 Additive Manufacturing

The concept of creating complex geometries by some method that uses only the required amount of material dates back to the 1960s. The first attempts used photopolymers cured by lasers in a process called stereolithography (SL) [4]. SL became the first method of AM commercially available in 1987 from 3D Systems, but metal based AM was not achieved until the early 1990s [4].

American Society for Testing and Materials (ASTM) Committee F42 generalizes AM into seven principle methods: material extrusion (MEX), vat photopolymerization (VPP), powder bed fusion (PBF), binder jetting (BJT), material jetting (MJT), directed energy deposition (DED), and sheet lamination (SHL) [5]. liquid metal jet printing (LMJP) is a new type AM developed in 2017 but was made commercially available in 2020 [6].

2.1.1 Material Extrusion

MEX based AM uses pressure to extrude bulk material in a semi-solid state to create layers of a part [7]. The bulk material is either a semi-solid that is made solid during post processing, or it is a solid that is made semi-solid by heating prior to extrusion. As long as the extrusion rate and speed of the extruder are constant (together, called the print speed), the dimensions of the extrusion will be constant. This allows uniform layers to be produced in the (X, Y) plane. At the completion of each layer, the extruder is indexed upward or the part indexed downward in the Z direction such that the next layer is extruded on top of the previous layer. The print speed and temperature are controlled such that consecutive layers fully bond together but the part does not deform under its own weight [7]. MEX is inexpensive and widely available making it the most prevalent form of AM for hobbyists and enthusiasts. One disadvantage of MEX is its inverse relationship between detail/quality and speed, a characteristic not present in some other AM processes. Another potential disadvantage is that part strength is anisotropic because of imperfect sintering between

layers. This characteristic can be made less severe with post-processing [7]. Figure 2.1 shows examples of MEX processes.

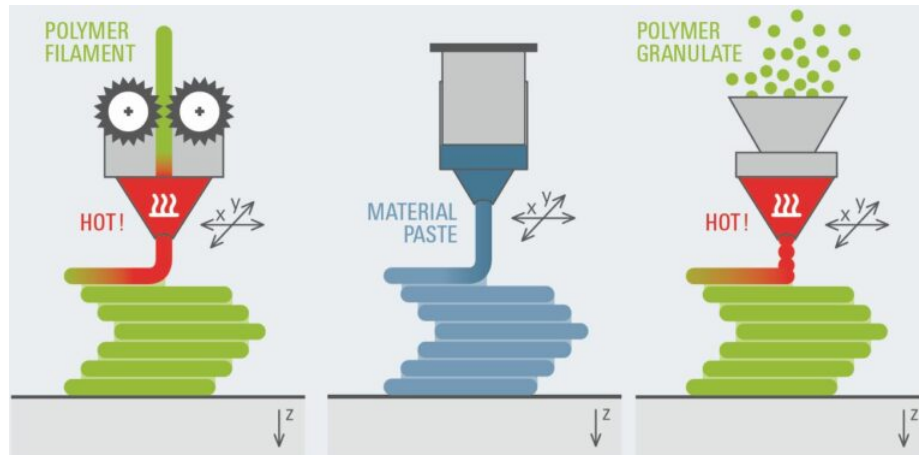


Figure 2.1. Schematic of Filament (left), Paste (center), and Pellets (right) MEX. Source: [4].

2.1.2 Vat Photopolymerization

Photopolymerization processes make use of radiation-curable resins, or photopolymers, in a low-viscosity fluid as their primary materials [7]. Photopolymers react to ultraviolet (UV) radiation, but some visible light systems exist. Once irradiated, these materials solidify by a chemical reaction called photopolymerization. VPP refers to a process that combines a vat of photopolymers and SL to form a part [7]. VPP has several advantages including high levels of detail, fast print speeds, and relatively large build areas. VPP disadvantages include higher initial costs due to required post processing equipment, long post-processing time (wash and cure), and material limitations [7]. Figure 2.2 shows two generic VPP process examples.

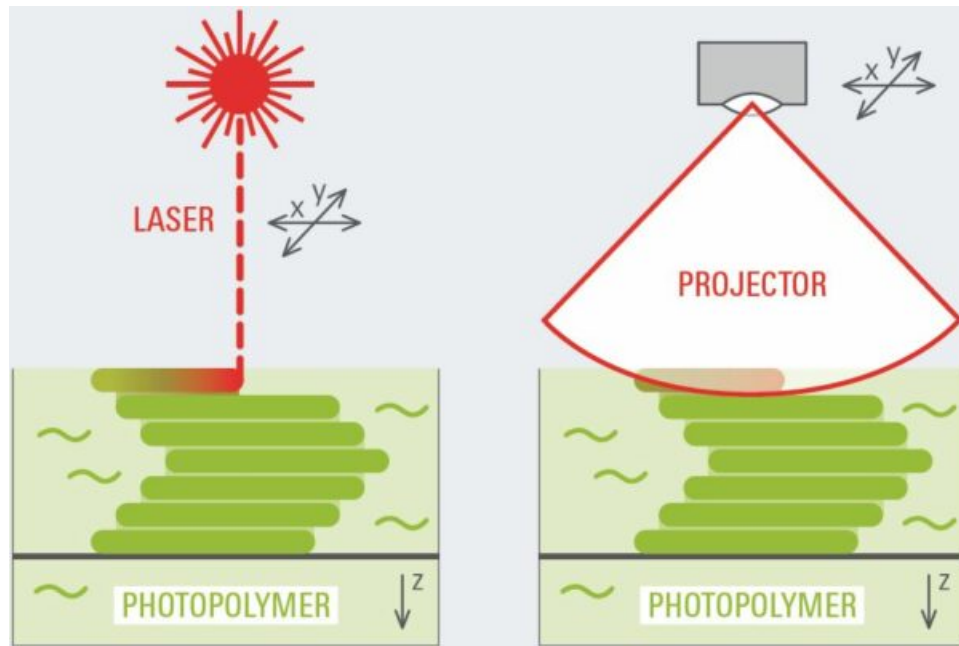


Figure 2.2. Schematic of VPP (left) and digital light processing (DLP)-based VPP (right). Source: [4].

2.1.3 Powder Bed Fusion

PBF works by laying an extremely fine layer of powdered material on the build surface then using a thermal source to selectively fuse the portion of the material layer required to form the part layer [7]. Once a layer is complete, the build surface moves downward and a new layer of powdered material is laid down and smoothed by a device called a recoater. When a part is complete, excess material is collected and can be reused. Lasers are the most common thermal sources for PBF. PBF processes which use lasers are called selective laser sintering (SLS) machines [7]. Some advantages for PBF are that it can be used with a wide variety of materials including polymers, metals, and ceramics. PBF can produce strong parts with a high level of detail. Some disadvantages are that not all unused powder can be recycled after a print. This leads to increased waste. Since most PBF machines use lasers, their power consumption is higher than other forms of AM [7]. Another disadvantage is that the powdered metal can create an explosion risk if improperly handled. Figure 2.3 shows two PBF methods.

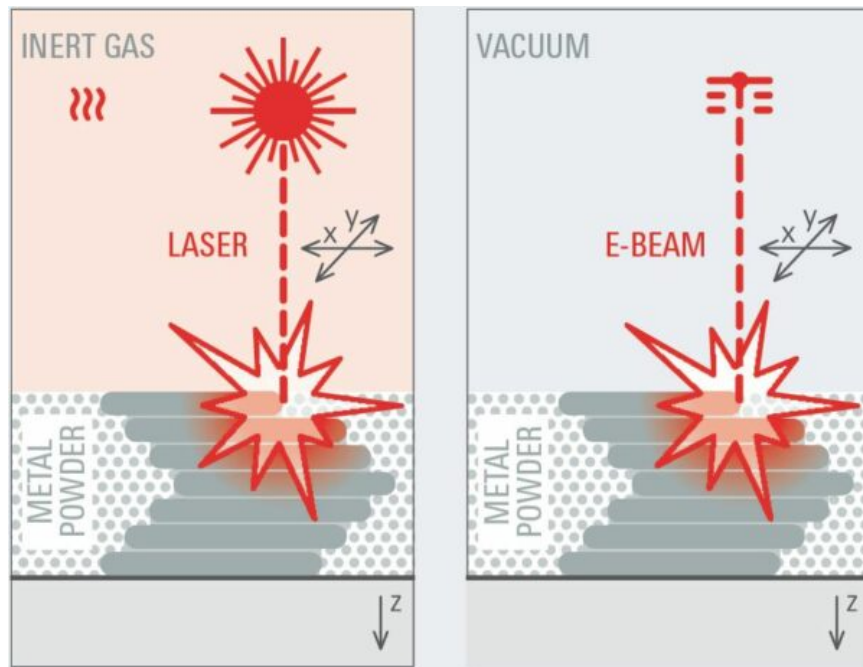


Figure 2.3. Schematic of Laser (left) and Electron Beam (right) PBF.
Source: [4].

2.1.4 Binder Jetting

BJT works similarly to PBF but instead of using a heat source to bind the material together, a binder is dispensed from the print head onto the material on the build surface [7]. Only a small amount of the part material is deposited by the print head. The majority of the material consists of powder from the print bed. Binder droplets are ejected onto the previously printed layer to form spherical agglomerates of binder liquid and powder. This forms the current layer. After each layer, the build plate is lowered so a new layer of powder can be spread onto it by the recoater, very similar to the recoding methods used in PBF [7].

BJT is a relatively fast process compared to other AM methods. BJT can be used with range of materials including polymers and metals and can colored using different color binders. BJT can have very large build volumes allowing increased efficiency at producing large numbers of parts simultaneously. BJT requires extensive post-processing for parts to be cleaned and strengthened which adds to total production time and cost. The extensive post-processing can lead to significant variance in final part dimension. Like PBF, not all the excess powder in BJT can be recycled leading to increased waste [7]. A schematic of the BJT process is shown in Fig. 2.4.

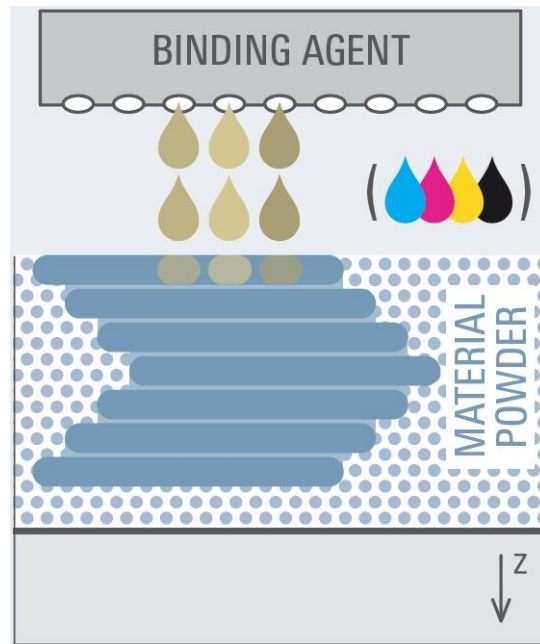


Figure 2.4. Schematic of BJT Process. Source: [4].

2.1.5 Material Jetting

MJT is similar to BJT except the base material (photopolymer) is jetted from the nozzle instead of present in bulk on the build surface [7]. A UV light cures the current layer as its jetted onto the previous layer. MJT is a medium speed process amongst AM. It has less waste material than BJT and PBF but is limited by viscosity requirements to polymers [7]. An example of MJT is shown in Fig. 2.5.

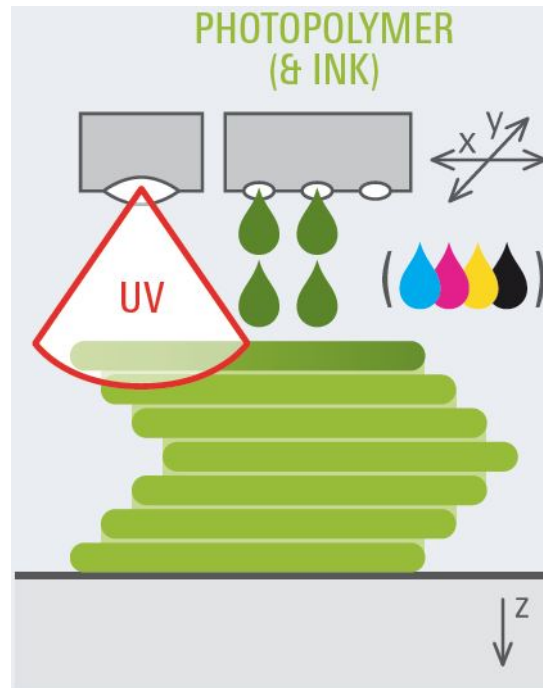


Figure 2.5. Schematic of MJT Process. Source: [4].

2.1.6 Directed Energy Deposition

DED is similar to PBF, but instead of melting material previously deposited on the build area, DED creates parts by melting the material as it is deposited [7]. Energy is focused on a region to heat the substrate, melting the material and partially melting the substrate. This effectively welds the new material to the pre-existing part. DED can be used for polymers, ceramics, and metals but is predominantly used for metal. DED produces parts with high levels of detail with the added benefit of being able to control grain structure. DED processes often require high levels of operator training because of their complexity [7]. Figure 2.6 shows two example methods of DED.

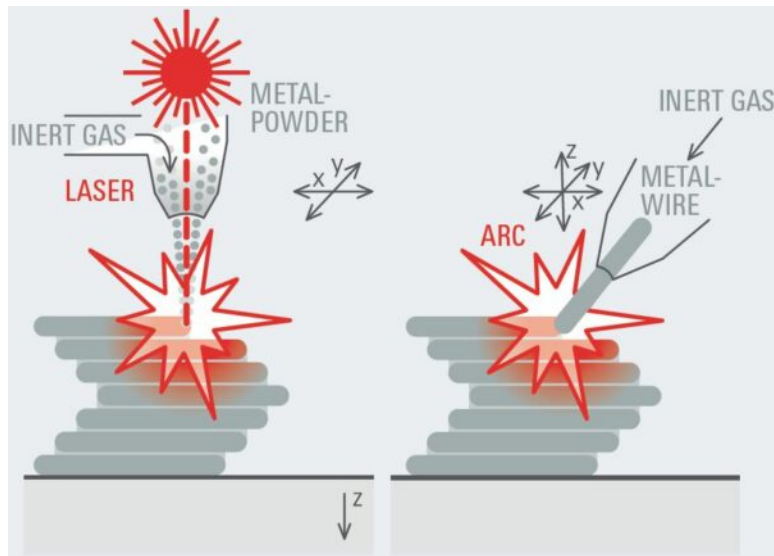


Figure 2.6. Schematic of Powder-Fed (left) and Wire-Fed (right) DED System. Source: [4].

2.1.7 Sheet Lamination

SHL works by cutting a cross-sectional area of the part from stock sheet material and adhering the sheets together to form the part [7]. SHL follows one of two orders of operation; either bond-then-form, where a layer of stock material is bonded to the previous layer and then cut to shape, or form-then-bond, where a layer of stock material is cut to shape then bonded with the previous layer. SHL is relatively low cost compared to other methods of AM, is faster than other methods, and its strength can far exceed that of other methods. SHL total production time is longer due to required post-processing. SHL material selection is limited [7]. Figure 2.7 illustrates SHL.

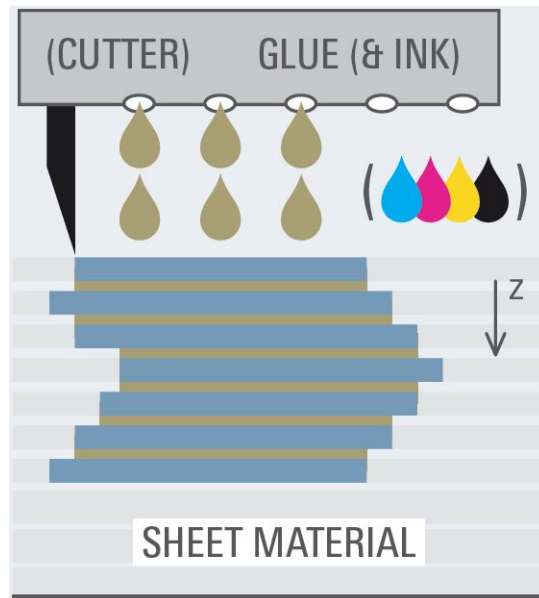


Figure 2.7. Schematic of SHL Process. Source: [4].

2.1.8 Liquid Metal Jet Printing

LMJP combines concepts from MEX and MJT to create a new AM method. LMJP uses bulk material in the form of spooled filament just like some forms of MEX. The filament is fed into a cauldron in the print head where it is melted to create a reservoir. A current conducting coil surrounds the reservoir. Electric pulses through the coil of varying current and frequency force rapid contraction of the molten pool causing small droplets to be jetted from the nozzle at the bottom of the reservoir. Inert shielding gas flows between the coil and the reservoir to help direct the jetted material. The jetted material coalesces on the build surface to form the part, layer by layer, in the same manner as MJT but without the need for UV curing or post-processing [8]. LMJP is relatively low cost to operate and has high cycle times, distinct advantages over MEX and MJT. Another advantage of LMJP, and why it was chosen for this research, is its ability to produce hollow metal parts with no post processing. LMJP is currently limited by build volume and material selection. At the time of this research, the only material commercially available is aluminum 4008. The most significant disadvantage to LMJP is that part mass is limited before the nozzle must be replaced, a lengthy process. Figure 2.8 illustrates LMJP.

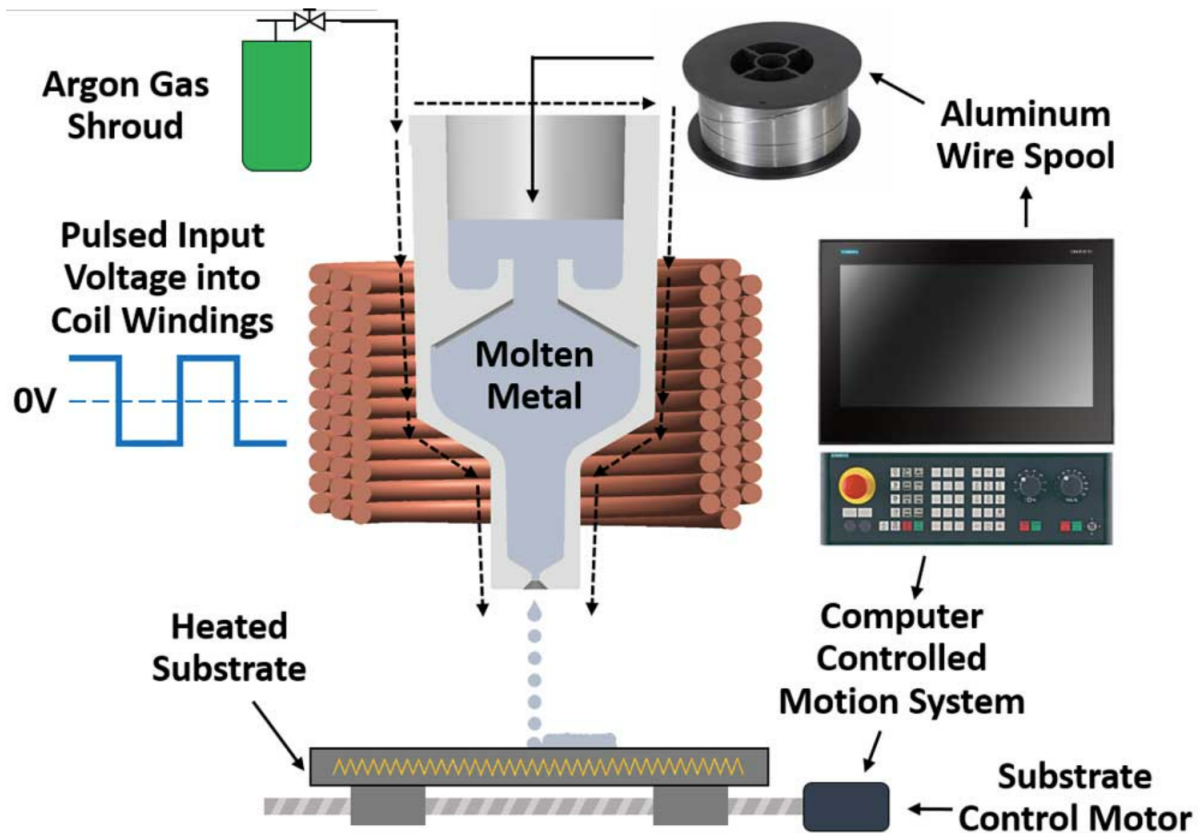


Figure 2.8. Schematic of LMJP Process. Source: [9]

2.2 Hollow Metal Parts

Traditional manufacturing processes and most AM processes require some form of post-processing to make hollow parts. This is because traditional manufacturing must produce hollow parts in either two pieces and join them or produce most of a hollow part using internal supports and then remove and seal the part. AM processes capable of producing hollow parts are either limited by materials, such as MEX, or require support materials, such as PBF which fills the part with powder. LMJP is one of the first metal AM methods capable of building hollow metal parts with no internal structures and no secondary processing.

2.3 Buckling

2.3.1 Eigenvalue Buckling vs. Non-Linear Buckling

Buckling is defined as the sudden deformation of a structural component under load. Buckling of a structure can be global, such as a column buckling, or it can be local, such as dimpling or deformation of a small area of a structure [10]. For a structure subjected to increasing load, the component is said to have buckled when a member suddenly deforms [10]. The load at which buckling occurs is referred to as the critical load.

The most basic method for studying buckling is a linear or eigenvalue buckling analysis. An eigenvalue buckling analysis solves a set of linear equations to predict a theoretical buckling load based on idealized elastic conditions and ignores material imperfections and non-linearities [11]. Equation 2.1 is the basic eigenvalue/eigenvector equation applied to a static structure under load,

$$\left([K] + \lambda_i[K_\sigma]\right)\{A_i\} = 0, \quad (2.1)$$

where K is the stiffness matrix of the structure, K_σ is the geometric stiffness matrix corresponding to the reference load F_{ref} , λ_i are the eigenvalues which represent the multipliers of the F_{ref} for the different modes of buckling, and A_i are the eigenvectors corresponding to the eigenvalues [12]. The lowest eigenvalue corresponds to the first mode of buckling and the corresponding load is called the critical load [13].

Because the eigenvalue approach assumes idealized elastic conditions, and it ignores material imperfections and non-linearities, the result is an over-prediction of the critical load [13]. An eigenvalue buckling analysis a simple and computationally inexpensive starting point but is ultimately inadequate for real world engineering applications.

Non-linear buckling analyses provide greater accuracy than the eigenvalue idealization because it takes into account large deformations such as those caused by snap-through [13]. As loading increases incrementally, a small change in load level causes a large change in deformation. This condition is called snap-through (Fig. 2.9), and it indicates that a structure has become unstable [13]. A nonlinear analysis recalculates K and K_{σ} for each iteration and uses that condition as the base case for the next iteration [13].

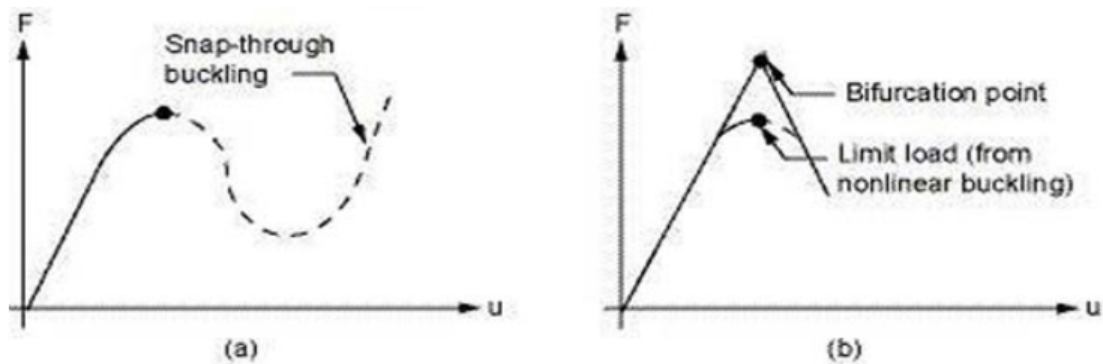


Figure 2.9. Load vs. Displacement for (a) Nonlinear Load-Deflection Curve
(b) Eigenvalue Load-Deflection Curve. Source: [14]

2.3.2 Buckling of the Cone-Cylinder Transition

The cone-cylinder transition was used based on the limitations of the LMJP process. This type of geometry has been studied extensively. Wenk Jr and Taylor (1953) conducted an analytical analysis of both reinforced and non-reinforced cone-cylinder transitions under external pressure and derived equilibrium equations for buckling [15]. Aylward (1973, 1975) compared the theoretical buckling equations of Wenk and Taylor to finite-difference program BOSOR 3 [16]. Galletly (1974) compared BOSOR 3 and BOSOR 5 theoretical calculations with experimental data. Galletly's experiments used machined thin-walled cone-cylinder models of aluminum alloy (HE-15) (Fig. 2.10) in a water-based pressure chamber [17].

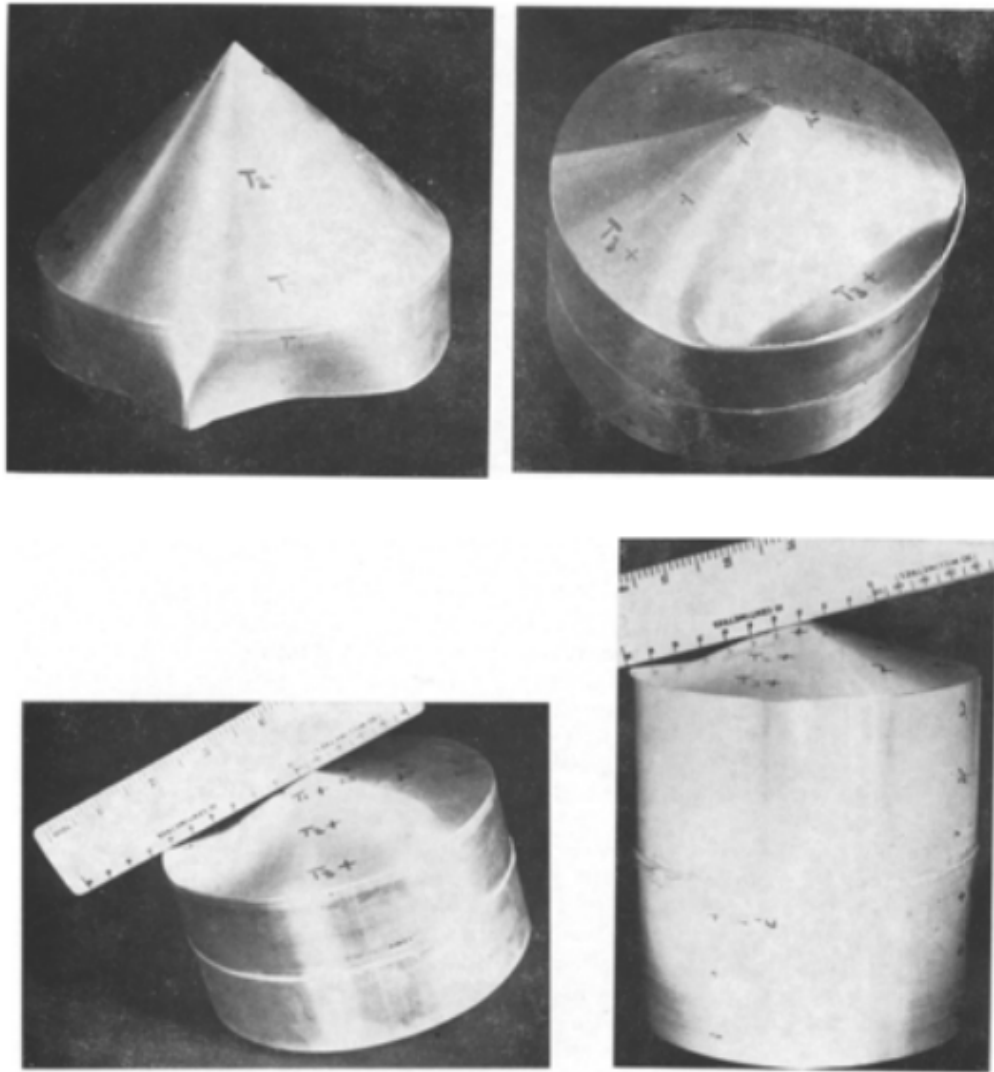


Figure 2.10. Galletly's Buckled Cone-Cylinder Models. Source: [17]

Galletly noted that:

1. The cone-cylinder assembly buckled in two ways:
 - locally at the cylinder or conical portion
 - at both the cylinder and conical portions at the same time
2. Theory and experimental data agreed within 5% for elastic asymmetric buckling.
3. Theory and experimental data agreed within 2% for elastic-plastic asymmetric buckling [17].

2.3.3 Previous Work

Gerald Qiu (2021) used the Xerox ElemX to produce and study hollow structures and energy absorbing structures. Qiu met limited success with the hollow parts because the initial release of the Xerox software was not able to accurately print overhangs and internal cavities (Fig. 2.11) [18]. Qiu help develop design guidelines for the Xerox ElemX, specifically, in regard to overhangs, inclines, and minimum wall thickness. Qiu's (2021) was foundational to the print success of Magnusson (2022).

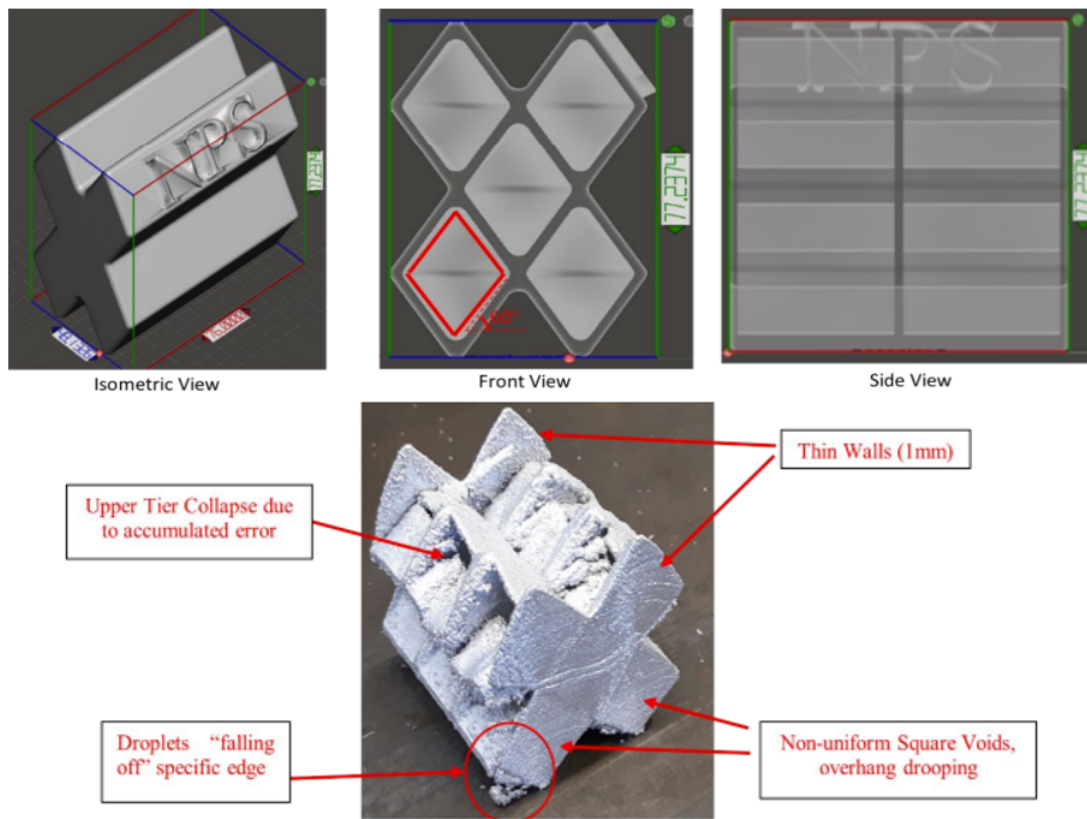


Figure 2.11. Hollow Parts Designed by Qiu (2021) for Marine Application (upper), Unsuccessful Production Using Xerox ElemX (lower). Source: [18]

Jacob Magnusson (2022) studied the cone-cylinder and compared Galletly's results with those of an eigenvalue analysis conducted in Ansys Mechanical Workbench FEA software [19]. Magnusson recreated Galletly's models in SolidWorks and used Ansys Mechanical Workbench to perform an eigenvalue buckling analysis. Six modes of buckling were

analyzed. Magnusson's Ansys eigenvalue buckling results did not achieve the same level of agreement with Galletly's. The closest result was 10.14% percent difference between Ansys and BOSOR, and 17.83% percent difference between Ansys and Galletly's experimental results [19].

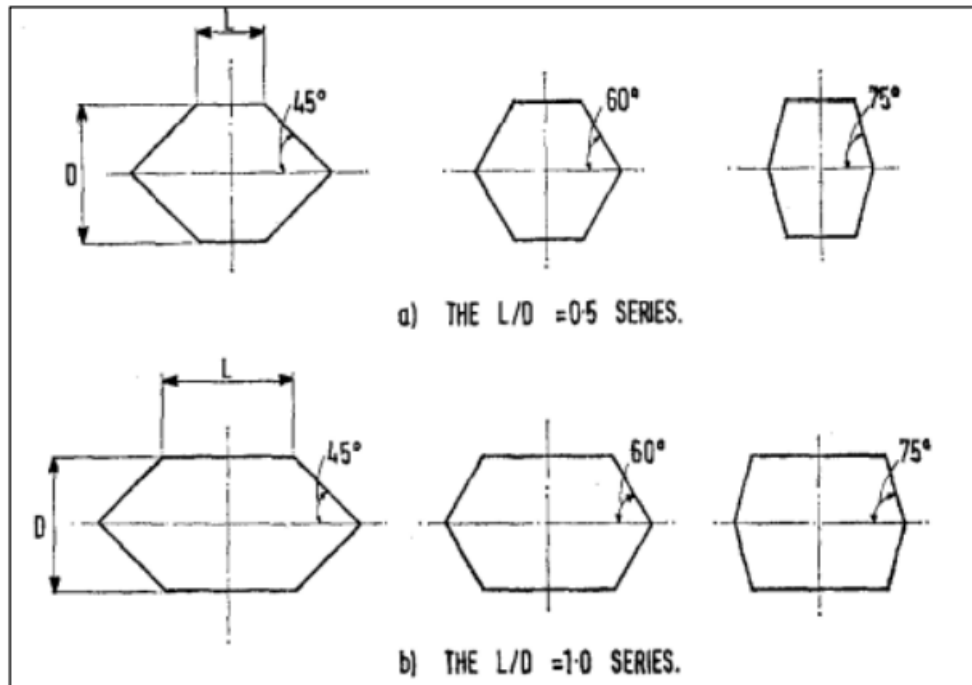


Figure 2.12. Dimensions of Galletly's Models. Source: [17]

Figure 2.12 shows the dimensions of the Galletly's models. Models 1 through 3 had a length to depth ratio (L/D) of 0.5, and models 4 through 6 had an L/D ratio of 1. Galletly also experimented with the angle of the cone from centerline, α , for both L/D [17].

Magnusson observed two trends in the eigenvalue analysis of Galletly's models:

1. α of 45° (models 1 and 4), were the closest match to Galletly.
2. α of 75° (models 3 and 6) deviated the most from Galletly [19].

CHAPTER 3:

Non-Linear Buckling Analysis

3.1 Hollow Part Design

The hollow parts manufactured for this work were produced using LMJP on a Xerox ElemX 3D printer. A standard operating procedure (SOP) for the Xerox ElemX was written for this work and is presented in appendix I. The SOP was adopted by the USN and is being employed onboard ships equipped with an ElemX.

The LMJP process has several design constraints that were considered. The first, is that the density of support material is not adjustable and affixes to the part with the same mechanical properties as the final part. This makes support removal difficult without specialty tools and impossible with complex part geometries. This property of LMJP informed the decision to focus on zero-post-processing designs. The second design consideration was the overhang constraint. Horizontal overhangs produced by LMJP are limited to 2 mm (0.08 in), and inclines must be greater than 50 degrees from horizontal or supports would be required [19]. This meant that spherical designs with a radius greater than 2 mm (0.08 in) were unachievable. Since the buoyancy difference between a spherical design for a 2 mm (0.08 in) overhang and a conical design is negligible, the conical design was selected for simplicity of manufacturing. Two different hollow designs were analyzed, each with multiple wall thickness variations. All parts were printed using only perimeters to minimize the length of print time.

3.1.1 Hollow Design Variant 1

Variant 1, shown in Fig. 3.1 and Fig. 3.2, is a single piece design meant to be attached externally to a payload [19]. Variant 1 was designed and printed by Jacob Magnusson (2022) for his contribution to this work (see reference [19]). Positive buoyancy was verified by Magnusson (2022). Experimental testing and results for variant 1 are presented in Section 4.2.1.

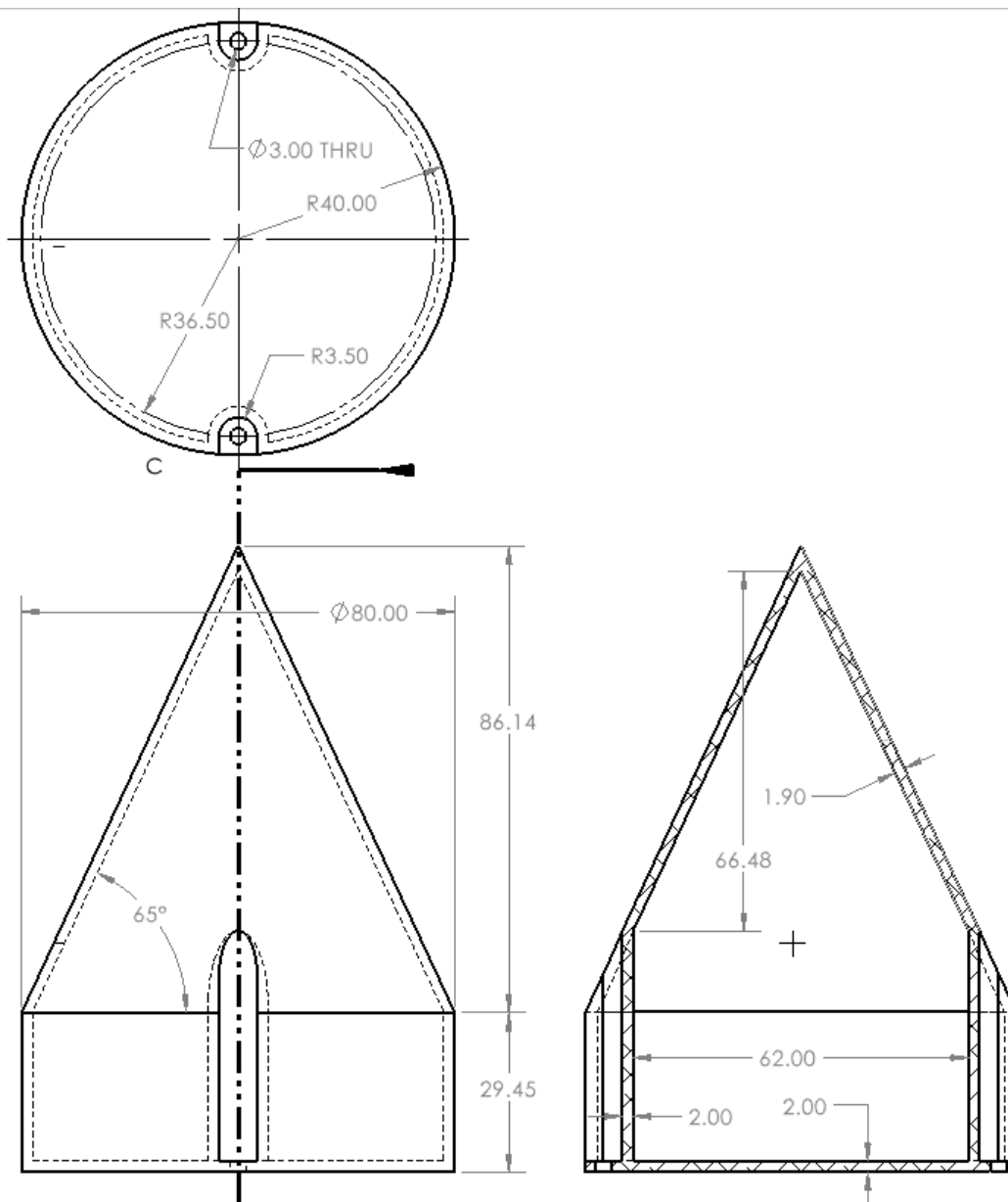


Figure 3.1. CAD: Buoyant Design Variant 1



Figure 3.2. Photo: Buoyant Design Variant 1

3.1.2 Hollow Design Variant 2

Variant 2, shown in Fig. 3.4, Fig. 3.5, and Fig. 3.6, was designed for this research and is a two-piece design comprised of the main body and end cap. Variant 2 is meant to carry a payload internally and is sealed with 1.90 mm (0.075 in) rubber gasket using eight (8) No.6 x 1/2" self-tapping torx screws. A computational analysis of variant 2 buoyancy was conducted for 1 mm (0.04 in), 2 mm (0.08 in), 3 mm (0.12 in), and 4 mm (0.15 in) wall thicknesses. The results were fit with a linear regression and are presented in Fig. 3.3. Experimental testing and results for variant 2 are presented in Section 4.2.2.

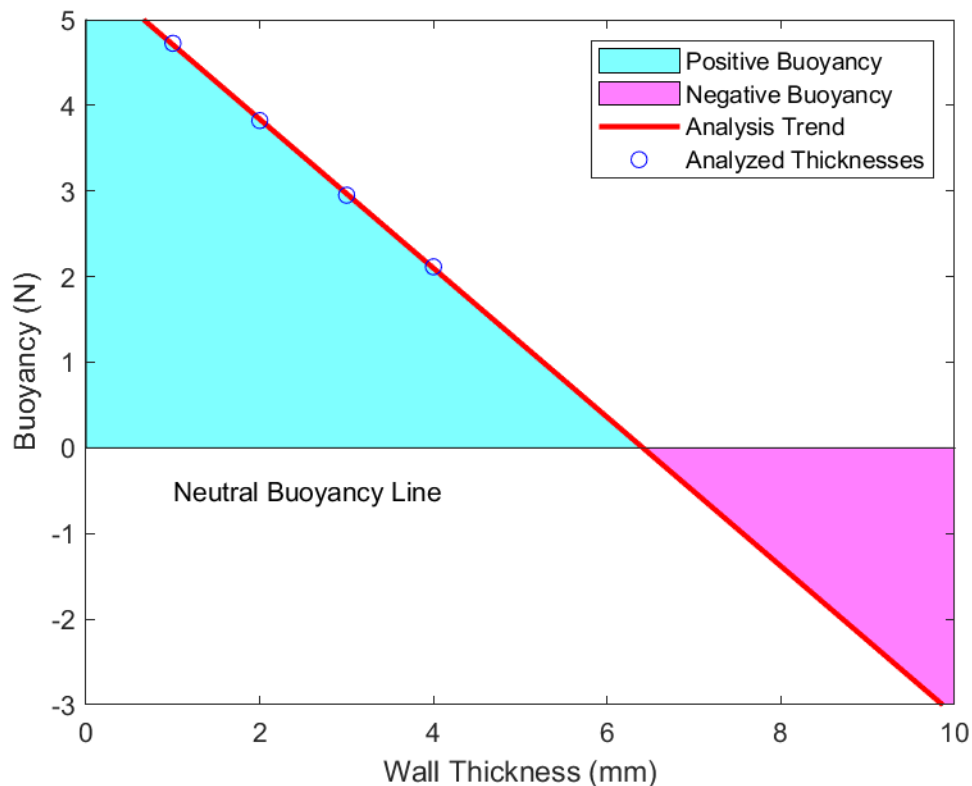


Figure 3.3. Buoyancy Calculation: Buoyant Design Variant 2

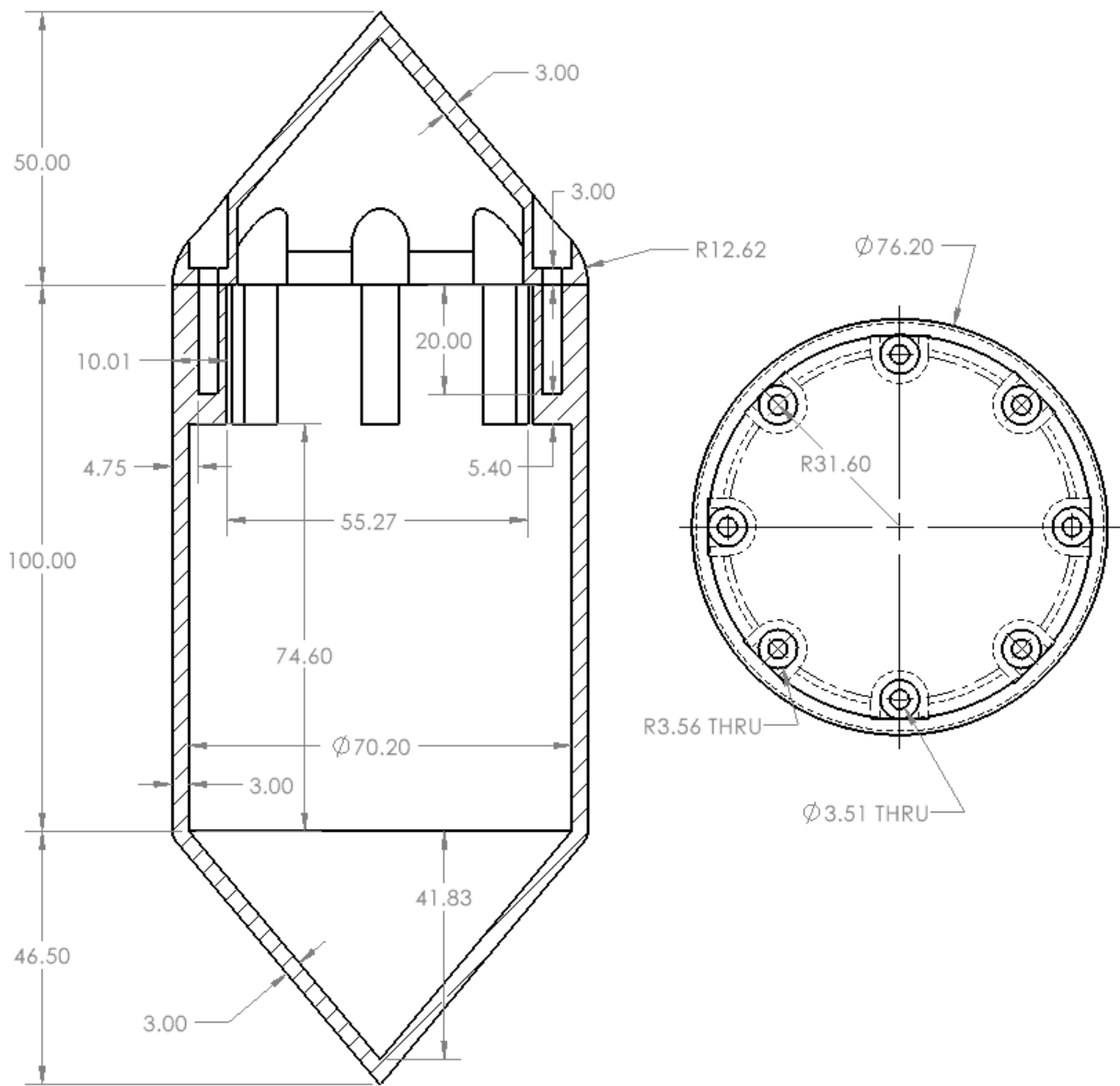


Figure 3.4. CAD: Buoyant Design Variant 2

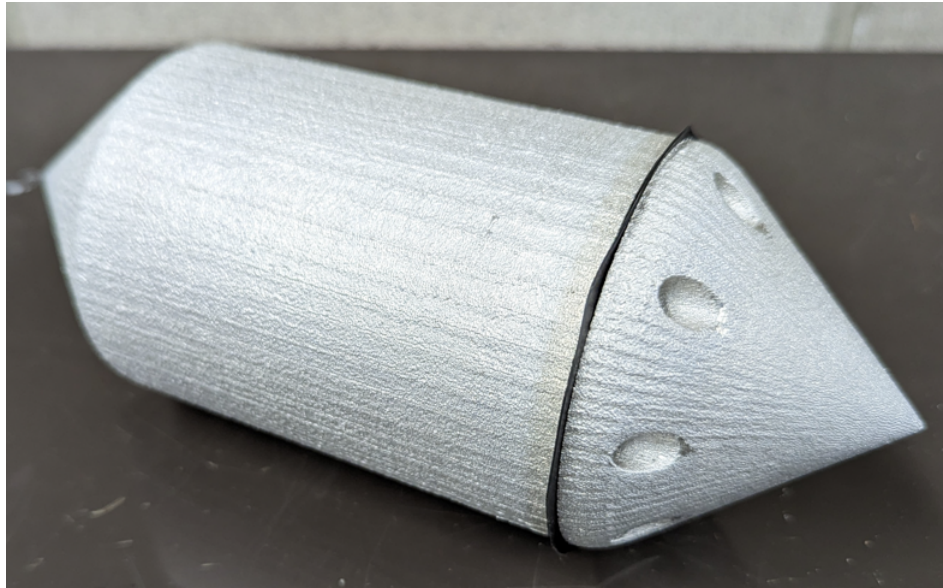


Figure 3.5. Buoyant Design Variant 2: Assembled

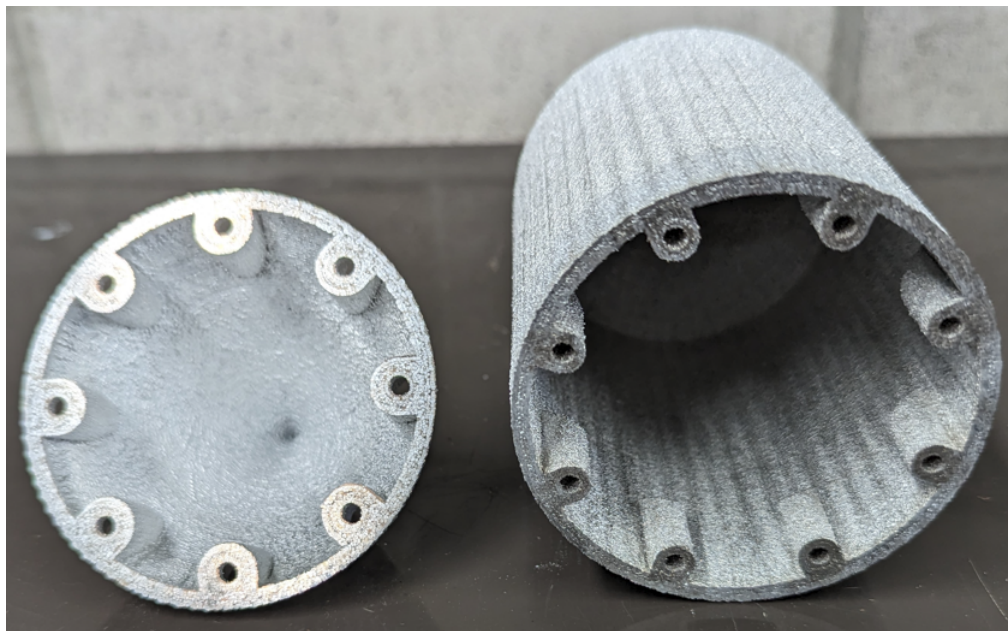


Figure 3.6. Buoyant Design Variant 2: Open

3.2 Non-Linear Analysis

A non-linear FEA was conducted using Ansys Mechanical Workbench software. The purpose of the FEA was to computationally determine the load at and mode of failure of hollow designs and compare it to the experimentally determined failure load and mode. The results of the comparison were used to inform the validity of using traditional structural engineering tools for the analysis of AM parts. Each hollow variant was subject to a static structural analysis. The results were then input into an eigenvalue buckling analysis to compute the buckling mode multipliers. A second static structural analysis was used to conduct a non-linear buckling analysis. The non-linear analysis used the buckling mode multipliers as a basis for the loading in conjunction with a second, small non-uniform load to create an instability. Figure 3.7 shows the Ansys block diagram.

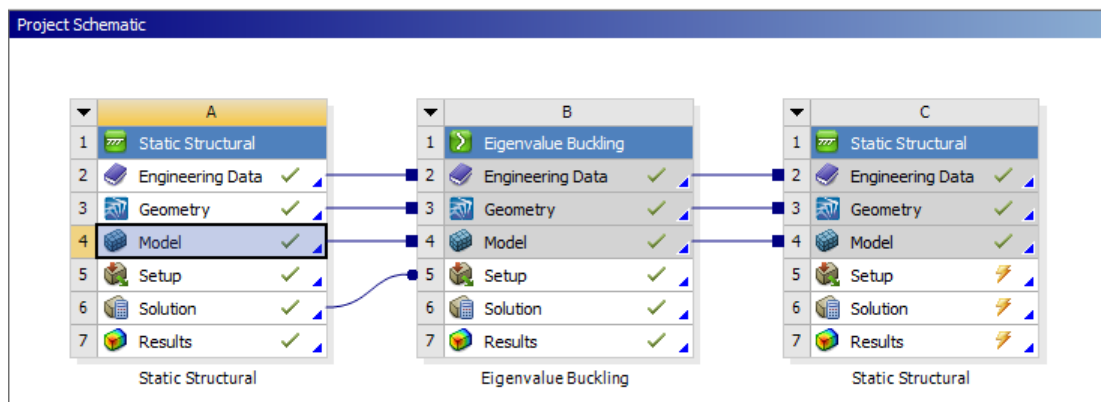


Figure 3.7. Ansys Mechanical Block Diagram

3.2.1 FEA Geometry

The hollow parts were designed in SolidWorks and exported in parasolid format (.x_t). The parasolid files were imported into Ansys and were bisected and rejoined to create artificial edges with which to establish boundary conditions. Figure 3.8 shows the geometry with the artificial edges generated.

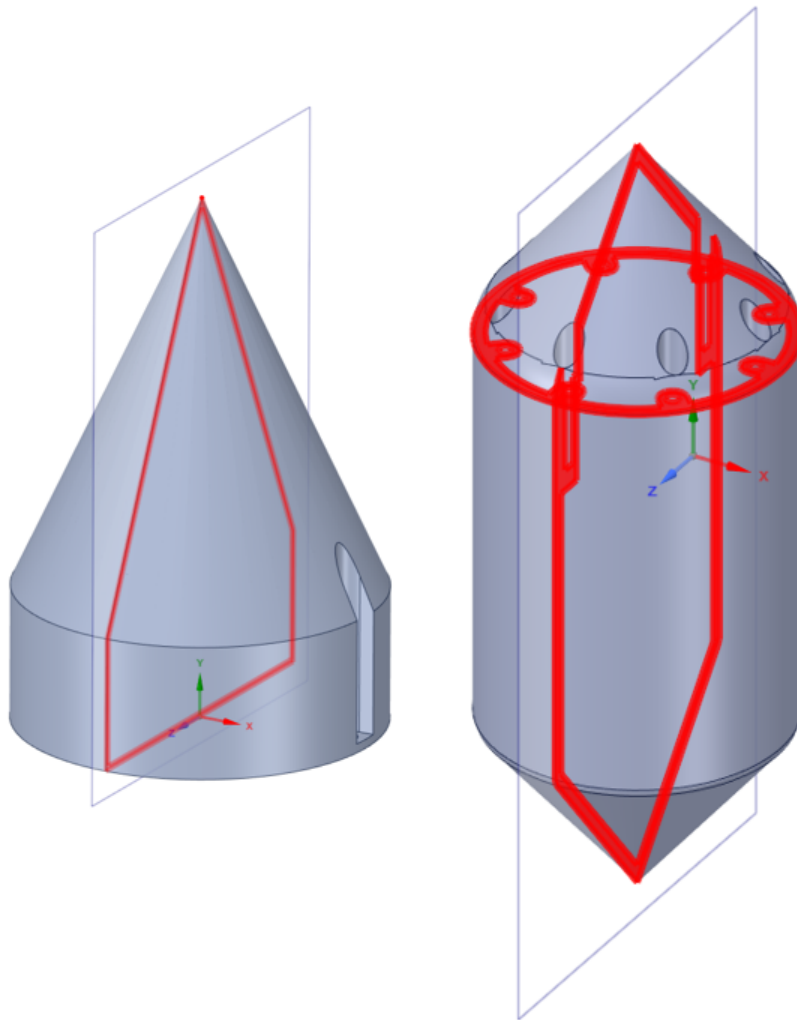


Figure 3.8. Ansys Mechanical Geometry

3.2.2 FEA Material Settings

Ansys default settings for aluminum alloy were used with the addition of the Ansys native bilinear isotropic hardening property. The material settings used are shown in Fig. 3.9. Ansys has the functionality to create a user defined material that could be populated with material properties for the LMJP specific aluminum. Collecting this data and creating an LMJP specific material for Ansys is discussed in Section 5.3 as potential further work.

Properties of Outline Row 3: Aluminum Alloy					
	A	B	C	D	E
1	Property	Value	Unit		
2	Material Field Variables	Table			
3	Density	2770	kg m ⁻³		
4	Isotropic Secant Coefficient of Thermal Expansion				
5	Coefficient of Thermal Expansion	2.3E-05	C ⁻¹		
6	Isotropic Elasticity				
7	Derive from	Young's M...			
8	Young's Modulus	7.1E+10	Pa		
9	Poisson's Ratio	0.33			
10	Bulk Modulus	6.9608E+10	Pa		
11	Shear Modulus	2.6692E+10	Pa		
12	Bilinear Isotropic Hardening				
13	Yield Strength	2.8E+08	Pa		
14	Tangent Modulus	7E+09	Pa		
15	S-N Curve	Tabular			
16	Interpolation	Semi-Log			
17	Scale	1			
18	Offset	0	Pa		
19	Tensile Yield Strength	2.8E+08	Pa		
20	Compressive Yield Strength	2.8E+08	Pa		
21	Tensile Ultimate Strength	3.1E+08	Pa		
22	Compressive Ultimate Strength	3.1E+08	Pa		

Figure 3.9. Ansys Mechanical Material Settings

3.2.3 FEA Mesh Settings

The mesh setting for this FEA, shown in Fig. 3.10, were set to the max value in Ansys without inserting additional refinements. This mesh setting was chosen to keep the mesh element size smaller than the minimum dimension of the limiting geometric feature but minimizing the complexity as to reduce computation cost.

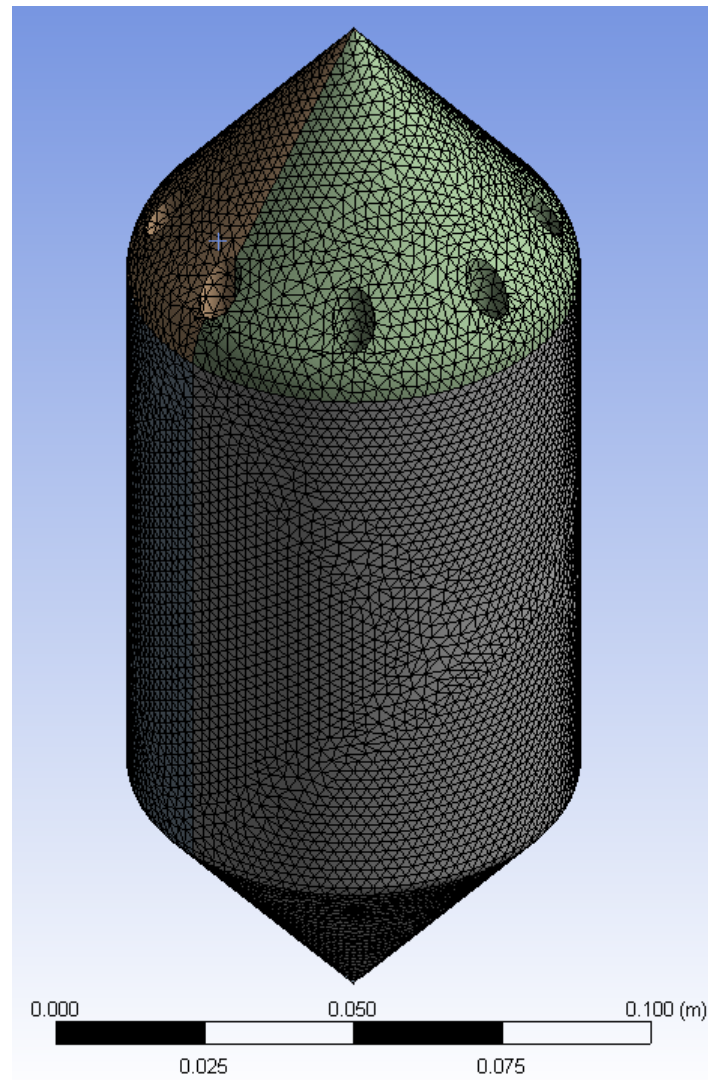


Figure 3.10. Ansys Mechanical Mesh Settings

3.2.4 FEA Boundary Conditions

There were three spacial boundary condition applied to these geometries. A zero-displacement constraint in the X , Y , and Z directions for a point on the bisection was imposed to anchor the geometry. A second zero-displacement constraint in the X and Y directions was placed on a point on the opposite side of the geometry, away from the first in only the Z direction. The second constraint restricted rotation about the X and Y axis. A third zero-displacement

constraint in the X direction was placed on a point in the YZ plane away from the second constraint in the Y direction. This constraint prevented rotation about the Z axis. Figure 3.11 shows the geometry with the applied constraints.

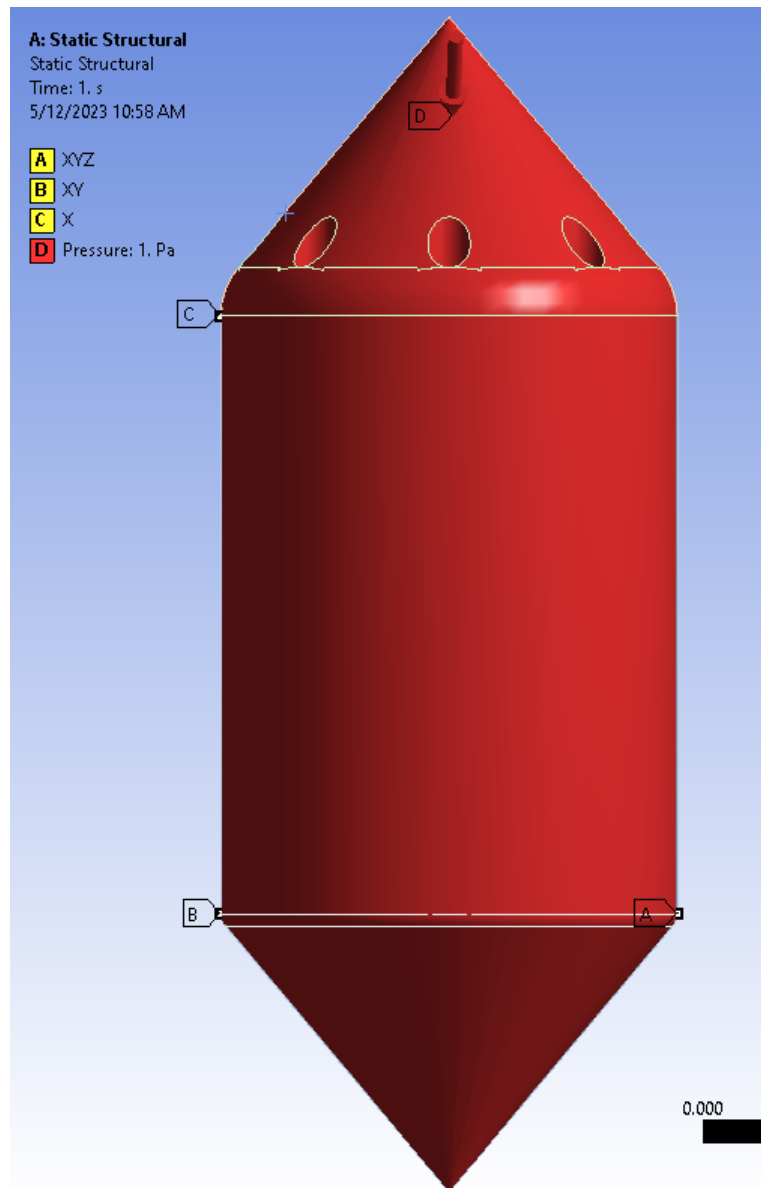


Figure 3.11. Ansys Mechanical Constraints

3.2.5 Static Structural Analysis

For the initial static structural analysis, a pressure of 1.0 Pa (1.45×10^{-4} psi) was applied perpendicular to all external surfaces using a single load step. 1.0 Pa (1.45×10^{-4} psi) was chosen so that the buckling multiplier in the eigenvalue buckling analysis would indicate the actual theoretical buckling pressures. Figure 3.11 shows the applied pressure.

3.2.6 Eigenvalue Buckling Analysis

For the eigenvalue buckling analysis, the first ten modes were analyzed. The resulting load multiplication factors were used for the non-linear analysis. Figure 3.12 shows an example result of the first buckling mode and for the load multiplication factors for hollow variant 2.

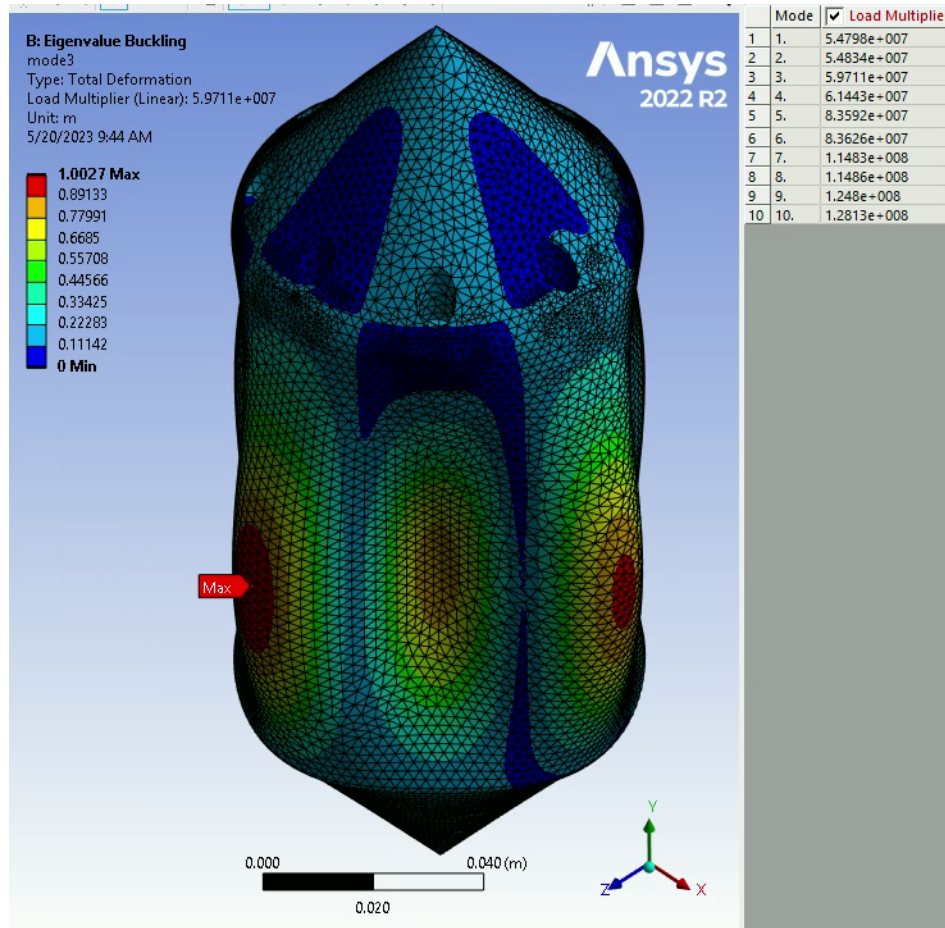


Figure 3.12. Example: Ansys Mechanical Mode 1 Eigenvalue Buckling Result for Variant 2

3.2.7 Non-Linear Buckling Analysis

The non-linear analysis was performed using a second static structural block with the same spacial constraints as the first. Two pressures were applied for the non-linear analysis. The first was a 3000 Pa (0.435 psi) load applied to a single section on one side of the geometry in an area likely to buckle based on the eigenvalue analysis. This load was used to generate the instability for non-linear buckling. The second load was applied to all external surfaces like the first static block, but this pressure was applied over 100 time steps. To ensure failure, the max pressure used was equal to the next order of magnitude above the highest eigenvalue load multiplier. For this analysis, the large deflections setting was enabled. Figure 3.13 shows the non-linear loading.

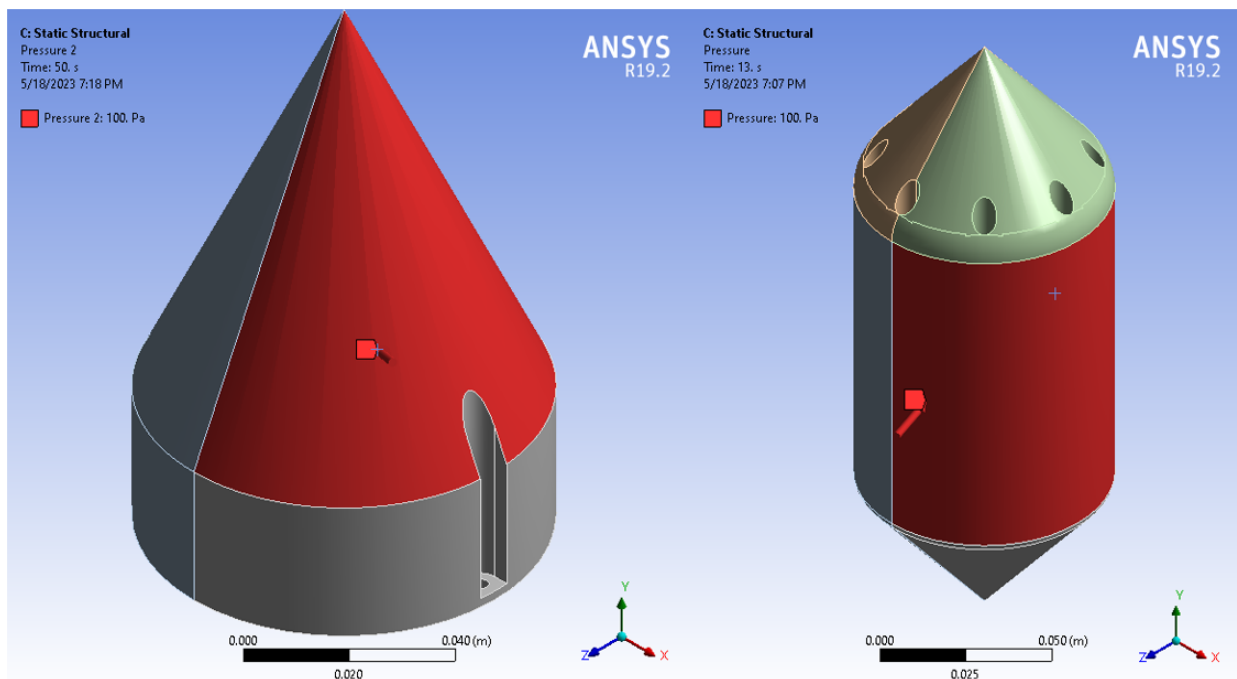


Figure 3.13. Example: Ansys Mechanical Non-linear Loading

3.2.8 Analysis: Design Variant 1

The non-linear analysis of variant 1 predicts buckling of the conical section on one side (Fig. 3.14) and on the bottom side (Fig. 3.15). Max deflection is predicted on one side of the conical section. Max principle stress is predicted on the bottom side near the mounting holes (Fig. 3.16). It was inferred based on the predicted principle stresses that as the bottom buckles inward, the small, flat sections around the mounting holes will act as pivot points, putting the outside of the bottom face under tension as it is stretched over the mounting flats. The predicted failure loads for variant 1 are presented in table 3.1.

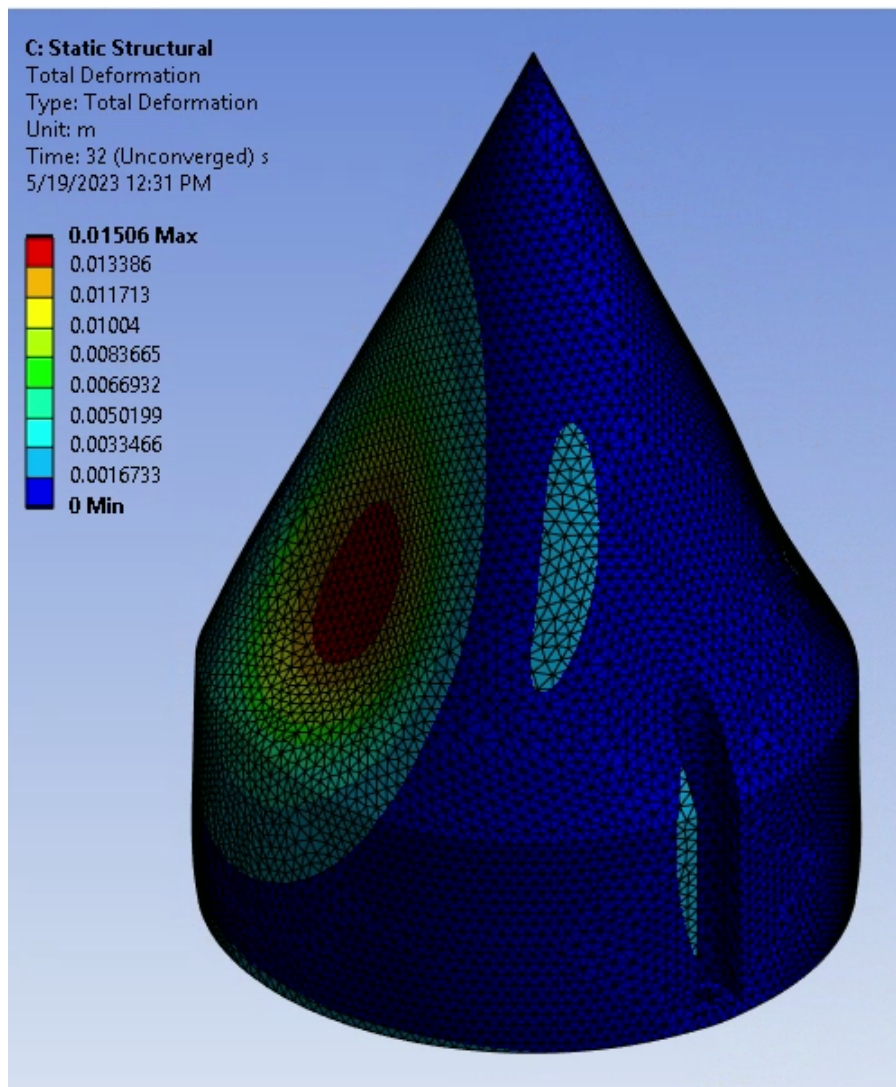


Figure 3.14. Non-Linear Deformation of 2 mm Variant 1: Cone Section

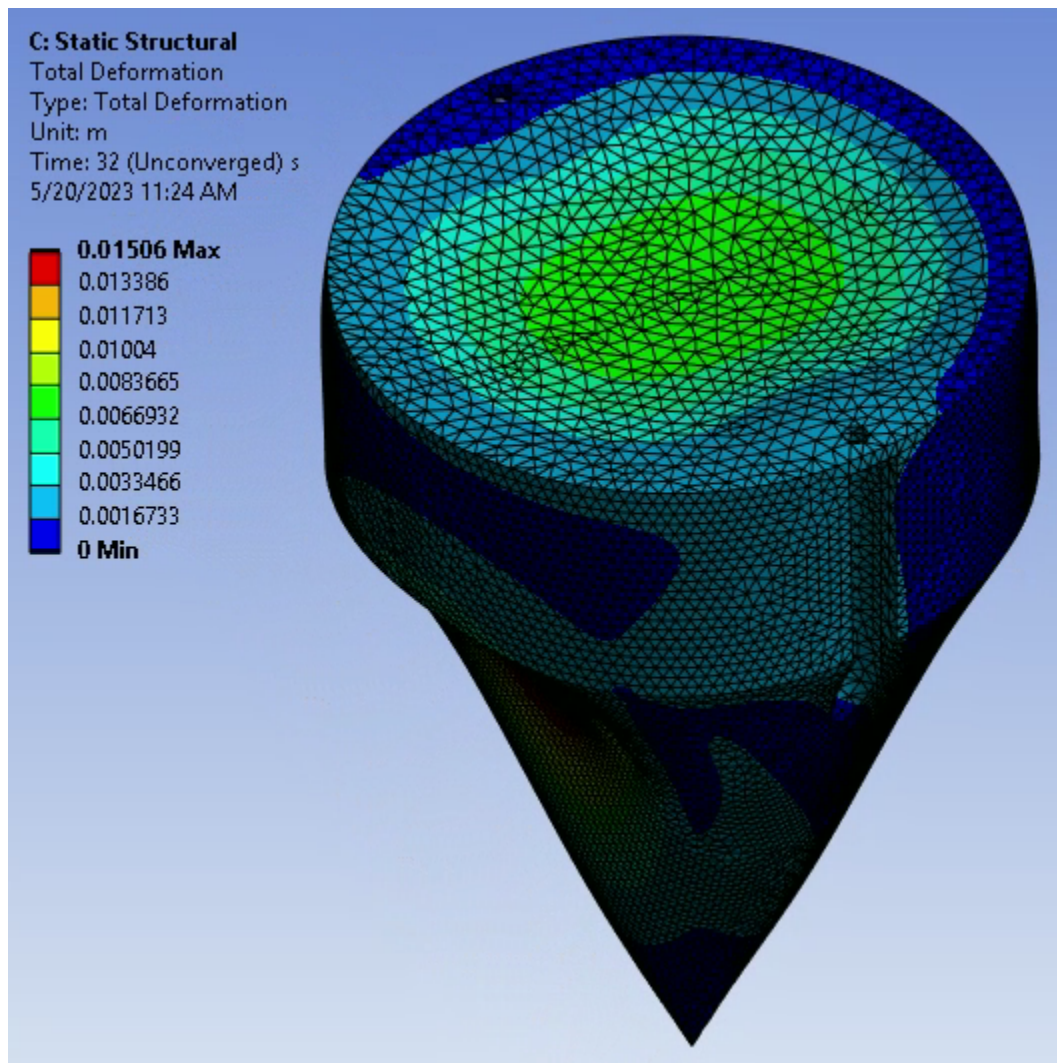


Figure 3.15. Non-Linear Deformation of 2 mm Variant 1: Bottom Section

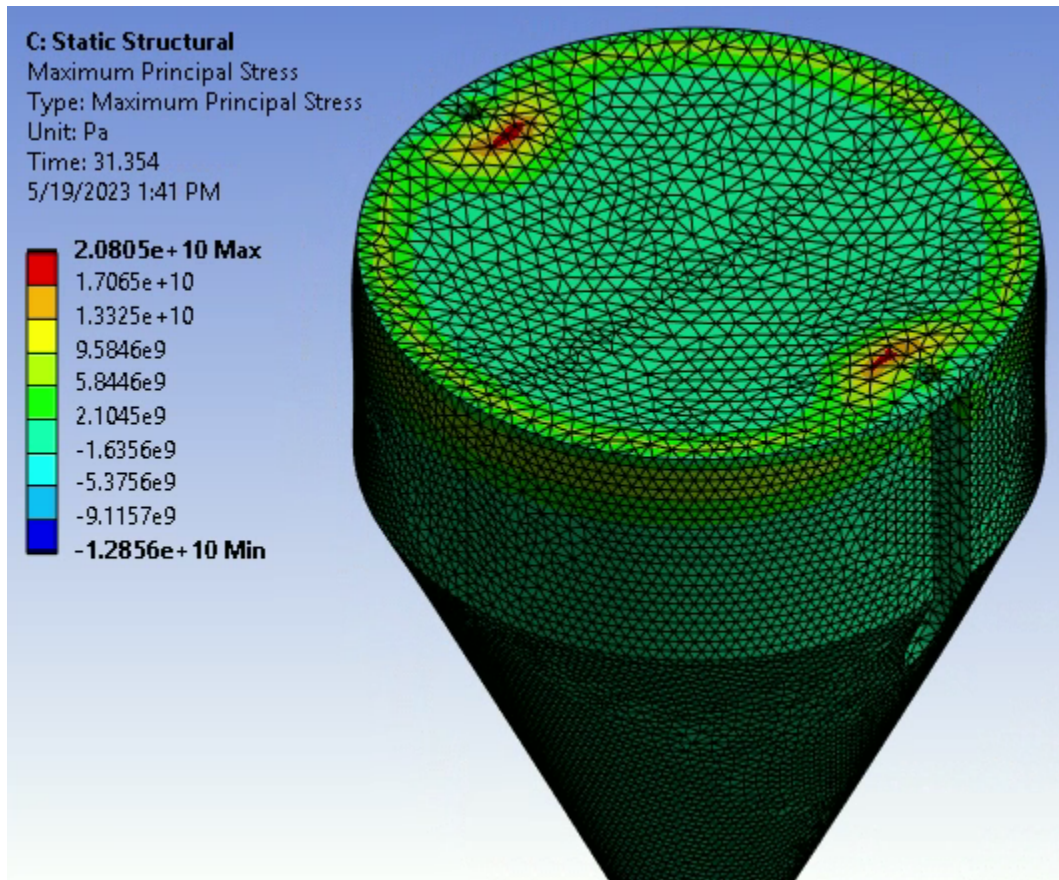


Figure 3.16. Max Principle Stress of a 2 mm Variant 1

Table 3.1. FEA Predicted Loads at Failure: Variant 1

Thickness (mm)	kPa	psi
1.0	1.03×10^7	1.49×10^6
1.5	1.61×10^7	2.34×10^6
2.0	2.10×10^7	3.05×10^6

3.2.9 Analysis: Design Variant 1

The non-linear analysis of variant 2, shown in Fig. 3.17, predicts buckling of the main housing with max deflection along the cylindrical section. Max principle stress is predicted along the curvature of the main housing where the conical section transitions to the cylindrical section (Fig. 3.18). Specific “hot spots” for max principle stress occurred at the locations of the boundary conditions. This was due to a numerical artifact in the FEA solver from the boundary conditions. This could be removed in future work with additional boundary condition refinements. The predicted failure loads for variant 1 are presented in table 3.2.

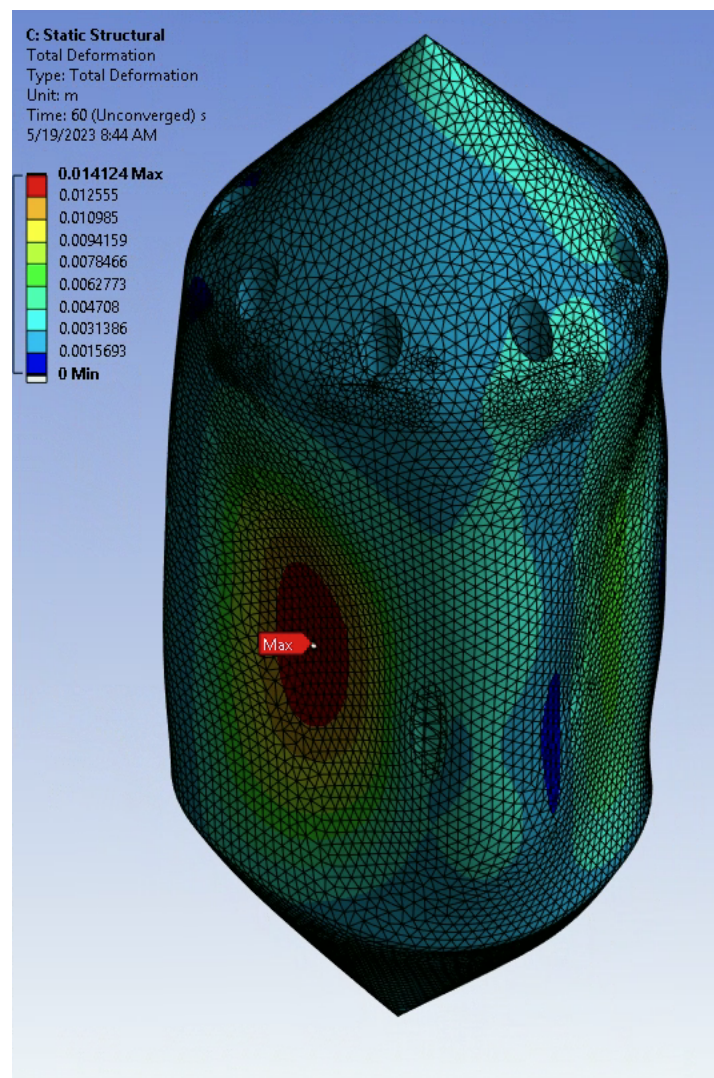


Figure 3.17. Non-Linear Deformation of Variant 2

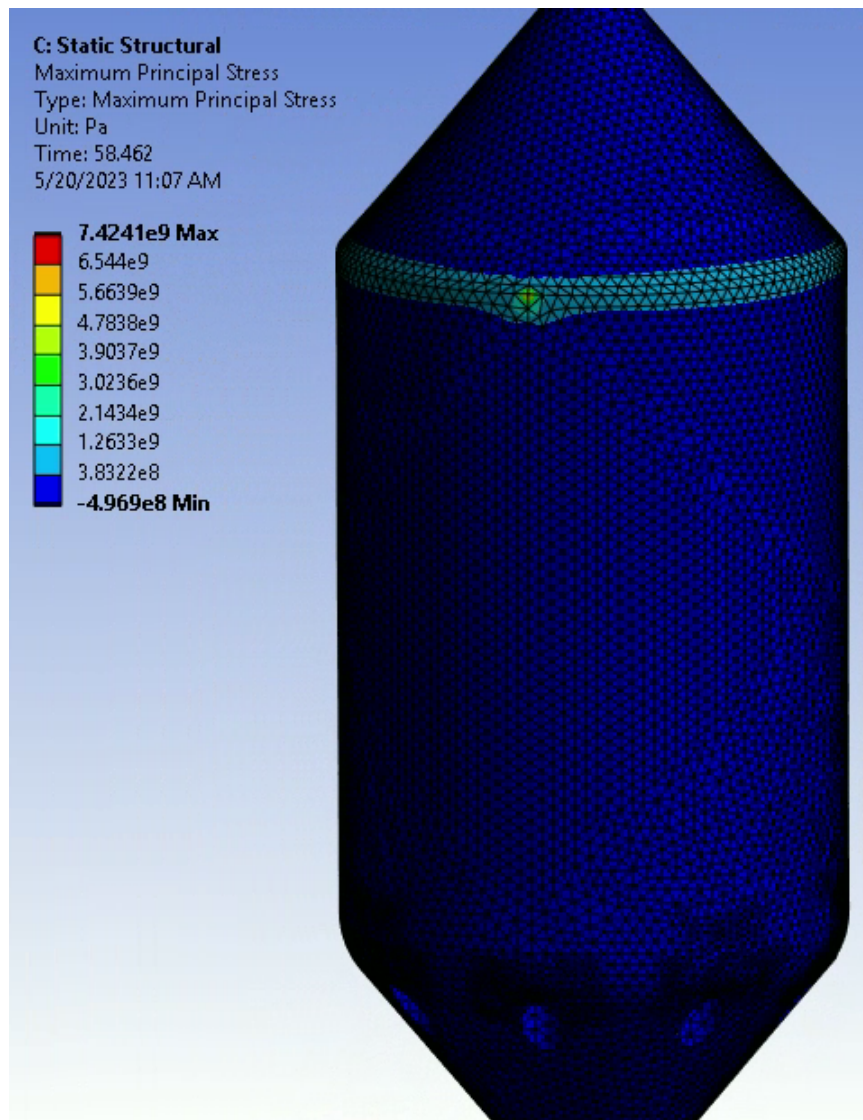


Figure 3.18. Max Principle Stress on Variant 2

Table 3.2. FEA Predicted Loads at Failure: Variant 2

Thickness (mm)	kPa	psi
2.0	7.61×10^6	1.10×10^6
3.0	3.94×10^7	5.71×10^6

CHAPTER 4:

Hydrostatic Testing

NOTE: All experimental pressures reported in this chapter are reported as gauge pressures unless otherwise stated.

4.1 Design and Testing of the Pressure Test Rig

To conduct pressurization experiments on the hollow parts, a test rig capable of safely operating up to 2.76×10^7 Pa (4000 psi) was needed. The test rig was constructed from thick-walled (1.65 mm (0.065 in)) stainless steel tubing (outer diameter 6.3 mm (0.25 in)) using 6.3 mm (0.25 in) National Pipe Thread (NPT) connections. Additional components included commercial off-the-shelf (COTS) valves, gauges, pressure transducers, and a manual hand pump. The manual hand pump was chosen to limit the rate of pressure increase for safety concerns. The test rig pressure vessel utilized was a heat treated 17-4 PH stainless steel canister previously used for diesel ignition monitoring and measurement [20]. The canister was cleaned and instrument penetrations were sealed.

4.1.1 Pressure Vessel End Cap Design

A new end cap was designed for the canister to replace a previously installed viewing window with a vent valve. The vent valve ensured no air was trapped in the pressure vessel. The replacement end cap was designed based on the opposite end cap. It was then modeled in SolidWorks computer-aided design (CAD) software and analyzed in Ansys Mechanical Workbench FEA software to verify adequate strength and safety. Figure 4.1 shows the results of the FEA. The maximum principle stress experienced by the new end cap design occurred in the ASTM grade 8 bolts used to secure the end cap to the vessel. The minimum tensile strength of ASTM grade 8 bolts at 1.03×10^9 Pa (1.5×10^5 psi) [21]. The predicted bolt stress was 8.87×10^8 Pa (1.29×10^5 psi) which results in a design margin of 0.14. The maximum operating pressure was reduced to 2.28×10^7 Pa (3300 psi) to provide a design margin of 0.29.

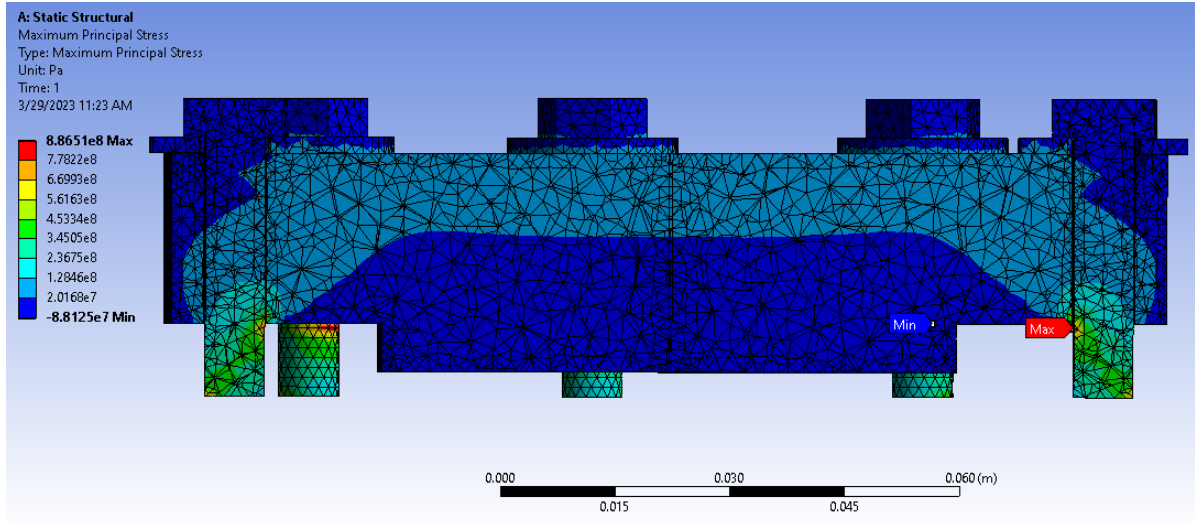


Figure 4.1. Ansys analysis of the pressure vessel end cap replacement

The minimum bolt engagement length required for the end caps was calculated using Eq. (4.1) through Eq. (4.4),

$$d_p = d_N - 0.6495p, \quad (4.1)$$

$$Le_{hole} \geq \frac{Rm_{screw}}{Re_{hole}} \frac{2A_s}{0.6\pi d_N}, \quad (4.2)$$

$$Le_{screw} \geq \frac{Rm_{screw}}{Re_{screw}} \frac{2A_s}{0.6\pi d_p}, \quad (4.3)$$

$$Le_{min} \geq \max(Le_{hole}, Le_{screw}), \quad (4.4)$$

where Le is the engagement length of the bolt, Rm is the ultimate strength, Re is the yield strength, A_s is the stressed cross sectional area, d_N is the nominal diameter, p is the thread pitch, and d_p is the pitch diameter [22].

A torque specification for the pressure vessel end caps was not provided with the pressure vessel so a specification was calculated. The torque specification was calculated using Eq. (4.5) through Eq. (4.10),

$$A_o = \frac{\pi}{4} \left(d_N - \frac{0.9743}{n} \right), \quad (4.5)$$

$$F_{failure} = \frac{\sigma_{yield} A_o}{1 - \sigma_{yield} \frac{\nu L_o}{E_Y A_o}}, \quad (4.6)$$

$$F_{pressure} = \frac{P_{max} \pi d_E^2}{32}, \quad (4.7)$$

$$F_{torq} = \frac{T}{k d_N} \frac{1}{1 - \frac{I}{100}}, \quad (4.8)$$

$$F_{torq} + F_{pressure} < F_{failure}, \quad (4.9)$$

$$T = k d_N \left(1 - \frac{I}{100} \right) \left(\frac{\sigma_{yield} A_o}{1 - \sigma_{yield} \frac{\nu L_o}{E_Y A_o}} - \frac{P_{max} \pi d_E^2}{32} \right), \quad (4.10)$$

where n is the threads per inch, σ_{yield} is the yield stress of the bolt, ν is Poison's ratio of the bolt, E_Y is Young's Modulus of the bolt, k is a constant dependent on bolt material and size, L_o is the stressed length of the bolt, d_E is the diameter of the area of the end cap exposed to system pressure, P_{max} is the maximum pressure of the system, I is the lubrication factor based on the thread lubricant used, T is the torque applied to the bolt, A_o is the original cross sectional area of the bolt under tensile stress, $F_{failure}$ is the force at bolt failure, $F_{pressure}$ is the force do to the system pressure on the bolt, and F_{torq} is the force due to the applied torque on the bolt [23]. The chosen specification from the calculation was 24.6 ± 2 N-m (218 ± 17 in-lbs). The Matlab code used to calculate the torque specification is presented in appendix A. The complete code block for the test rig is presented in appendix B.

4.1.2 Data Acquisition

DAQ was conducted using two Balluff BSP00YH / BSP B400-EV009-P00S2B-S4 pressure transducer (PT) rated to 4.0×10^7 Pa (5800 psi) and a National Instruments (NI) PS-15 power supply coupled with an NI USB6003 terminal block. The terminal block was connected via USB to a laptop utilizing NI LabView for DAQ and GUI. The PTs were wired in voltage mode due to limitations of the terminal block. Figure 4.2 shows the wiring diagram for the DAQ instrumentation. A block diagram for the NI code and a screenshot of the GUI are presented in appendix C.

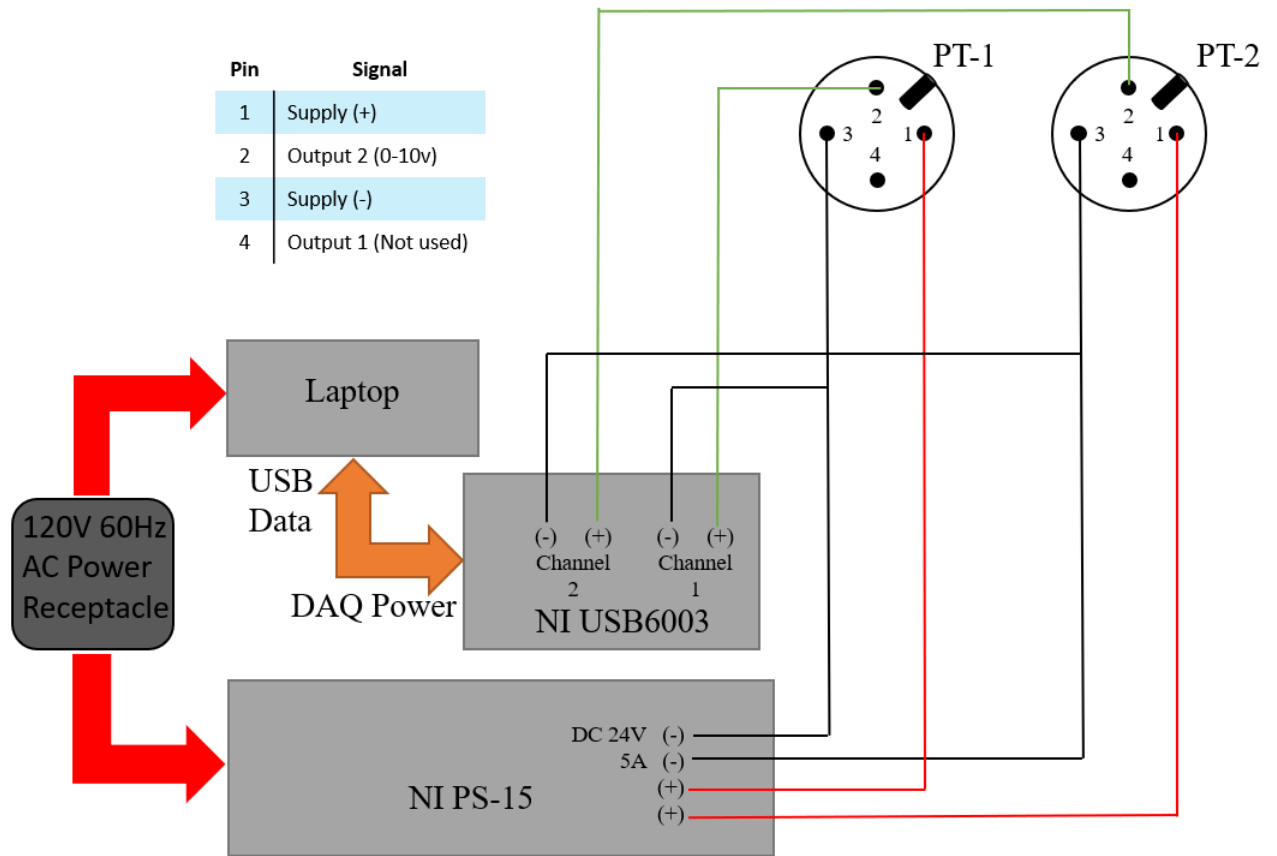


Figure 4.2. DAQ 4-Pin Wiring Diagram

The error associated with the PTs and the DAQ provided by the manufacturer were used to estimate a total system error using the error propagation equation (Eq. (4.11)),

$$\delta Q = \sqrt{(\delta PT)^2 + (\delta TB_{input})^2 + (\delta TB_{noise})^2}, \quad (4.11)$$

where δQ is the total system error, δPT is the error associated with the PTs, δTB_{input} is the input error associated with the NI terminal block, and δTB_{noise} is the system noise error associated with the NI terminal block. The component and system errors are presented in table 4.1.

Table 4.1. Component and System DAQ Errors

Percent Error			
PT	NI Terminal Block Input	NI Terminal Block Input Noise	Total
0.5	0.03	0.002	0.5009

A study was conducted to determine the error of the DAQ system. A set of pressure data sampled at 500Hz was analyzed for a 15 minute hold at 1.03×10^7 Pa (1500 psi). There was large variance in the reported pressure so the data was plotted on a histogram (Fig.). The histogram showed bi-nodal behavior, which indicated a significant source of noise in the data signal.

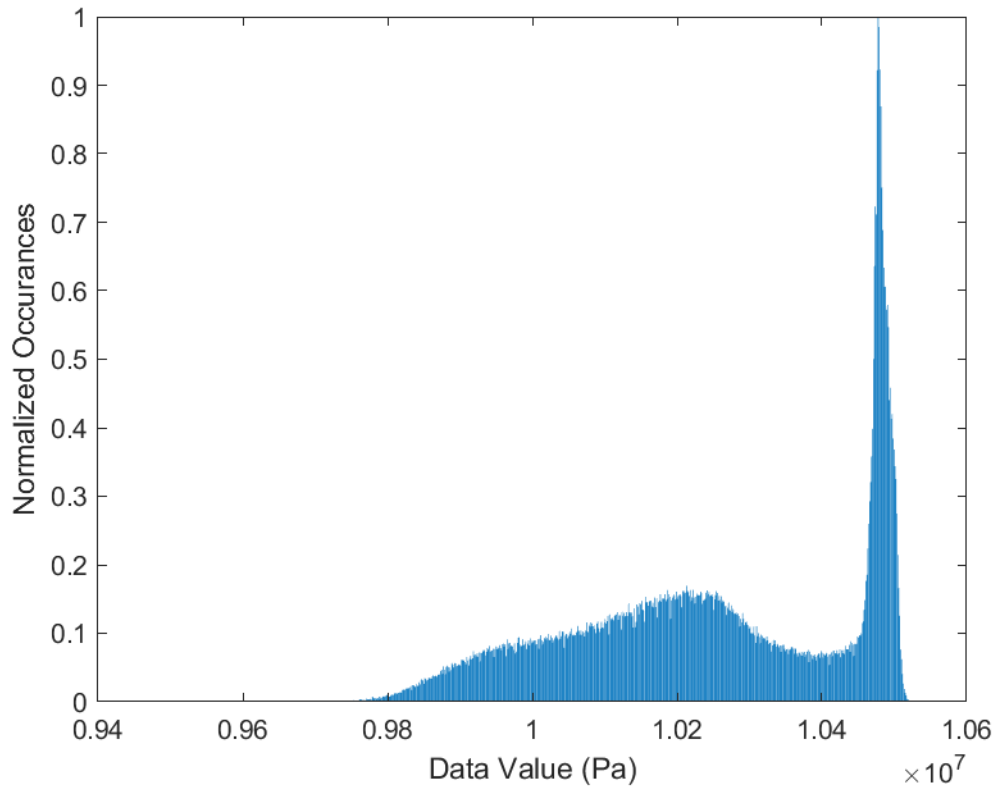


Figure 4.3. DAQ Raw Signal Histogram

A power spectral density (PSD) estimate revealed two noise sources, a weak noise signal at 20Hz and a stronger noise signal at 60Hz. The data was filtered using a moving mean with averaging windows of 20 to 200 samples in increments of 20 samples. The 20-sample window removed the noise signal peak at 100Hz and in subsequent 100 Hz intervals. It also reduced all other noise strengths by a decade (Fig. 4.5). The 100-sample window and multiples of 100 removed all noise peaks. Figure 4.4 shows a comparison of the raw data signal and the 20 and 100-sample windows. Other sample window sizes minimized some noise peaks but did not eliminate any peaks. A sample window of 500 was chosen to reduce signal noise by two decades.

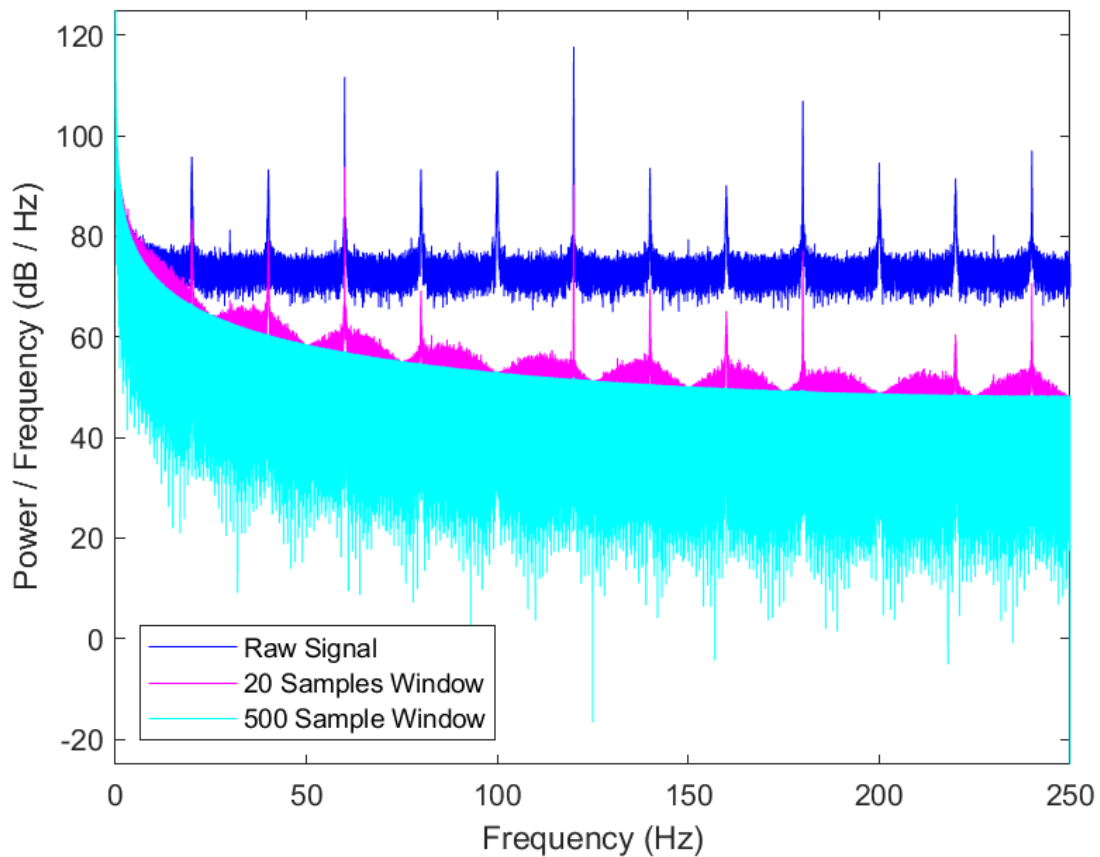


Figure 4.4. Comparison of Raw and Filtered DAQ Signal Noise

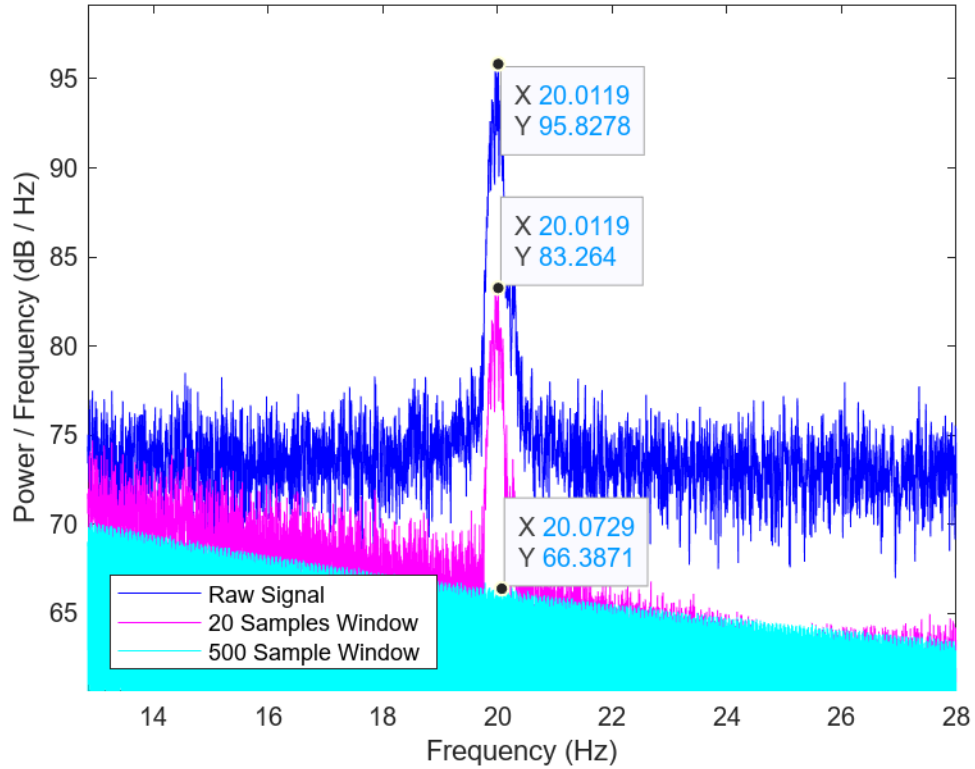


Figure 4.5. Signal Noise Reduction at 20Hz

An experimentally determined error, based on the data processed for this study, was calculated using a statistical analysis. The Gaussian distributions for the filtered and unfiltered data are presented in Fig. 4.6. The statistical analysis showed a DAQ system error of $\pm 0.05\%$. Table 4.2 presents the experimentally determined system error data.

Table 4.2. Experimentally Determined DAQ System Error

Error Magnitude		Percent Error
kPa	psi	%
± 20.6	± 3.0	0.0514

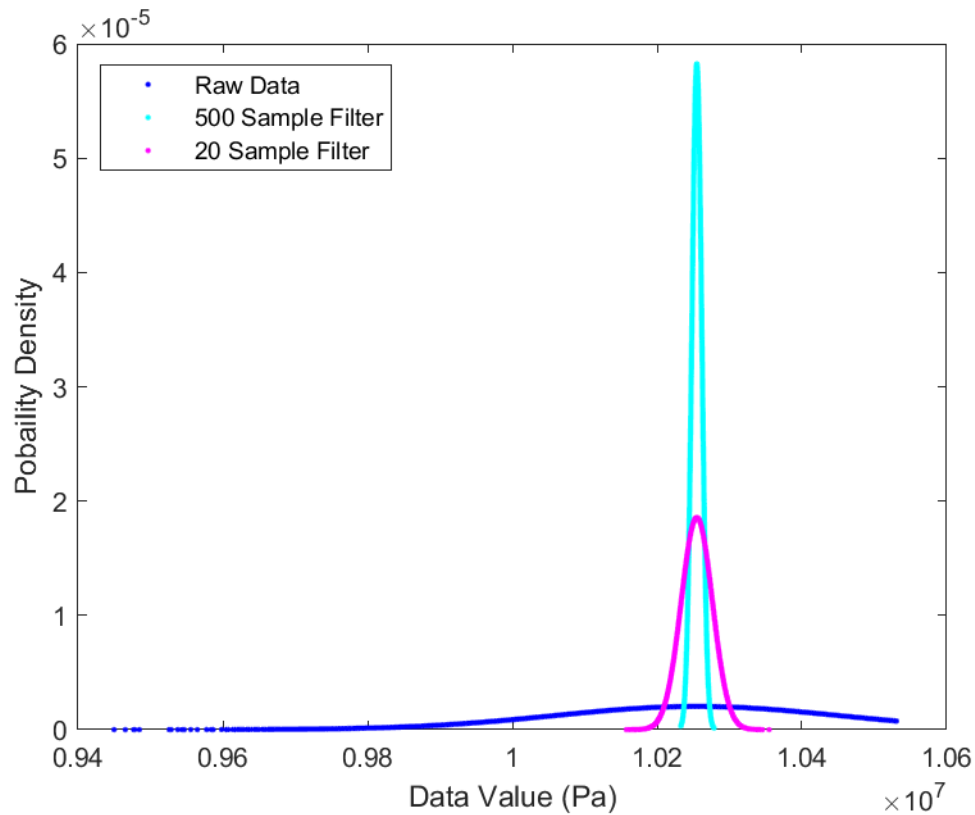


Figure 4.6. Gaussian Distributions for Experimentally Obtained Data Error

4.1.3 Pressure System Testing

To evaluate the effectiveness of hollow parts produced by LMJP at substantial sea depth, the porosity of these parts needed to be tested. In order to evaluate leak-by of the parts, the nominal system response of the pressure test rig was evaluated. This evaluation was conducted by series of 15 minute hold at 6.89×10^5 Pa (100 psi) intervals to measure pressure drop due to system losses. The data collected from the pressure holds was evaluated in Matlab (see appendix D for code) to determine the nominal system response.

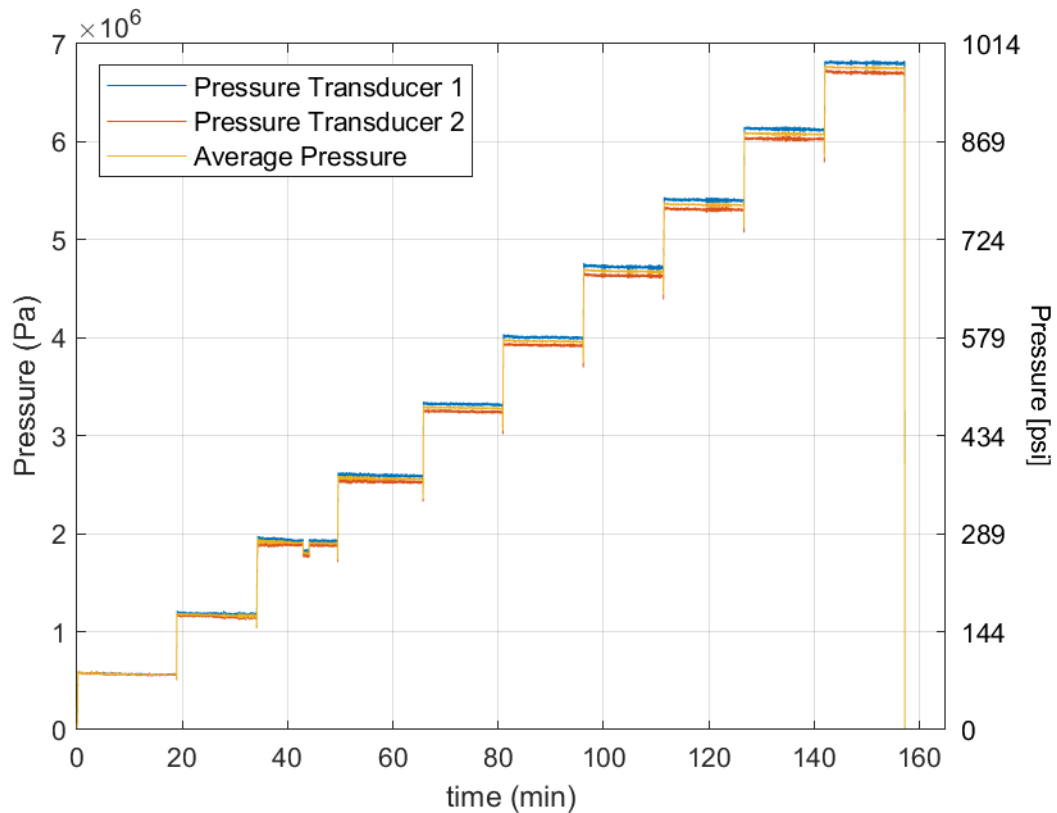


Figure 4.7. PT1, PT2, and Average Pressure from 0 to 6.9×10^6 Pa (1000 psi)

Figure 4.7 shows a comparison of the two PTs and average pressure against time from 0 to 6.89×10^6 Pa (1000 psi). PT1 and PT2 show diverging readings as pressure increases but are in agreement for pressure trend. This suggests a gain difference between the two that should be examined in further work with this test apparatus. For the purposes of this work and to err conservatively, PT2 was selected for all follow on calculations. Sharp pressure drops at the end of each interval were due to opening the isolation valve between the pressure source (hand pump) and the vessel. The cause for the brief pressure drop during the third interval is unknown and suspected to be due to an instrumentation anomaly.

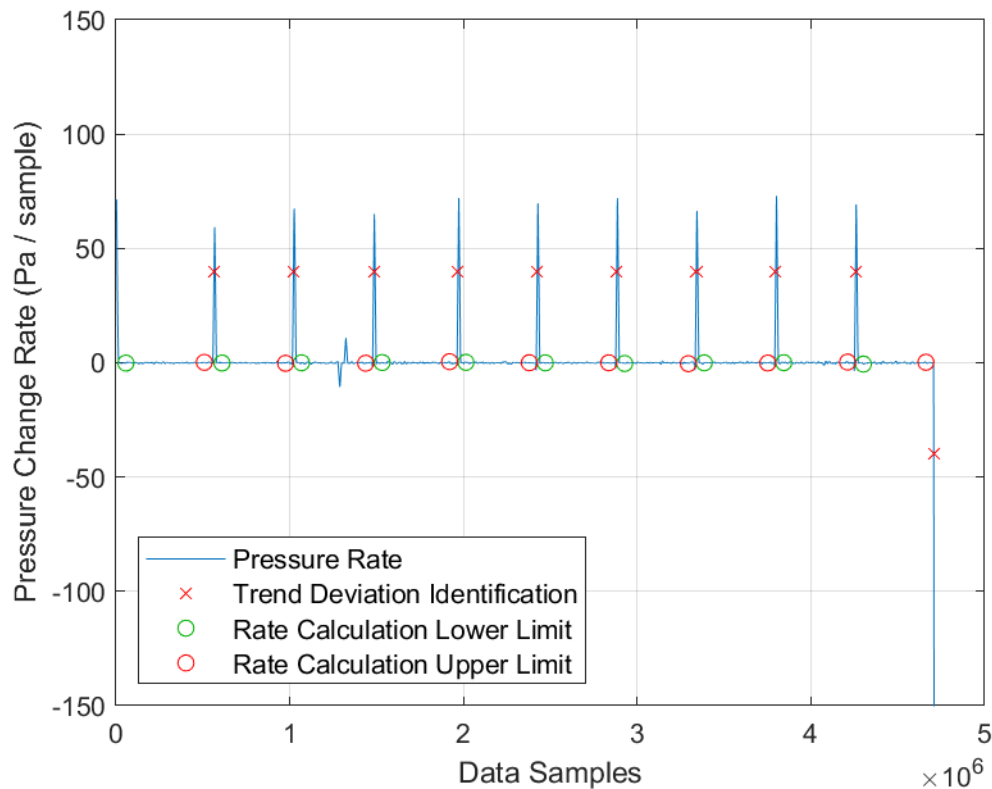


Figure 4.8. Pressure Rate Response from 0 to 6.9×10^6 Pa (1000 psi)

Figure 4.8 shows the rate change of pressure against time and shows the selected data points for loss rate calculations from 0 to 6.89×10^6 Pa (1000 psi). Pressure rate was measured against data samples to easily identify the points where the pressure being tested was changed. The identified trend deviation points were then used to select endpoints for the loss rate calculation at that pressure. Loss rate was calculated linearly using end points selected at 10 and 90 percent of the time tested. This range of data was adequately representative of the loss at the selected pressure and allowed for easy automation of end point selection. Figure 4.9 shows the selected trend analysis data points overlaid with pressure response from 0 to 6.89×10^6 Pa (1000 psi). Additional figures for testing data up to 1.93×10^7 Pa (2800 psi) can be found in appendix E.

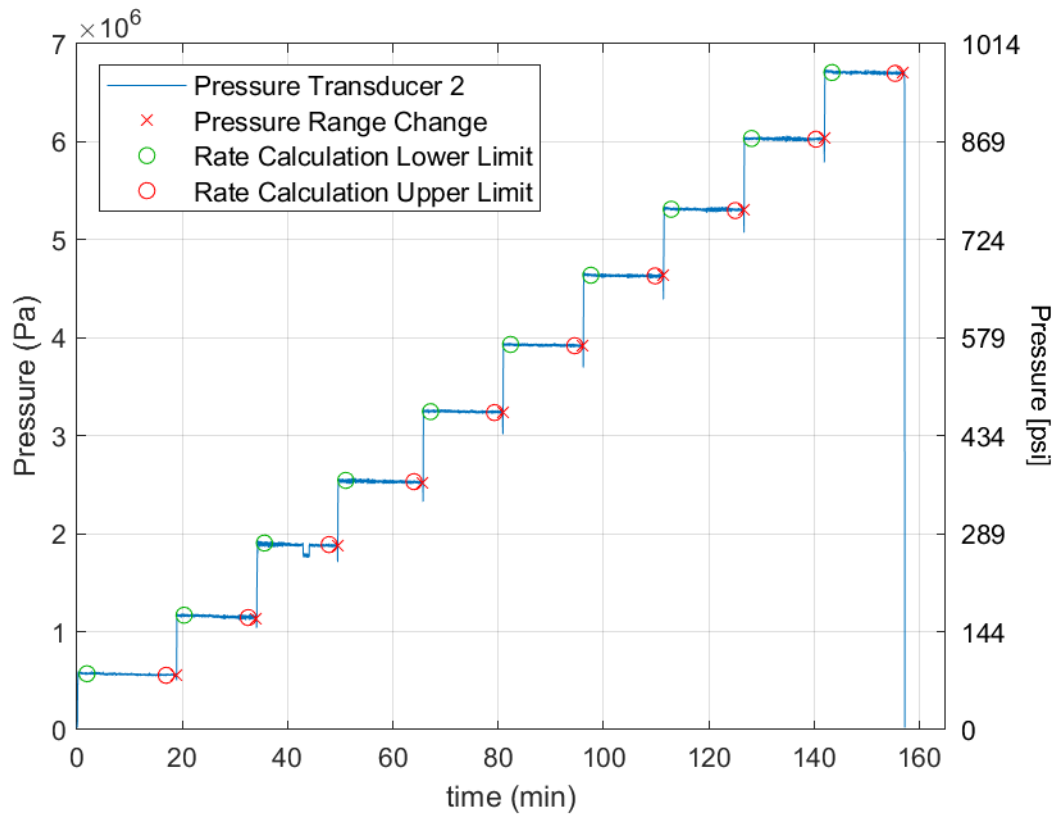


Figure 4.9. Pressure vs. Time from 0 to 6.9×10^6 Pa (1000 psi)

The pressure loss rate for each hold was plotted against the pressure at the start of the hold and fit with a linear regression as shown in Fig. 4.10. The average pressure lost during each hold and the average loss rate are presented in table 4.3 and were shown to be minimal compared to the length of the experiments. The Matlab code for these calculations is presented in appendix F.

Table 4.3. Average System Pressure Losses from 0 to 2.0×10^7 Pa (2800 psi)

Average Pressure Loss		Average Loss Rate	
kPa	psi	kPa / min	psi / min
-18.8	-2.7	-1.5	-0.2

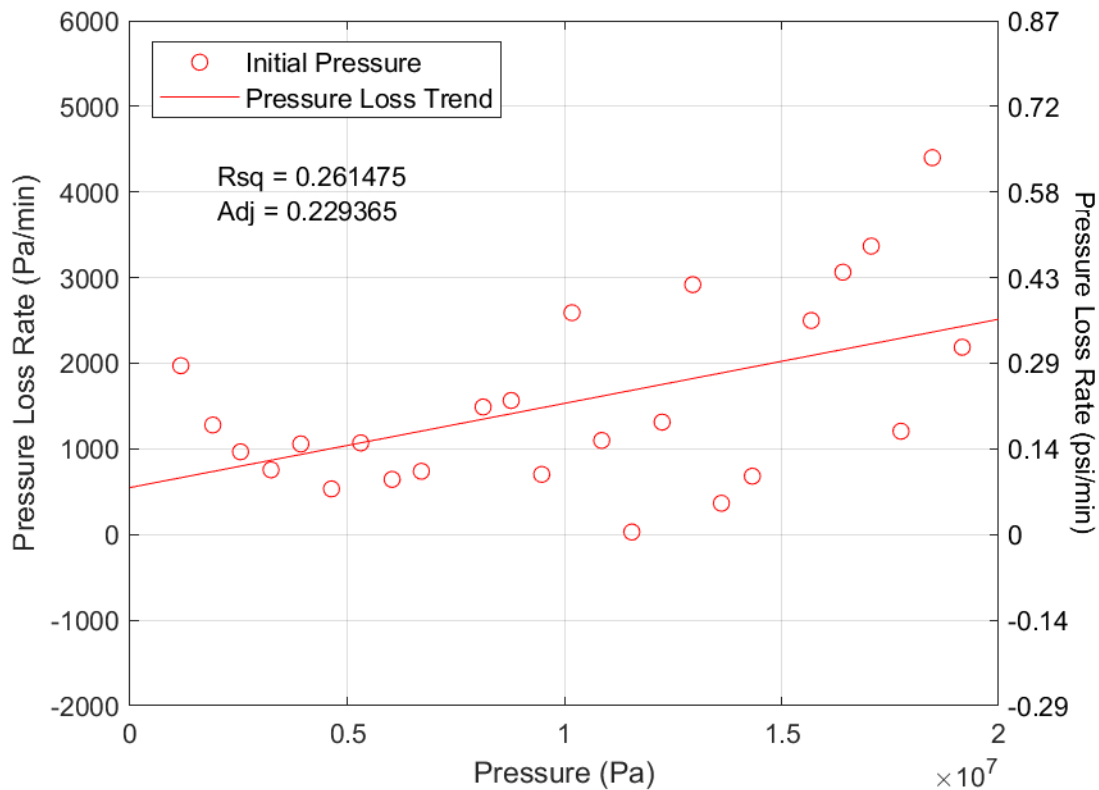


Figure 4.10. Pressure Loss vs. Pressure (0 to 2.0×10^7 Pa (2800 psi))

4.2 Buoyant Part Testing

The hollow parts were evaluated by series of 2 minute holds at 6.89×10^5 Pa (100 psi) intervals until part failure was indicated by a rapid pressure drop (total failure) or an inability to increase pressure (leak by). Table 4.4 lists the number of each variant tested by wall thickness.

Table 4.4. Quantity of Variants Tested by Wall Thickness

Thickness (mm)	Number Tested	
	Variant 1	Variant 2
1.0	5	
1.5	5	
2.0	8	1
3.0		5

4.2.1 Testing: Buoyant Design Variant 1

The test results for one of the 2 mm samples of variant 1 are shown in Fig. 4.11. Max pressure readings are presented in table 4.5. Total pressure loss and pressure loss rates for each interval are presented in table 4.6.

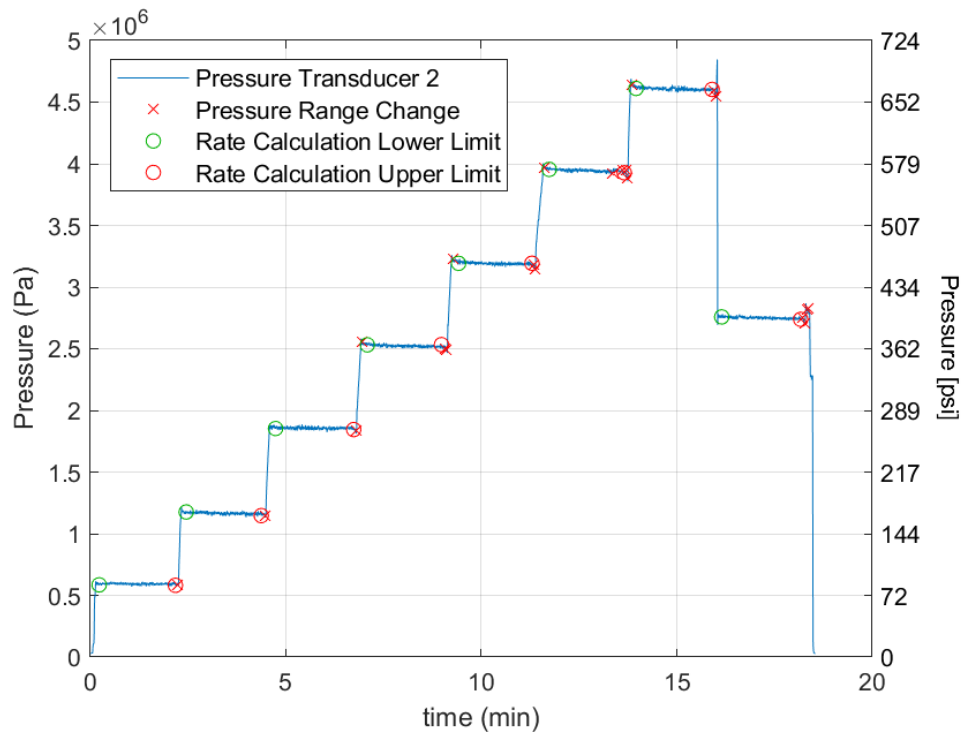


Figure 4.11. Pressure vs. Time for a 2 mm Thick Variant 1 Design

Buckling of the test piece occurred at a max pressure of 4.85×10^3 kPa (702.6 psi) which equates to 482 meters of seawater (1577 feet of seawater). Pressure loss and rates were mostly consistent with the measured system response. Intervals 2 and 6 were significantly higher than the other losses. This was most likely due to the instability of the PT instrument signal coupled with the data point selection resulting in the calculation of a higher-than-actual loss. The final interval is the pressure at which the test piece reached stability after buckling. The loss and rate at for this interval is also consistent with the system response indicating that buckling did not compromise the water tight integrity of the piece. After a 2 minute verification, an attempt to increase pressure was made. Despite continued pumping, pressure rose only slightly before rapidly dropping indicating failure of the pressure boundary.

Table 4.5. Max Pressure: 2 mm Variant 1 Test Sample

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of seawater	Feet of seawater
PT1	4937.5	715.8	491	1607
PT2	4846.0	702.6	482	1577
Avg	4890.3	709.0	487	1592

Table 4.6. Pressure Loss: 2 mm Variant 1 Test Sample

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-3.8	-0.6	-2.0	-0.3
2	-29.6	-4.3	-15.4	-2.2
3	-8.4	-1.2	-4.2	-0.6
4	1.7	0.3	0.9	0.1
5	0.8	0.1	0.4	0.1
6	-27.7	-4.0	-14.4	-2.1
7	-8.1	-1.2	-4.2	-0.6
8	-19.7	-2.9	-9.7	-1.4
Avg	-10.7	-1.6	-5.5	-0.8

4.2.2 Testing: Buoyant Design Variant 2

The test results for one of the 3 mm samples of variant 2 are shown in Fig. 4.12. Max pressure readings are presented in table 4.7. Total pressure loss and pressure loss rates for each interval are presented in table 4.8.

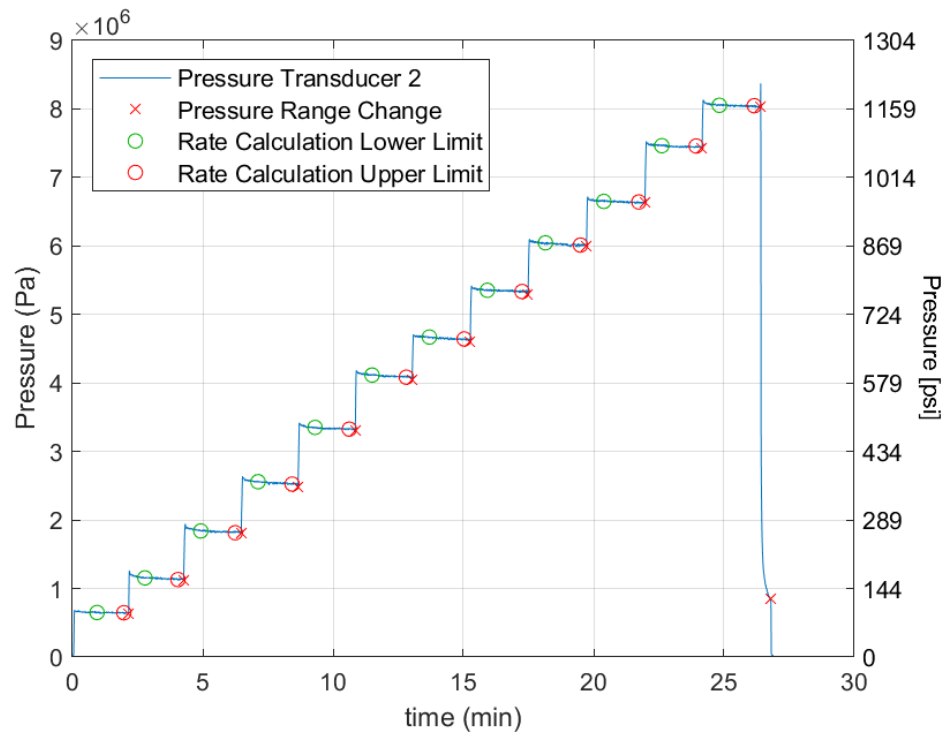


Figure 4.12. Pressure vs. Time for a 3 mm Thick Variant 2 Design

Buckling occurred at a max pressure of 8.37×10^3 kPa (1213.8 psi) which equates to 833 meters of seawater (2825 feet of seawater). Buckling compromised the interface between the main body and the cap causing leak-by past the gasket, resulting in failure of the pressure boundary. Pressure loss and rates were 2x to 3x higher than that of the measured system response. The cause for the increased leak rate has three suspected causes. The first is that a flow path existed around the torx screws and past the gasket into the main body. The second is that, due to the limitations of the LMJP process, the mating surfaces of the main body and the end cap have to be the first layer printed. This results in smooth mating surfaces against the gasket which, potentially, offers little resistance to inward flow. The third possible

cause is that, compared to variant 1, variant 2 has significant cylindrical length. As pressure increased, the cylindrical portion could collapse slowly prior to buckling, resulting in the higher pressure loss rates.

Table 4.7. Max Pressure: 3 mm Variant 2 Test Sample

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of seawater	Feet of seawater
PT1	8499.5	1232.2	846	2766
PT2	8372.4	1213.8	833	2725
Avg	8435.3	1222.9	839	2745

Table 4.8. Pressure Loss: 3 mm Variant 2 Test Sample

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-3.3	-0.5	-3.2	-0.5
2	-24.8	-3.6	-19.5	-2.8
3	-25.8	-3.7	-19.5	-2.8
4	-34.5	-5.0	-26.4	-3.8
5	-24.9	-3.6	-19.0	-2.8
6	-30.8	-4.5	-23.4	-3.4
7	-27.7	-4.0	-20.8	-3.0
8	-17.8	-2.6	-13.4	-1.9
9	-33.0	-4.8	-24.5	-3.6
10	-8.4	-1.2	-6.2	-0.9
11	-4.5	-0.7	-3.4	-0.5
12	-6.0	-0.9	-4.5	-0.7
Avg	-20.1	-2.9	-15.3	-2.2

4.2.3 Performance Summary

Tables 4.9 and 4.10 summarize the performance of variants 1 and 2. Additional figures and tables for each sample of part variant 1 are presented in appendix G and each sample of part variant 2 in appendix H .

Table 4.9. Performance Summary: Max Pressure

Average Max Pressure			Average Depth	
Variant 1				
	kPa	psi	Meters of seawater	Feet of seawater
1.0 mm	670.3	97.2	67	218
1.5 mm	2718.0	394.0	270	885
2.0 mm	4074.6	590.7	405	1326

Variant 2				
	kPa	psi	Meters of seawater	Feet of seawater
2.0 mm	4295.9	622.8	427	1398
3.0 mm	5439.2	788.5	541	1770

Table 4.10. Performance Summary: Pressure Loss

Average Pressure Loss			Average Loss Rate	
Variant 1				
	kPa	psi	kPa / min	psi / min
1.0 mm	-10.0	-1.5	-11.0	-2.0
1.5 mm	-29.2	-4.2	-18.4	-2.7
2.0 mm	-15.0	-2.2	-9.6	-1.4

Variant 2				
	kPa	psi	kPa / min	psi / min
2.0 mm	-150.1	-21.8	-114.1	-16.5
3.0 mm	-63.9	-9.3	-57.3	-8.3

The distribution for max pressure performance followed as expected for variant 1, but was indeterminate for variant 1. Generally, an increase in wall thickness corresponds to an increase in the max pressure tolerated. This trend was apparent in variant 1, but could only be inferred for variant 2 based on the averaged max pressure of variant 2. A scatter plot of max pressure distribution for variants 1 and 2 is presented in Fig. 4.13. The distribution error between variant 2 and variant 1 suggests that a significant source of inconsistency is introduced by the two-piece design.

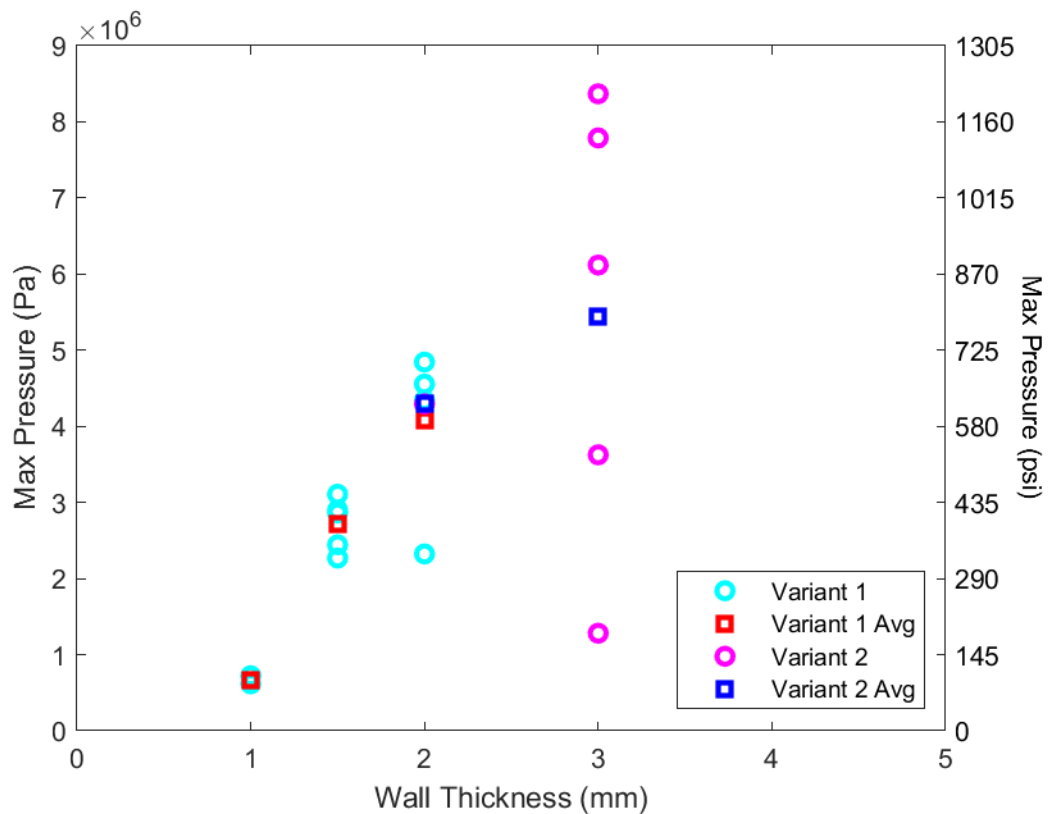


Figure 4.13. Max Pressure vs. Wall Thickness

The distribution for pressure loss for variants 1 and 2 are presented in Fig. 4.14. Variant 1 pressure losses indicate a very weak correlation between wall thickness and loss rate. It is likely that additional tests with a wider range of wall thicknesses would reveal inconsequential correlation. This result for the single-piece design suggests that hollow parts produced by LMJP have negligible, if any, porosity. Losses for variant 2 show a strong correlation, but an insufficient number of test samples for 2 mm wall thickness and too narrow of a range of wall thicknesses were tested to draw any meaningful conclusions.

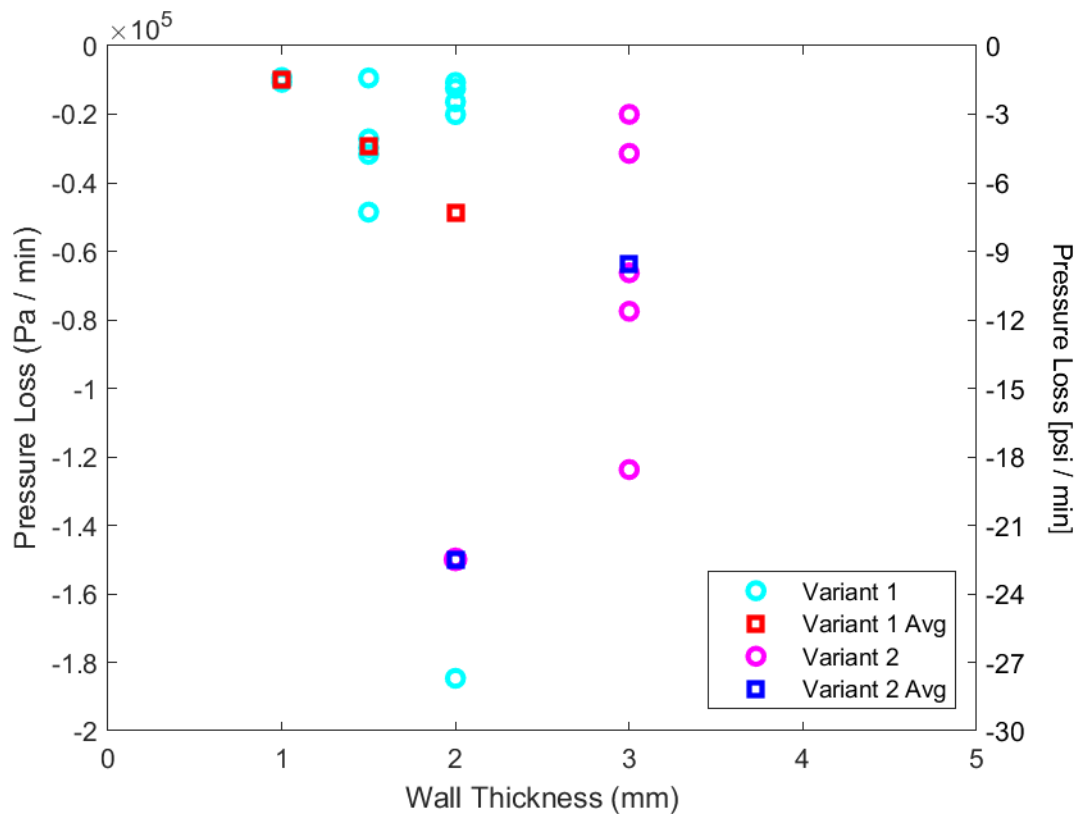


Figure 4.14. Pressure Loss vs. Wall Thickness

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CHAPTER 5:

Results and Conclusions

5.1 Discussion of Results

The actual mechanism for failure of the pressure boundary for each design varied. Variant 1 failed on the bottom side near the mounting holes. As the bottom of variant 1 collapsed inward, the flat portion of the mounting surface created a stress concentration area that exceeded the tensile strength of the aluminum and formed a leak path. The location and mode of experimental failure for variant 1 matched the predicted location of max principle stress and the inferred method of failure discussed in Section 3.2.8. Additionally, the non-linear FEA accurately predicted the buckling shape of variant 1. Results for variant 1 are shown in Fig. 5.1.

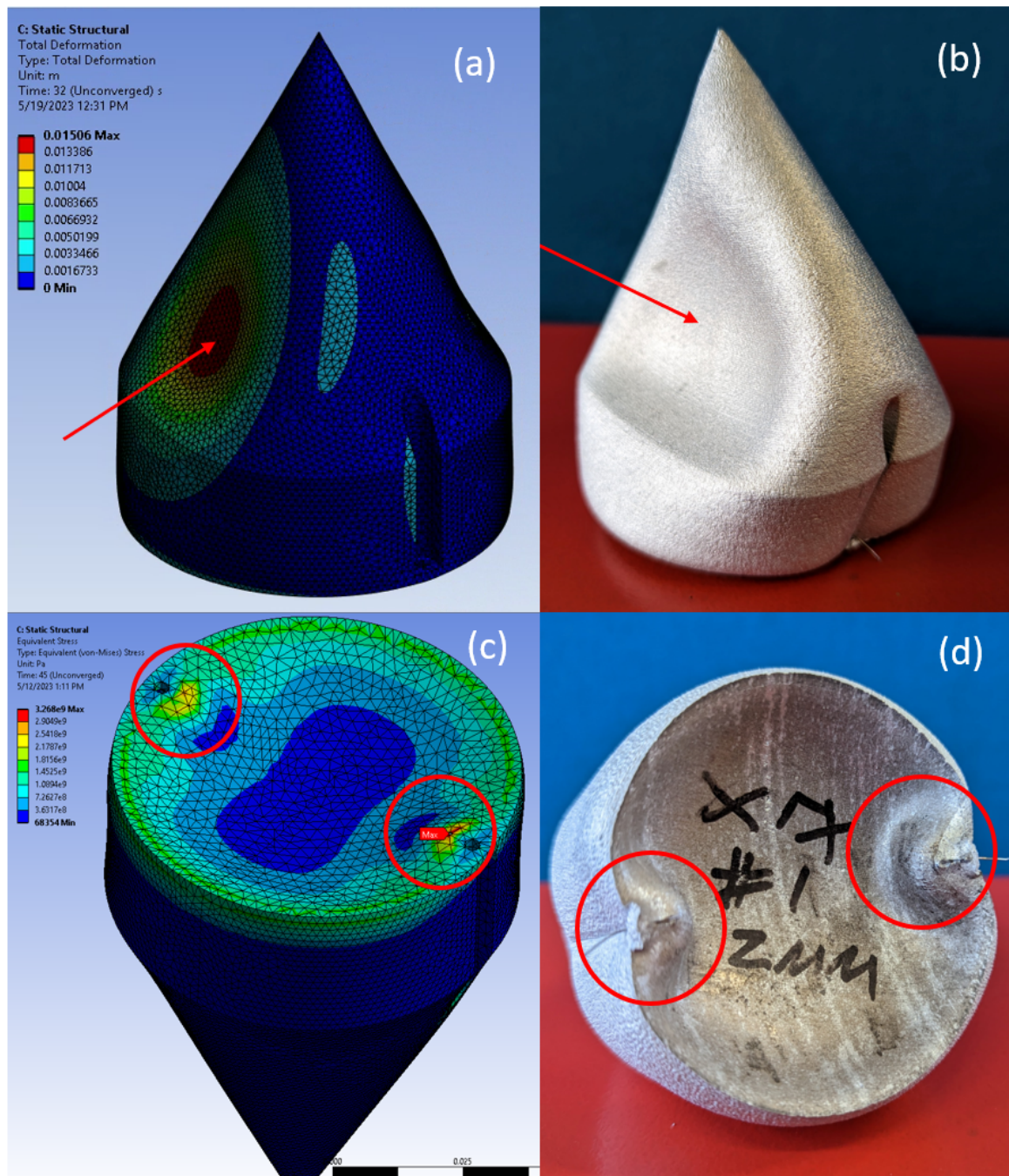


Figure 5.1. Comparison of Predicted Buckling (a), Experimental Buckling (b), Predicted Area of Max Stress (c), and Area of Failure (d) for Variant 1

Variant 2 had two suspected failure mechanisms. The first was suspected leak-by at either the gasket or screws, as discussed in Section 4.2.2. Longer interval hold testing is required to see if the pressure loss would go to zero which would indicate that the pressure losses observed before failure were due to dimensional loss under compression. The second mechanism suspected is that when the cylindrical portion of the main body buckled, the mating surface with the gasket deformed such that it introduced a leak path into the main body. The max principle stress predicted by the non-linear FEA occurred around the cone-cylinder transition of the main body. It was inferred that the stress developed in this location because of the buckling of the cylindrical portion of the main body. The experimental results suggest that this was a correct assumption except that stress accumulated at the interface between the main body and the end cap. The interface end of the main body is not as rigidly fixed as the other end therefore deformation occurred more easily, creating a leak path. Figure 5.2 shows the a comparison of the predicted buckling and the experimental buckling. Figure 5.3 shows a comparison of the predicted area of max stress and the suspected area of failure from experimentation.

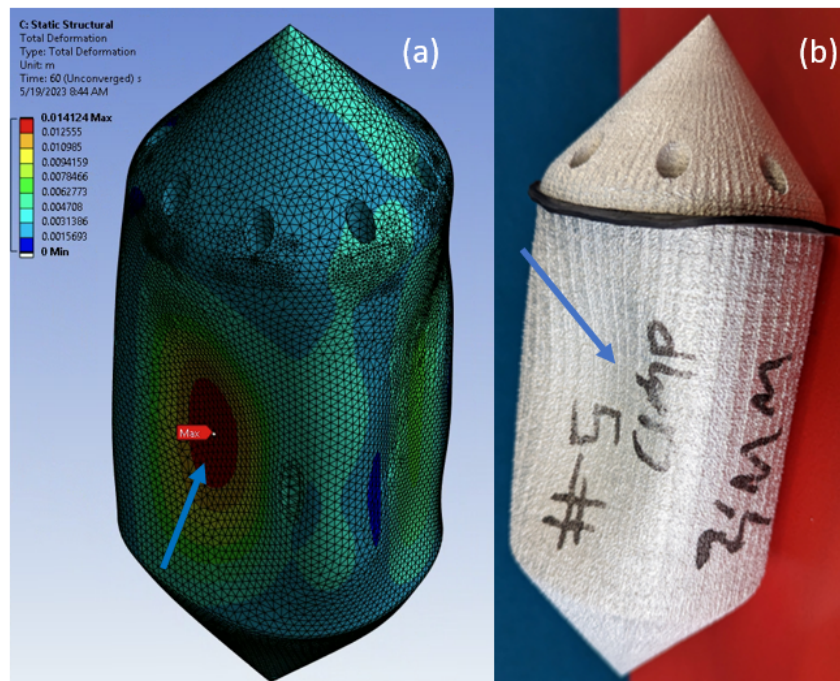


Figure 5.2. Comparison of Predicted Buckling (a) and Experimental Buckling (b) for Variant 2

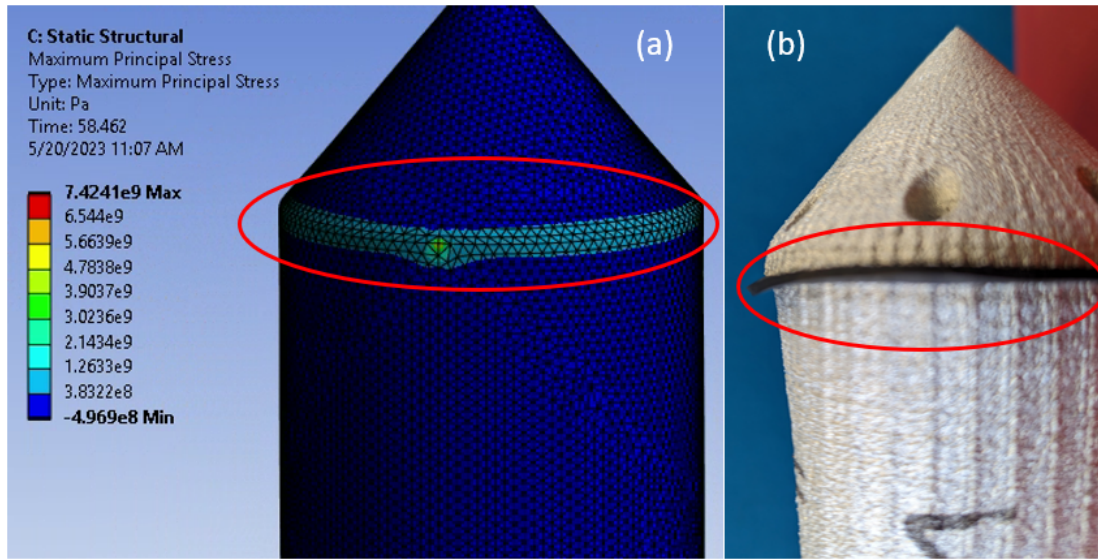


Figure 5.3. Comparison of Predicted Area of Max Stress (a) and Area of Failure (b) for Variant 2

The experimental failure loading for both variants was several orders of magnitude higher than that predicted by their respective non-linear FEAs. A comparison of the predictions and results is presented in table 5.1. This was likely due to using the default settings for aluminum alloy in Ansys. It is likely the LMJP process alters the mechanical properties of aluminum and introduces anisotropic properties that are not adequately modeled by Ansys.

Table 5.1. Comparison of Failure Loading for Variants 1 and 2

Thickness (mm)	Predicted kPa (psi)	Experimental kPa (psi)	Difference kPa (psi)	Percent Error %
Variant 1				
1.0	1.03×10^7 (1.49×10^6)	670.3 (97.2)	10299.3 (1493.8)	99.99
1.5	1.61×10^7 (2.34×10^6)	2718.0 (394.0)	16097.3 (2334.7)	99.98
2.0	2.10×10^7 (3.05×10^6)	4074.6 (590.7)	20995.9 (3045.2)	99.98
Variant 2				
2.0	7.61×10^6 (1.10×10^6)	4295.9 (622.8)	7605.7 (1103.1)	99.94
3.0	3.94×10^7 (5.71×10^6)	5439.2 (788.5)	39394.6 (5713.7)	99.97

5.2 Conclusions

This work reports the results of a non-linear buckling analysis of two variants of hollow, buoyant metal parts produced by LMJP. The experimental results from hydrostatic testing of these parts is also presented. A comparison of these results shows that the areas of max stress predicted in Ansys Mechanical Workbench FEA correspond to the location of experimental failure of the variants tested with one exception. The two-piece design variant was predicted to fail at the cone-cylinder transition of the main body due to buckling of the cylindrical region. Instead, the buckling of the cylindrical region caused decoupling of the mating surface of the main housing and the end cap, the other cone-cylinder transition. Additional boundary conditions for the two-piece FEA could yield a more accurate location of failure. Ansys Mechanical Workbench was accurate in predicted the buckling shape of both geometries. The magnitude of the predicted load at failure was several orders of magnitude higher than the loading achieved. This is most likely due to unique mechanical properties imparted to the aluminum by LMJP. These results show that the current tools for structural analysis provide valid insight into the behavior of AM parts, but that AM techniques introduce unique material challenges that must be studied further.

5.3 Further Work

The continuation of this work would benefit from the following additional studies. An in-depth analysis of the mechanical properties of the aluminum parts produced by LMJP, specifically, any anisotropic properties should be determined to build an accurate model of the material for Ansys Mechanical. Additionally, a study of how droplet deposition affects density, porosity, and mechanical properties should be investigated.

Continuation of this work should include testing additional wall thicknesses and geometries to define a better correlation between wall thickness and depth survivability and to optimize geometry for survivability and payload capacity. Additional designs with a larger numbers of clamping screws should be evaluated to test if the second theory for leak-by, discussed in Section 5.1, can be defeated or postponed.

Additional methods of AM should be assessed for their use in this application. Specifically, PBF should be examined for manufacturing two-piece designs. The advantages of using a PBF machine include the ability to produce titanium parts which are lighter therefore more

buoyant, stronger therefore could be deployed deeper, and more corrosion resistant therefore could be employed longer than their aluminum counterparts. Additional advantages include being able to manufacture more complicated mating surfaces and finer tolerances. A more torturous mating surface could significantly reduce the likelihood of developing a leak path and the finer tolerances could enable threads allowing for higher clamping torque.

The final recommendation is study on the effects that external pressure have on dimensionality prior to failure. This study would further inform the correlation between geometry and depth survivability and could adjudicate the third theory for variant 2 leak-by discussed in Section 5.1.

APPENDIX A:

Pressure Vessel End Cap Bolt Torque Calculation

The Matlab code used to calculate the torque specification for the pressure vessel end caps is presented in this appendix.

```

ys = 150000;    % psi % min tensile strength of ASTM grade 8 bolts
d = 0.3042;    % inch % min tolerance of 5/15-24
nu = 0.3;      %      % max tol of poisson ration
L = 1.75;      % inch % length of bolts being used
E = 27557000;  % psi % min tol of young's modulus for ASTM grade 8 bolts
De = 3.26;     % inch % diameter of area of end cap exposed to pressure per tech drawing
k = 0.2;       %      % max tol of coefficient of friction of bolt
n = 24;        % threads per inch
D = d-0.9743/n; % inch % diameter correction for force applied to threads

```

```

P_select = 4000 % psi % selected max pressure

```

```

P_select = 4000

```

```

% axial force (lbs) to yield bolt corrected for elongation
Fy = ys*pi*D^2/(4*(1-(4*ys*nu*L/(E*pi*D^2))));

```

```

% axial force (lbs) on bolts from pressure inside vessel
Fp = P_select.*pi*De^2/32;

```

```

% axial force (lbs) on bolt from torque allowed
Ft = Fy-Fp;

```

```

% max torque (t = in-lbs, T = ft-lbs) based on selected max pressure
Max_t = k*Ft*D

```

```

Max_t = 235.4068

```

```

Max_T = Max_t/12

```

```

Max_T = 19.6172

```

```

Wrench_Range_inlbs_MIN_MAX = [Max_t/0.9 Max_t/0.2]

```

```

Wrench_Range_inlbs_MIN_MAX = 1x2
10^3 ×
    0.2616    1.1770

```

```

Wrench_Range_ftlbs_MIN_MAX = [Max_T/0.9 Max_T/0.2]

```

```

Wrench_Range_ftlbs_MIN_MAX = 1x2
    21.7969    98.0862

```

```

% calculate max pressure (psi) based on selected torque
t = 200

```

```

t = 200

```

```

T = t/12

```

```

T = 16.6667

```

```
ft = t/(k*D);
fp = Fy-ft;
max_P = fp*32/(pi*De^2)
```

```
max_P = 4.6437e+03
```

```
% Torque Spec (t = in-lbs, T = ft-lbs)
t_spec = mean([Max_t t])
```

```
t_spec = 217.7034
```

```
t_tol = [Max_t-t_spec t_spec-Max_t]
```

```
t_tol = 1x2
    17.7034   -17.7034
```

```
T_spec = mean([Max_T T])
```

```
T_spec = 18.1419
```

```
T_tol = [Max_T-T_spec T_spec-Max_T]
```

```
T_tol = 1x2
    1.4753   -1.4753
```

```
% Torque Spec (N-m)
t_met = t_spec * 0.112984825
```

```
t_met = 24.5972
```

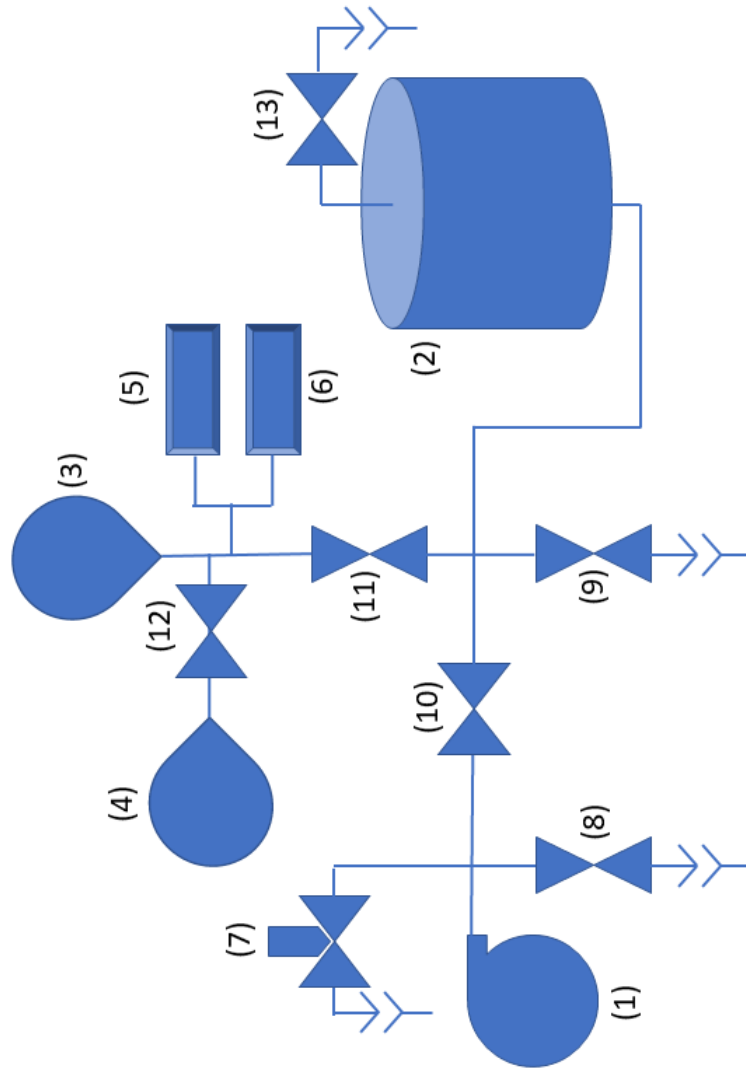
```
met_tol = t_tol * 0.112984825
```

```
met_tol = 1x2
    2.0002   -2.0002
```

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APPENDIX B: Pressure Test Rig Code Block

The code block for the pressure test rig is presented in this appendix.



Function		Pressure Rating (ksi)
1	Hand Pump	6
2	Pressure Vessel	4 (tested)
3	PG-1 Pressure Gage (0-4000 psi)	6
4	PG-2 Pressure Gage (0-1000 psi)	6
5	PT-1 Pressure Transducer	10.8
6	PT-2 Pressure Transducer	10.8
7	PV-1 Relief Valve	3.3 (set)
8	PV-2 Pump Pressure Release Valve	10
9	PV-3 Vessel Pressure Release Valve	10
10	PV-4 Pump/Vessel Isolation Valve	10
11	PV-5 Instrumentation Isolation Valve	10
12	PV-6 PG-2 Isolation Valve	10
13	PV-7 Vessel Vent	10

Max Operating Pressure	Thread Type	Tube Size	Bolt Torque
3300 psi	NPT	0.25 inch	218±17 in-lbs

Figure B.1. NI LabView GUI

APPENDIX C: Data Acquisition

The NI LabView block diagram and GUI used for DAQ are presented in this appendix.

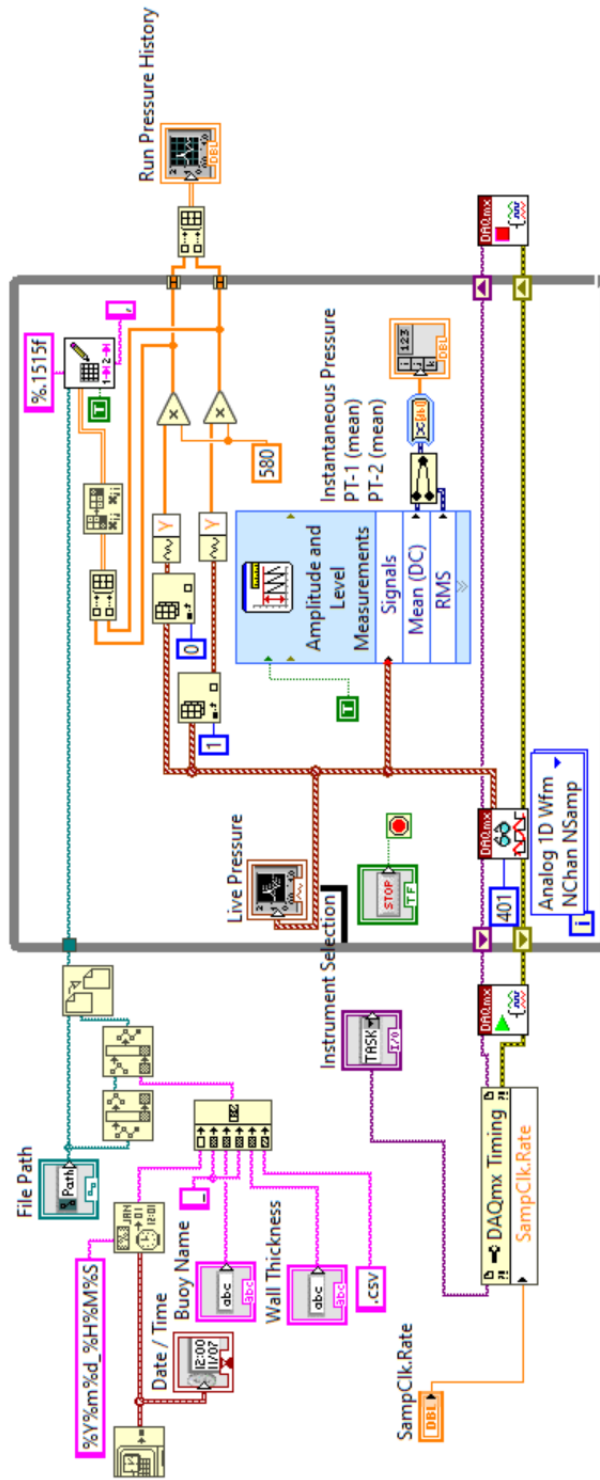


Figure C.1. NI LabView Block Diagram for DAQ

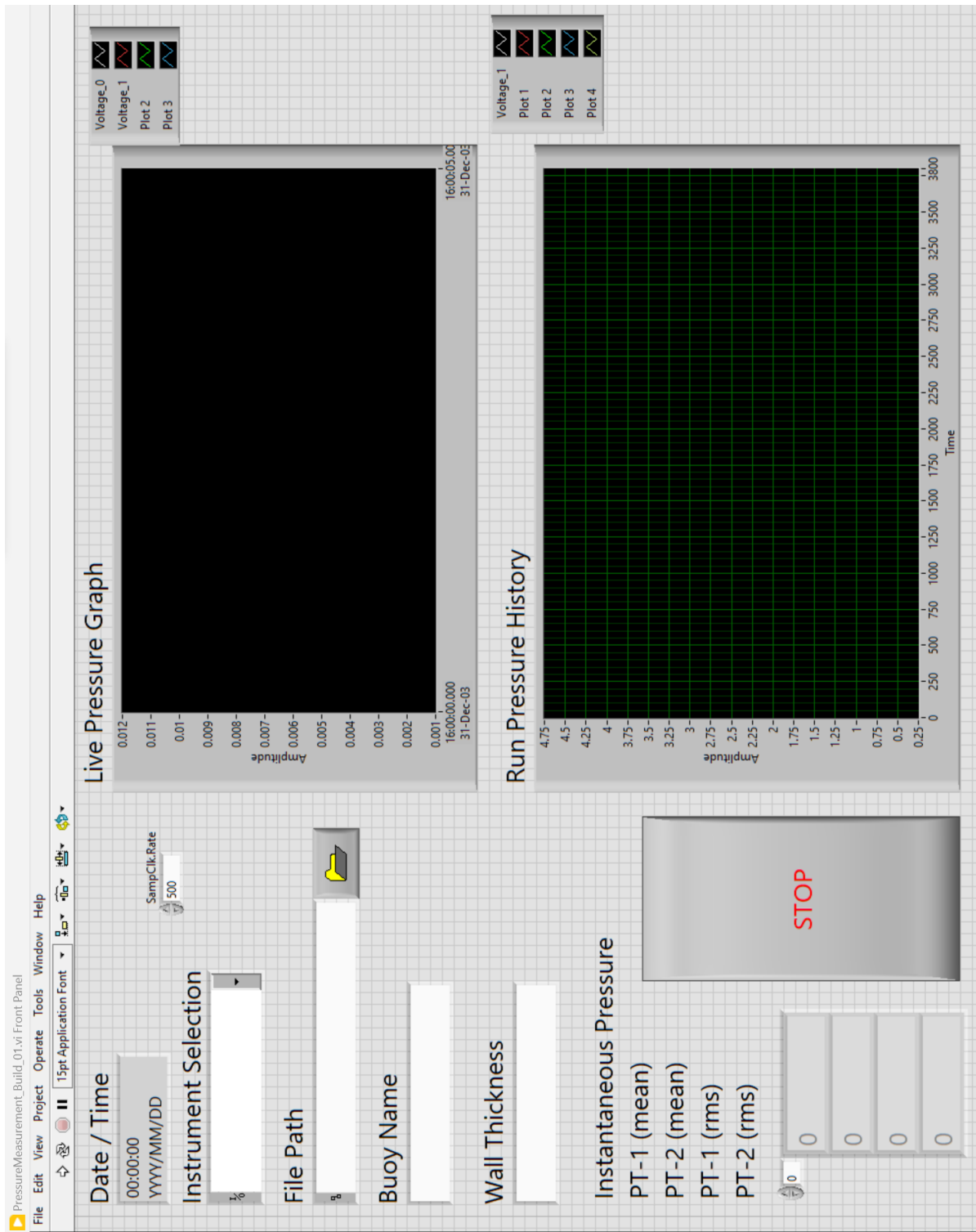


Figure C.2. NI LabView GUI

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APPENDIX D: Data Processing Code

The Matlab code used to process experimental data from the system pressure response evaluation and the buoyant part testing is presented in this appendix.

```

% Clear everything
close all;
clear all;
clc;
clf;
format short g

% Read the File & Declare Data Sampling Rate
pt = csvread("20230424_073117_hold_1k.csv",1,0);
samplerate = 500;

% Comb the Desert
psi2pa = 6897.76;
psi1 = pt(:,1);
psi2 = pt(:,2);
pt1 = pt(:,1)*psi2pa;
pt2 = pt(:,2)*psi2pa;
pta = (pt1+pt2)/2;
samples_averaged = floor(samplerate/2)+1;
pt1a = movmean(pt1,samples_averaged);
pt2a = movmean(pt2,samples_averaged);
ptf = movmean(pta,samples_averaged);

% Write PT data o .csv
% writematrix(pt1a,'pt1a_hold_1k.csv')
% writematrix(pt2a,'pt2a_hold_1k.csv')
% writematrix(ptf,'ptf_hold_1k.csv')

% Make time variable
n1 = numel(pt1);
dt = 1/(samplerate*60);
t = (1:n1)*dt;

```

```

% Identify Points of Change
sa = 10000;
windowSize = sa;
b = (1/windowSize)*ones(1,windowSize);
a = 1;
y = filtfilt(b,a,pt2a);

dptf = diff(pt2a);
dy = diff(y);
dyx = 1:length(dy);

dev = 40;
ij = find(abs(dy)>dev);
ijz = zeros(size(ij));
for iii = 2:length(ij)
    if ij(iii)-ij(iii-1)>1
        ijz(iii) = 1;
    end
end
ij(ijz==0)=[];

```

```

% Calculate Local Slopes
slopes = zeros(numel(ij),1);
lim = zeros(numel(ij),2);
plo = zeros(numel(ij),1);
phi = zeros(numel(ij),1);
tols = zeros(numel(ij),1);
tol = 0.8;
n = 1;
while n <= numel(ij)
    if n-1==0
        b = 0;
    else
        b = ij(n-1);
    end
    a = ij(n);
    tols(n) = a-b;
    phi(n) = floor(ij(n)-tols(n)*0.1);
    plo(n) = floor(ij(n)-tols(n)*0.9);
    slopes(n) = (pt2a(phi(n))-pt2a(plo(n)))/(phi(n)-plo(n));
    lim(n,2) = phi(n);
    lim(n,1) = plo(n);
    n = n+1;
end

```

```

loss_abs = pt2a(phi)-pt2a(plo);
loss_rate = slopes./dt;
limits = lim;
lo = lim(:,1);
hi = lim(:,2);

```

```

% List of Figures

```

```

% PT1 vs PT2 vs Avg
% Zero-Phase Digital Filter
% Data Deviance
% Pressure vs Time
% Pressure Loss Rate vs Pressure

```

```

% Common Figure Properties
fs = 10; % font size

```

```

% Figure: PT1 vs PT2 vs Avg

```

```

fig_pt = figure;

hpt1 = plot(t,pt1a);
hold all
hpt2 = plot(t,pt2a);
hptf = plot(t,ptf);
hold off

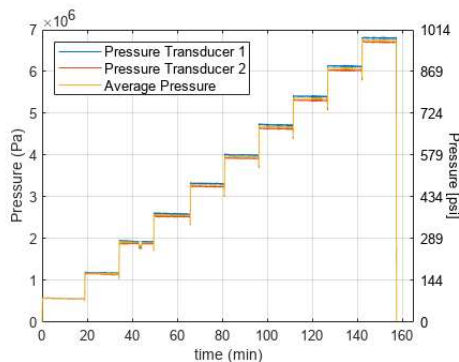
haPT = (gca);
haPT.YAxisLocation = 'left';
haPT.YTick;
for iii = 1:length(haPT.YTick)
    text(1.05*diff(xlim),haPT.YTick(iii),num2str(floor(haPT.YTick(iii)/psi2pa)), 'HorizontalAlignment', 'left')
end

box on;
grid on;
xlabel('time (min)')
xlim([0 165])
ylabel('Pressure (Pa)')
%ylim()
h1PT = legend('show');

set(h1PT,'location','northwest','fontsize',fs)
set(hpt1,'displayname','Pressure Transducer 1')
set(hpt2,'displayname','Pressure Transducer 2')
set(hptf,'displayname','Average Pressure')
set(gca,'fontsize',fs);

text(max(haPT.XLim)*1.11, mean(haPT.YLim), 'Pressure [psi]', 'Rotation', -90, 'HorizontalAlignment', 'center')

```



```

% Figure: Data Deviance

```

```

fig_dev = figure;

hdy = plot(dy);
hold all
hvar = plot(dyx(ij),dy(ij),'xr');
hlo = plot(dyx(lo),dy(lo),'og'); set(hlo,'color',[ .0 0.7 .0])
hhi = plot(dyx(hi),dy(hi),'or');
hold off

box on;
grid on

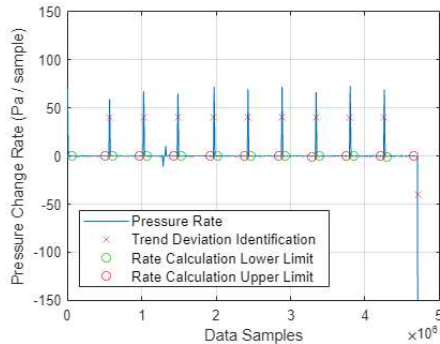
```

```

xlabel('Data Samples')
%xlim()
ylabel('Pressure Change Rate (Pa / sample)')
ylim([-150 150])
hlDEV = legend('show');

set(hlDEV,'location','southwest','fontsize',fs)
set(hdy,'displayname','Pressure Rate')
set(hvar,'displayname','Trend Deviation Identification')
set(hlo,'displayname','Rate Calculation Lower Limit')
set(hhi,'displayname','Rate Calculation Upper Limit')
set(gca,'fontsize',fs);

```



Warning: Graphics timeout occurred. To share details of this issue with MathWorks technical support, please include that this is an unresponsive graphics client with your service request.

```

% Figure: Pressure vs Time
fig_pvt = figure;

hp = plot(t,pt2a);
hold on
hd = plot(t(ij),pt2a(ij),'xr');
hsl = plot(t(plo),pt2a(plo),'og'); set(hsl,'color',[.0 0.7 .0])
hsh = plot(t(phi),pt2a(phi),'or');
hold off

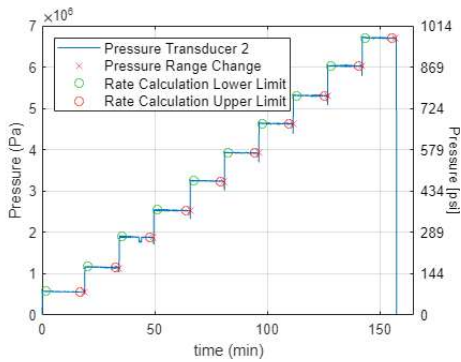
haPVT = (gca);
haPVT.YAxisLocation = 'left';
haPVT.YTick;
for iii = 1:length(haPVT.YTick)
    text(1.05*diff(xlim),haPVT.YTick(iii),num2str(floor(haPVT.YTick(iii)/psi2pa)),'HorizontalAlignment','left')
end

box on;
grid on;
xlabel('time (min)')
xlim([0 165])
ylabel('Pressure (Pa)')
%ylim()
hlPVT = legend('show');

set(hlPVT,'location','northwest','fontsize',fs)
set(hp,'displayname','Pressure Transducer 2')
set(hd,'displayname','Pressure Range Change')
set(hsl,'displayname','Rate Calculation Lower Limit')
set(hsh,'displayname','Rate Calculation Upper Limit')
set(gca,'fontsize',fs);

text(max(haPVT.XLim)*1.11, mean(haPVT.YLim),'Pressure [psi]','Rotation',-90,'HorizontalAlignment','center')

```



```

%% Figure: Pressure Loss Rate vs Pressure
% fig_fit = figure;
%
% % writematrix(pt2a(plo),'pt21k.csv')
% % writematrix(loss_rate,'LR1k.csv')
%
% [bestfit,gof] = fit(pt2a(plo),abs(loss_rate),'poly1')
%
% hip = plot(pt2a(plo),abs(loss_rate),'ro');
% hold all
% hbf = plot(bestfit);
% hold off
%
% % haLR = (gca);
% % haLR.YAxisLocation = 'left';
% % haLR.YTick;
% for iii = 1:length(haLR.YTick)
%     text(1.01*diff(xlim),haLR.YTick(iii),num2str(round(haLR.YTick(iii)/psi2pa,2)), 'HorizontalAlignment','left')
% end
%
% box on;
% grid on;
% xlabel('Pressure (Pa)')
% %xlim()
% ylabel('Pressure Loss Rate (Pa/min)')
% %ylim()
% h1LR = legend('show');
% h1LR = legend('off');
% h1LR = legend('show');
%
% set(h1LR,'location','northeast','fontSize',fs)
% set(hip,'displayname','Initial Pressure')
% set(hbf,'displayname','Pressure Loss Trend')
% set(gca,'fontSize',fs)
%
% text(max(haLR.XLim)*1.11, mean(haLR.YLim),'Pressure Loss Rate [psi/min]','Rotation',-90,'HorizontalAlignment','center')
% text(0.69*diff(xlim),1.0*diff(ylim),sprintf('Rsq = %g\nAdj = %g',gof.rsquare,gof.adjrsquare))

```

```

% Print Figures

```

```

% print(fig_pt,'pt1k','-dpng');
% print(fig_dev,'dev1k','-dpng');
% print(fig_pvt,'pvt1k','-dpng');
% print(fig_fit,'fit1k','-dpng');

```

```

% List of Tables

```

```

% Experimental Duration
% Max Pressure
% Pressure Losses

```

```

% Table: Experiment Duration
t_s = round(n1/samplerate,2);
t_m = round(n1/(samplerate*60),2);
t_h = round(n1/(samplerate*3600),2);
Test_Duration = seconds(t_s);
Test_Duration.Format = 'hh:mm:ss'

```

```

Test_Duration = duration
02:37:17

```

<pre>time_sec_min_hr = [t_s t_m t_h]; TestDuration = array2table(time_sec_min_hr, 'VariableNames', {'Seconds', 'Minutes', 'Hours'})</pre>				
<pre>TestDuration = 1x3 table</pre>				
	Seconds	Minutes	Hours	
1	9437.1	157.29	2.62	
<pre>% Table: Max Pressure MaxPressure = array2table([max(pt1a),max(pt1a)/psi2pa;max(pt2a),max(pt2a)/psi2pa;max(ptf),max(ptf)/psi2pa], 'VariableNames',{'Pa','psi'}, 'f</pre>				
<pre>MaxPressure = 3x2 table</pre>				
	Pa	psi		
1 PT1	6.8365e+06	991.12		
2 PT2	6.7405e+06	977.2		
3 Avg	6.778e+06	982.63		
<pre>%MaxPressure = round(MP,2)</pre>				
<pre>% Table: Pressure Losses PressureLosses = table(loss_abs,loss_abs/psi2pa,loss_rate,loss_rate/psi2pa, 'VariableNames',{'Pa','psi','Pa/min','psi/min'})</pre>				
<pre>PressureLosses = 10x4 table</pre>				
	Pa	psi	Pa/min	psi/min
1	-16213	-2.3504	-1077.1	-0.15615
2	-24031	-3.4838	-1972	-0.28588
3	-15744	-2.2825	-1279.1	-0.18544
4	-12517	-1.8147	-966.42	-0.14011
5	-9192.3	-1.3327	-755.81	-0.10957
6	-12898	-1.8699	-1057.8	-0.15335
7	-6510.8	-0.9439	-533.05	-0.077279
8	-13052	-1.8923	-1070	-0.15512
9	-7890.2	-1.1439	-644.1	-0.093378
10	-8862.9	-1.2849	-738.54	-0.10707
<pre>%PressureLosses = round(PL,2) % writetable(PressureLosses, 'PL1k.csv')</pre>				
<pre>% Tables 2 LaTeX % table2latex(TestDuration, 'TD1k') % table2latex(MaxPressure, 'MP1k') % table2latex(PressureLosses, 'PL1k')</pre>				

APPENDIX E: System Pressure Response Data

Additional figures for system response testing in the ranges from 6.89×10^6 to 1.45×10^7 Pa (1000 to 2100 psi) and 1.45×10^7 to 1.93×10^7 Pa (2100 to 2800 psi) is presented in this appendix.

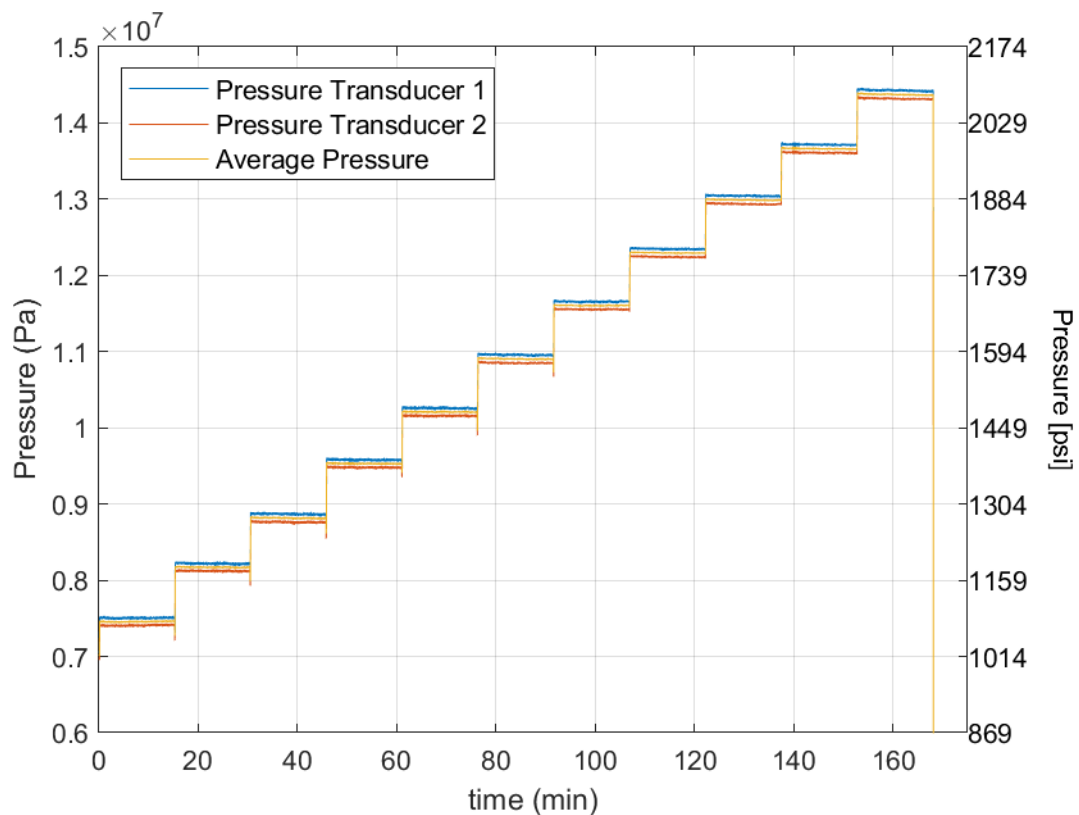


Figure E.1. PT1, PT2, & Average Pressure from 6.89×10^6 to 1.45×10^7 Pa (1000 to 2100 psi)

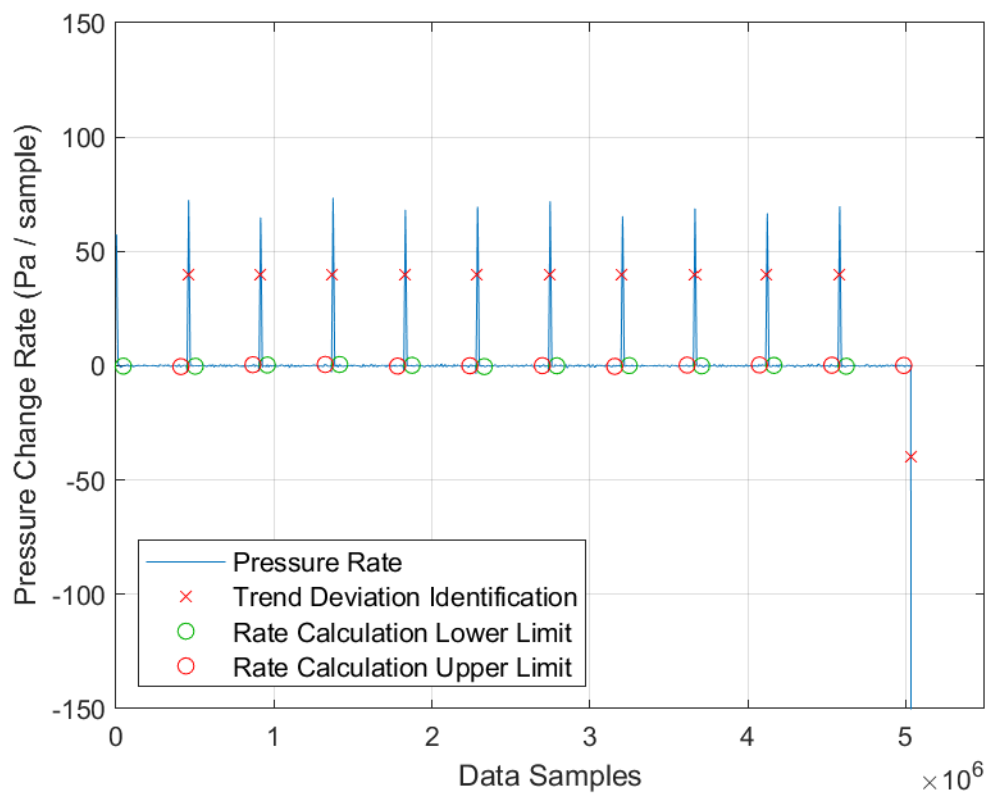


Figure E.2. Pressure Rate Response from 6.89×10^6 to 1.45×10^7 Pa (1000 to 2100 psi)

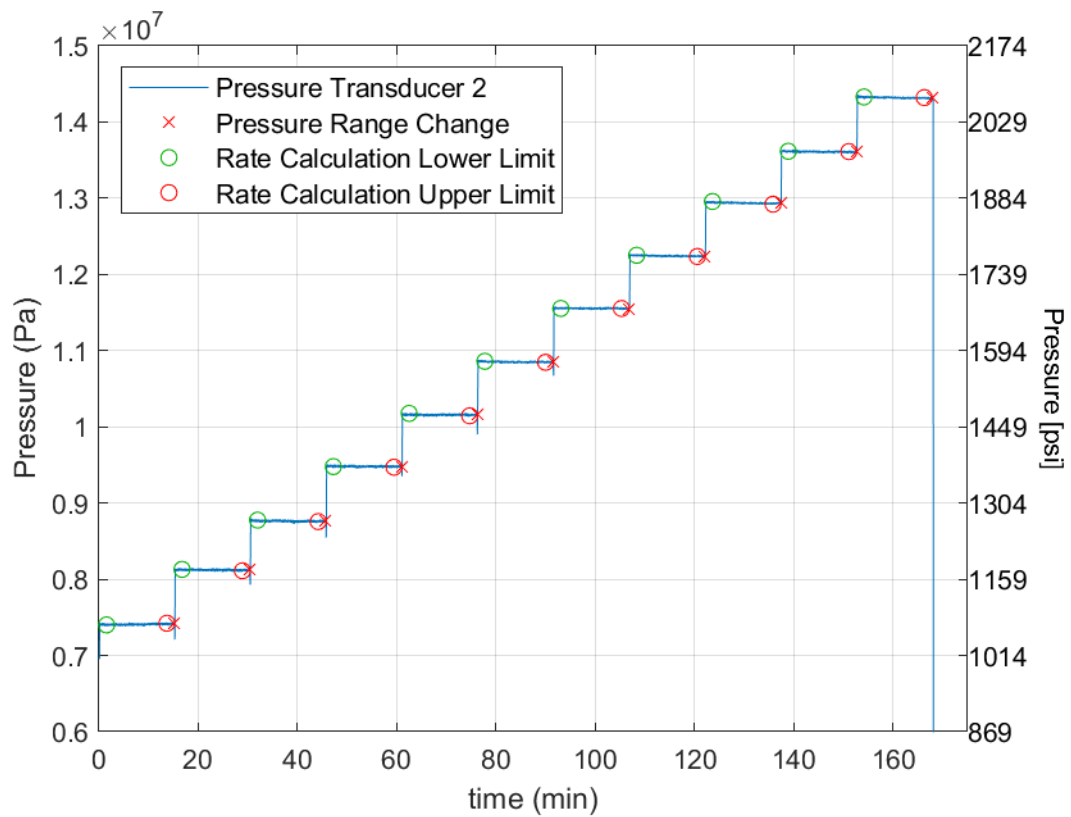


Figure E.3. Pressure vs. Time from 6.89×10^6 to 1.45×10^7 Pa (1000 to 2100 psi)

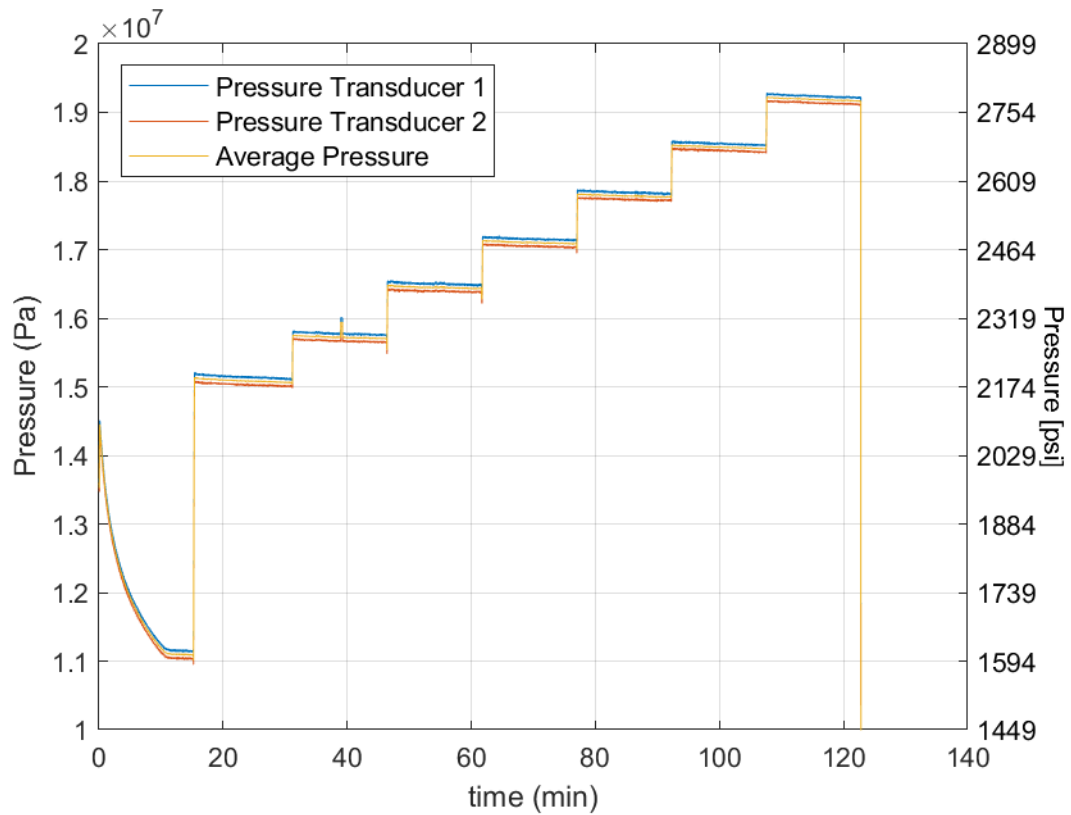


Figure E.4. PT1, PT2, & Average Pressure from 1.45×10^6 to 1.93×10^7 Pa (2100 to 2800 psi)

A system leak occurred during testing at 1.45×10^7 Pa (2100 psi) which produced erroneous data. Calculations for pressure loss at 1.45×10^7 Pa (2100 psi) were done with data captured during the previous trial (6.89×10^6 to 1.45×10^7 Pa (1000 to 2100 psi)).

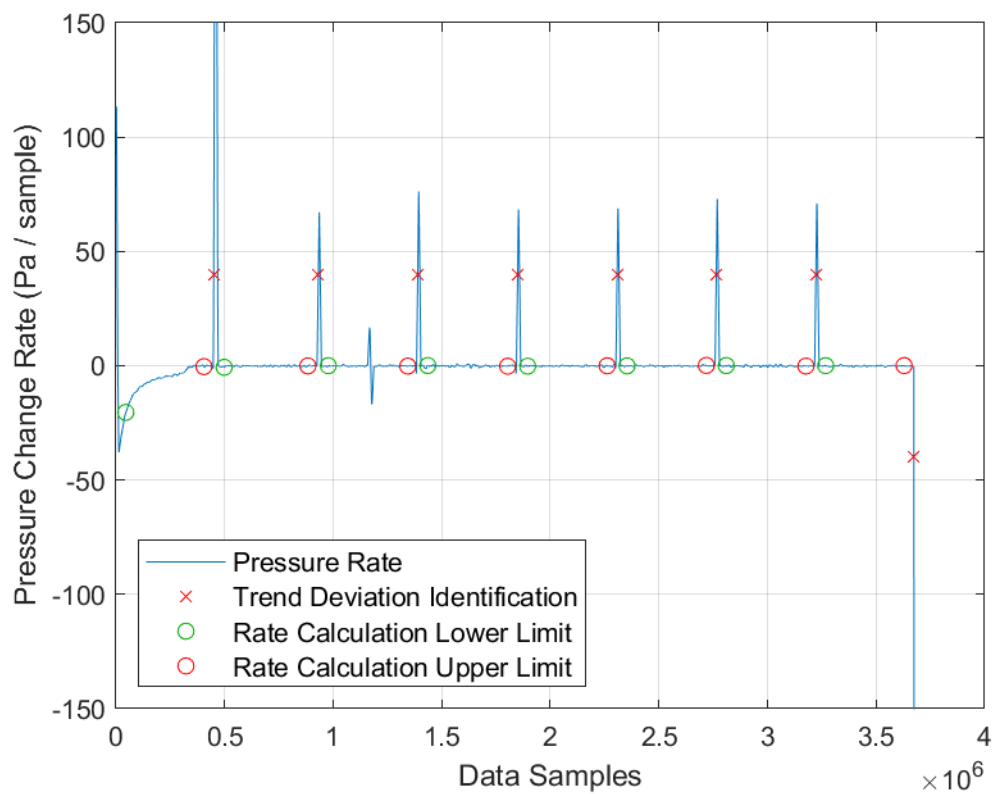


Figure E.5. Pressure Rate Response from 1.45×10^6 to 1.93×10^7 Pa (2100 to 2800 psi)

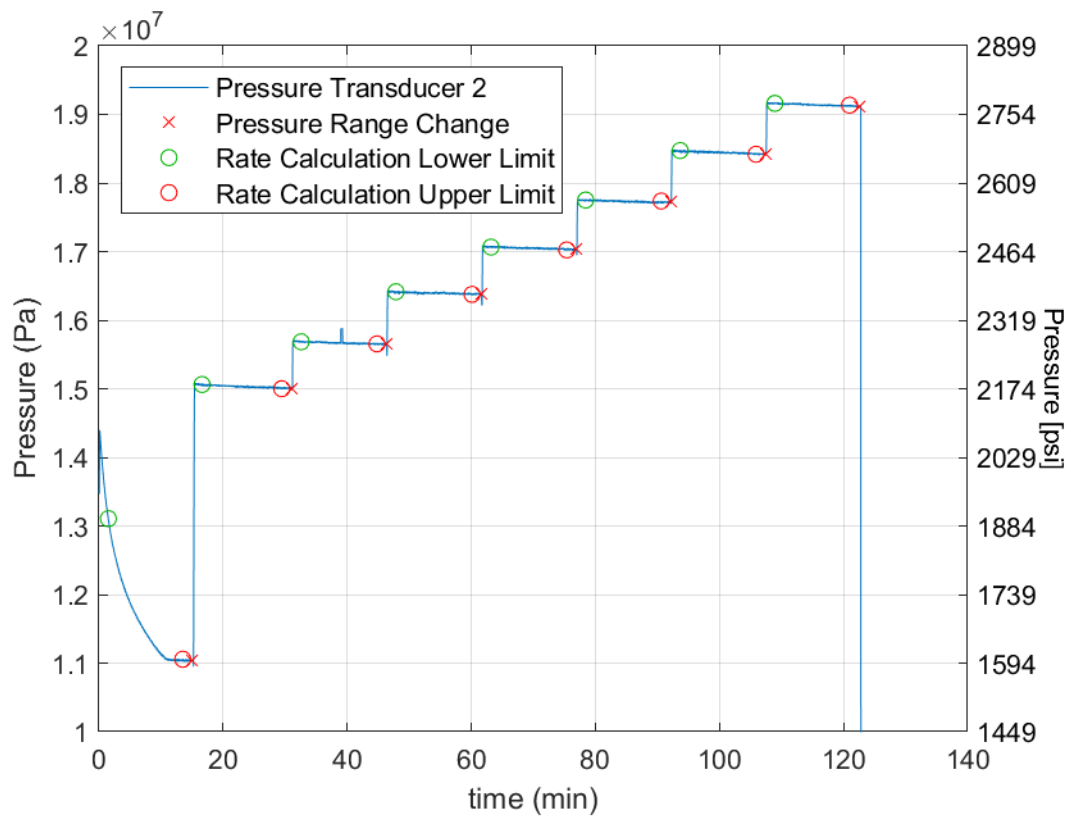


Figure E.6. Pressure vs. Time from 1.45×10^6 to 1.93×10^7 Pa (2100 to 2800 psi)

APPENDIX F: Pressure Loss Code

The Matlab code used to calculate system pressure loss from the system pressure response evaluation is presented in this appendix.

```

LR1 = csvread("LR1k.csv",1,0);
LR2 = csvread("LR2k.csv",1,0);
LR3 = csvread("LR3k.csv",1,0);
LR = [LR1;LR2;LR3(2:end)];

pt1k = csvread("pt21k.csv",1,0);
pt2k = csvread("pt22k.csv",1,0);
pt3k = csvread("pt23k.csv",1,0);
pt = [pt1k;pt2k;pt3k(2:end)];

p11 = csvread("PL1k.csv",1,0);
p12 = csvread("PL2k.csv",1,0);
p13 = csvread("PL3k.csv",1,0);

Pa_abs = round(mean([p11(:,1);p12(:,1);p13(2:end,1)]),2);
psi_abs = round(Pa_abs/psi2pa,2);

Pa_rate = round(mean([p11(:,3);p12(:,3);p13(2:end,3)]),2);
psi_rate = round(Pa_rate/psi2pa,2);

fig_LR = figure;
fs = 10;

[bestfit,gof] = fit(pt,abs(LR),'poly1')

bestfit =
    Linear model Poly1:
    bestfit(x) = p1*x + p2
    Coefficients (with 95% confidence bounds):
        p1 = 9.826e-05 (2.703e-05, 0.0001695)
        p2 = 548.8 (-269, 1367)
    gof = struct with fields:
        sse: 2.0504e+07
        rsquare: 0.26147
        dfe: 23
        adjrsquare: 0.22937
        rmse: 944.18

hplot = plot(pt,abs(LR),'ro');
hold all
hbestfit = plot(bestfit);
hold off

box on;
grid on;
xlabel('Pressure (Pa)')
%xlim()
ylabel('Pressure Loss Rate (Pa/min)')
ylim([-2000 6000])

haxLR = (gca);
haxLR.YAxisLocation = 'left';
haxLR.XAxisLocation = 'bottom';
haxLR.YTick;
haxLR.XTick;
for iii = 1:length(haxLR.YTick)
    text(1.01*diff(xlim),haxLR.YTick(iii),num2str(round(haxLR.YTick(iii)/psi2pa,2)), 'HorizontalAlignment','left')
end

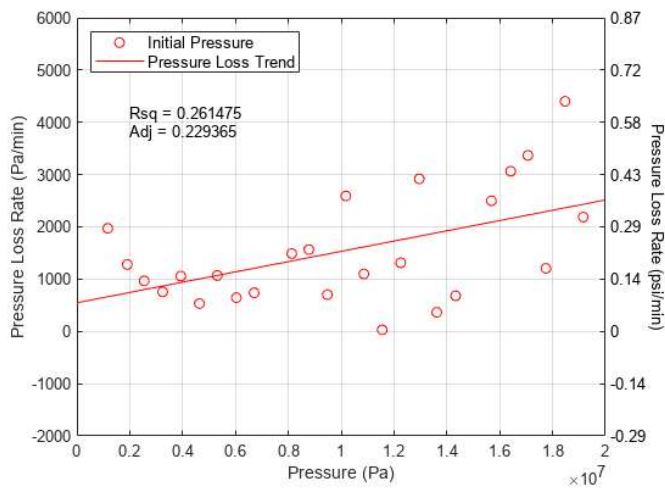
hlegend = legend('show');
hlegend = legend('off');
hlegend = legend('show');

set(hlegend,'location','northwest','fontSize',fs)
set(hplot,'displayname','Initial Pressure')
set(hbestfit,'displayname','Pressure Loss Trend')
set(gca,'fontSize',fs)

text(1.1*max(haxLR.XLim), mean(haxLR.YLim), 'Pressure Loss Rate (psi/min)', 'Rotation',-90, 'HorizontalAlignment','center')

```

```
text(0.1*diff(xlim),0.5*diff(ylim),sprintf('Rsquared = %g\nAdjusted = %g',gof.rsquared,gof.adjustedrsquared))
```



```
A = table(Pa_abs,psi_abs,Pa_rate,psi_rate,'VariableNames',{'Pa','psi','Pa/min','psi/min'})
```

A = 1x4 table

	Pa	psi	Pa/min	psi/min
1	-18819	-2.73	-1521.8	-0.22

```
% Print Things!
% table2latex(A,'PLAvg')
% print(fig_LR,'AvgLR','-dpng');
```

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APPENDIX G:

Experimental Data for Hollow Part Variant 1

Graphs of pressure vs. time and tables for max pressure and pressure loss are presented for each tested sample of buoyant part design variant 1.

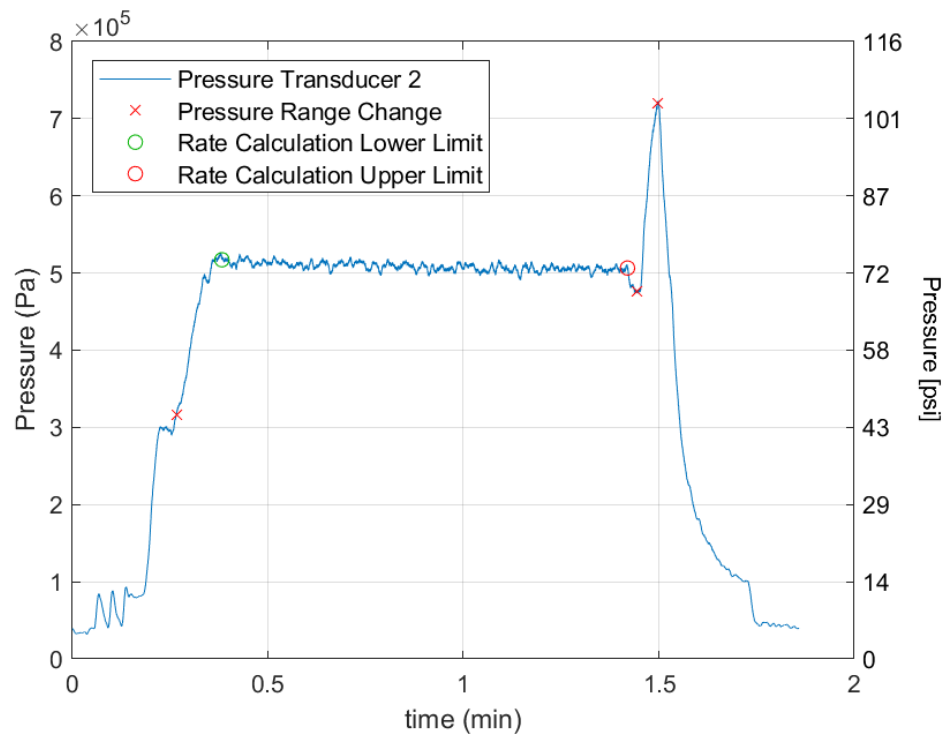


Figure G.1. Pressure vs. Time: Variant 1, 1 mm, Sample 1

Table G.1. Max Pressure: Variant 1, 1 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	711.7	103.2	71	232
PT2	723.1	104.9	72	235
Avg	717.0	104.0	71	233

Table G.2. Pressure Loss: Variant 1, 1 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-10.7	-1.5	-10.3	-1.5

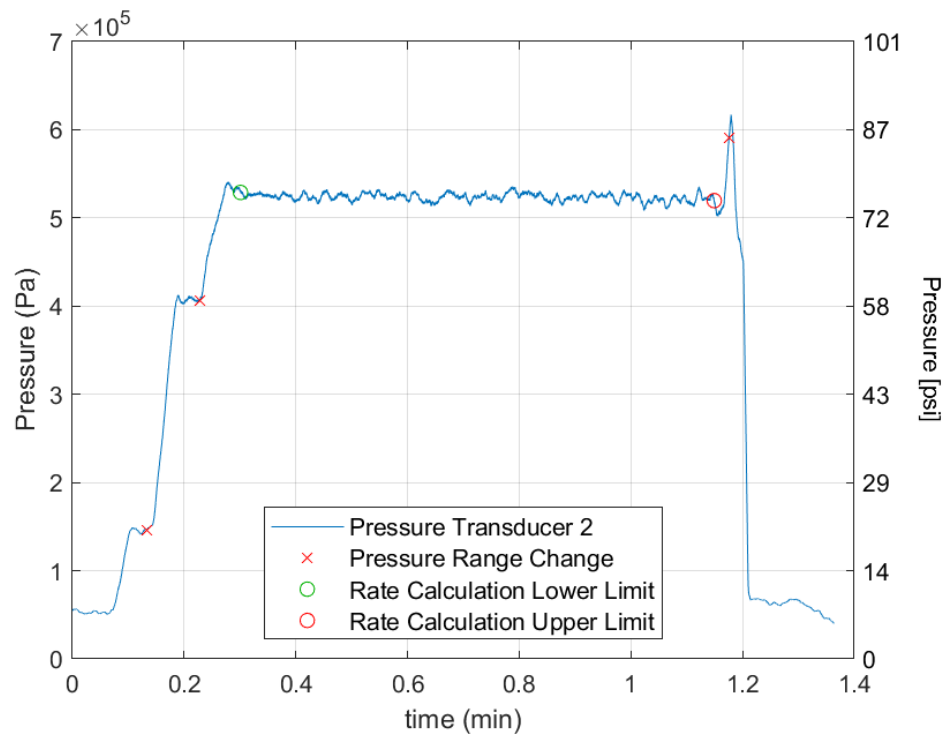


Figure G.2. Pressure vs. Time: Variant 1, 1 mm, Sample 2

Table G.3. Max Pressure: Variant 1, 1 mm, Sample 2

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	601.2	87.2	60	196
PT2	616.8	89.5	61	201
Avg	608.8	88.3	61	198

Table G.4. Pressure Loss: Variant 1, 1 mm, Sample 2

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-9.3	-1.3	-11.0	-1.6

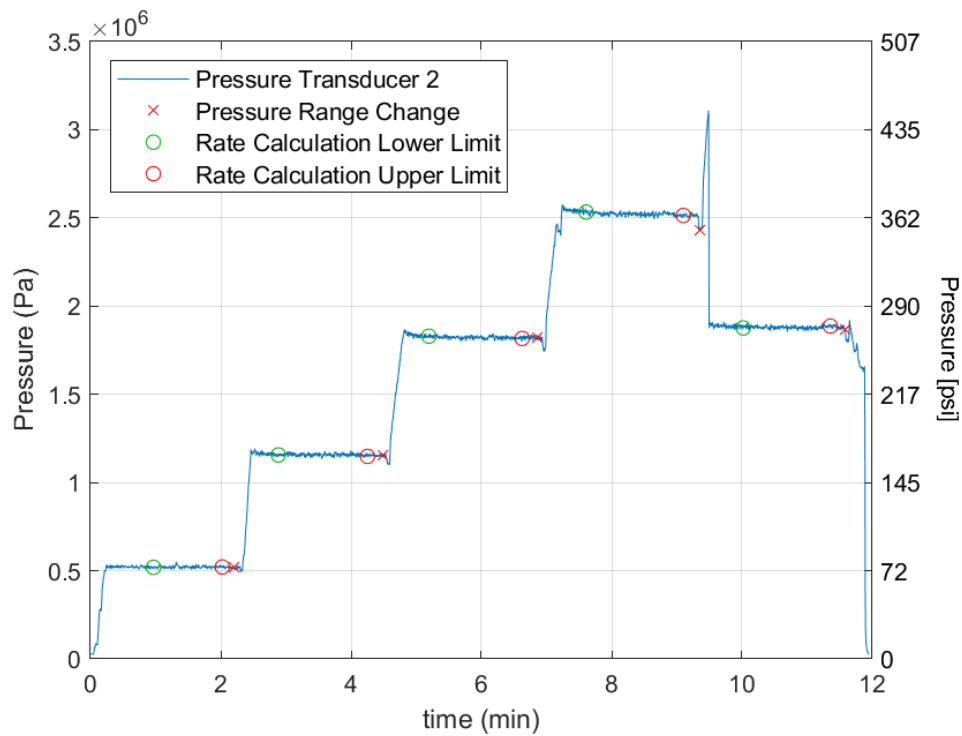


Figure G.3. Pressure vs. Time: Variant 1, 1.5 mm, Sample 1

Table G.5. Max Pressure: Variant 1, 1.5 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	3184.2	461.8	317	1037
PT2	3108.1	450.8	309	1012
Avg	3146.1	456.3	313	1024

Table G.6. Pressure Loss: Variant 1, 1.5 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	2.1	0.3	2.0	0.3
2	-8.3	-1.2	-6.0	-0.9
3	-12.5	-1.8	-8.8	-1.3
4	-19.0	-2.8	-12.7	-1.8
5	10.9	1.6	8.1	1.2

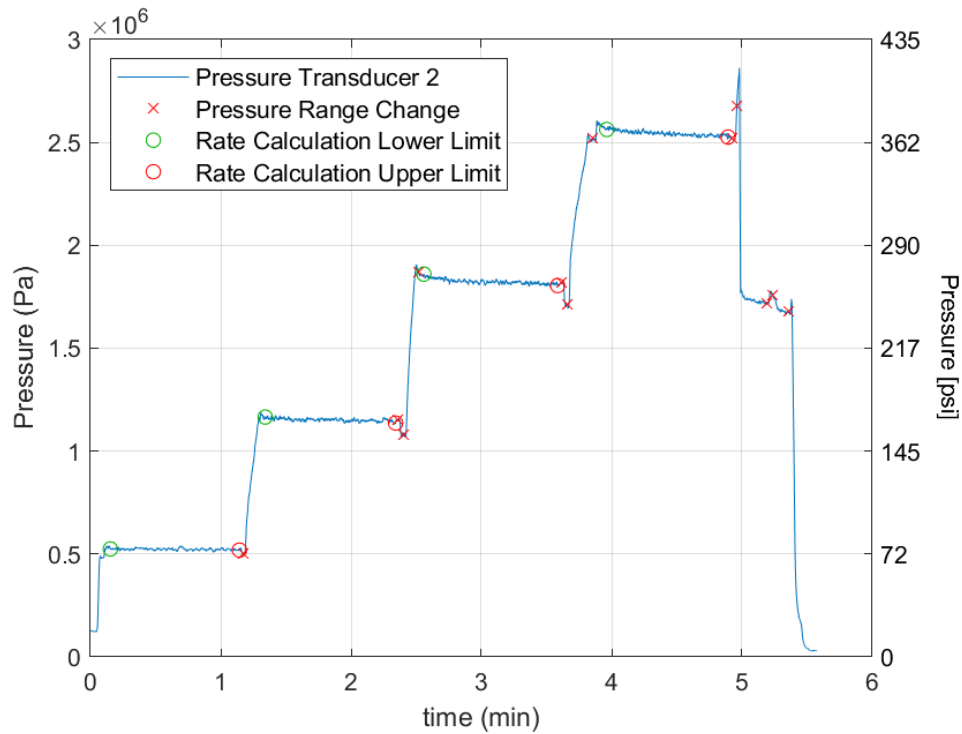


Figure G.4. Pressure vs. Time: Variant 1, 1.5 mm, Sample 2

Table G.7. Max Pressure: Variant 1, 1.5 mm, Sample 2

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	2925.2	424.3	291	952
PT2	2862.3	415.1	285	932
Avg	2893.6	419.7	288	942

Table G.8. Pressure Loss: Variant 1, 1.5 mm, Sample 2

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-5.4	-0.8	-5.5	-0.8
2	-29.2	-4.2	-29.2	-4.2
3	-55.6	-8.1	-54.2	-7.9
4	-36.7	-5.3	-39.4	-5.7

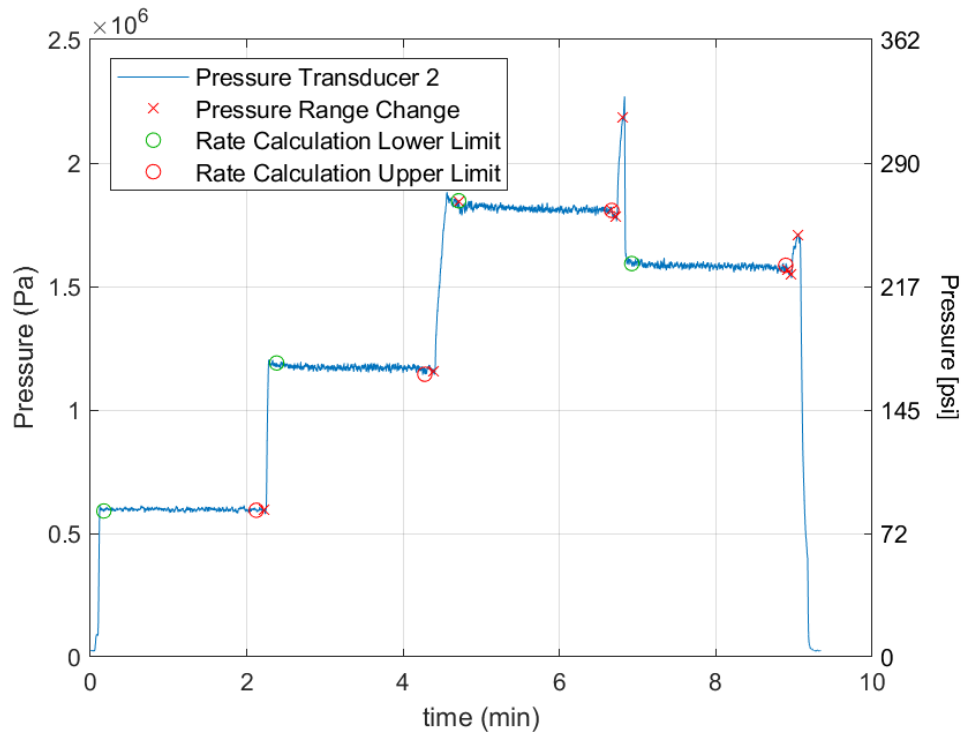


Figure G.5. Pressure vs. Time: Variant 1, 1.5 mm, Sample 3

Table G.9. Max Pressure: Variant 1, 1.5 mm, Sample 3

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	2344.3	340.0	233	763
PT2	2269.5	329.2	226	739
Avg	2306.3	334.5	229	751

Table G.10. Pressure Loss: Variant 1, 1.5 mm, Sample 3

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	2.9	0.4	1.5	0.2
2	-45.6	-6.6	-24.1	-3.5
3	-38.9	-5.6	-19.9	-2.9
4	-7.2	-1.0	-3.6	-0.5

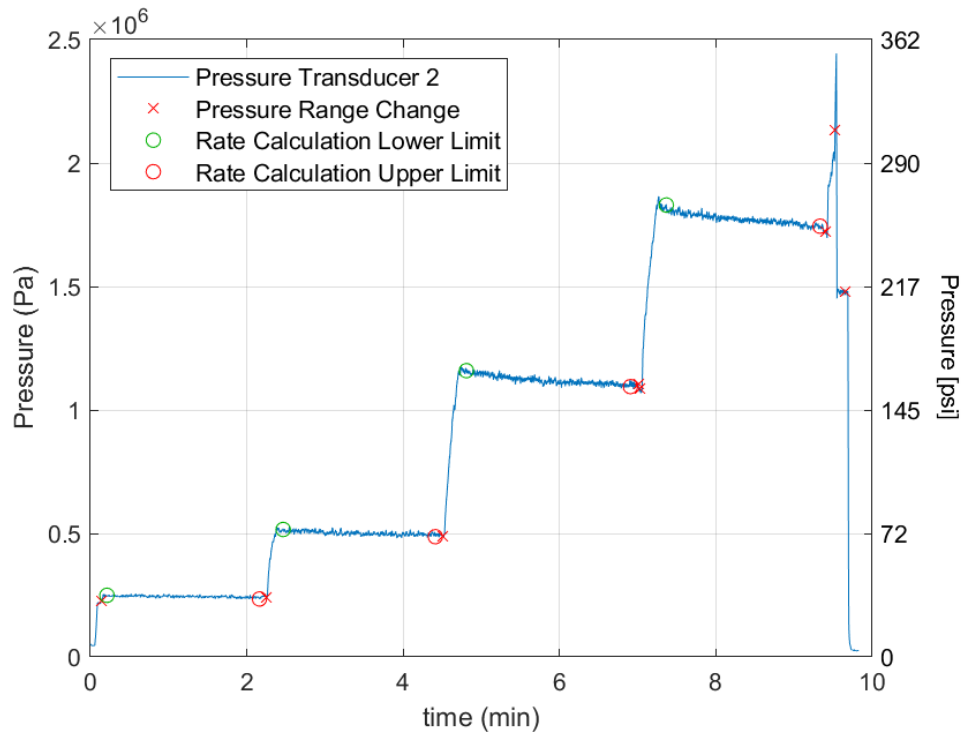


Figure G.6. Pressure vs. Time: Variant 1, 1.5 mm, Sample 4

Table G.11. Max Pressure: Variant 1, 1.5 mm, Sample 4

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	2508.2	363.8	250	817
PT2	2444.0	354.5	243	796
Avg	2476.0	359.1	246	806

Table G.12. Pressure Loss: Variant 1, 1.5 mm, Sample 4

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-14.8	-2.1	-7.6	-1.1
2	-29.2	-4.2	-15.0	-2.2
3	-64.7	-9.4	-30.8	-4.5
4	-85.6	-12.41	-43.5	-6.3

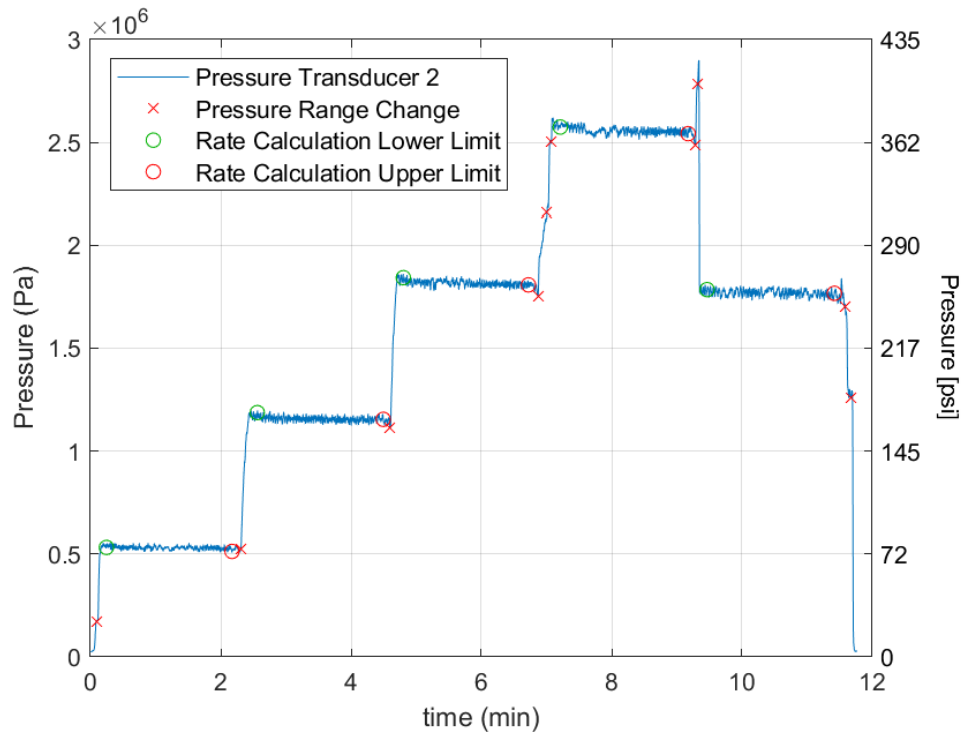


Figure G.7. Pressure vs. Time: Variant 1, 1.5 mm, Sample 5

Table G.13. Max Pressure: Variant 1, 1.5 mm, Sample 5

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	2962.0	423.0	295	964
PT2	2900.1	420.6	289	944
Avg	2929.6	424.9	291	954

Table G.14. Pressure Loss: Variant 1, 1.5 mm, Sample 5

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-19.3	-2.8	-10.0	-1.5
2	-32.7	-4.8	-16.9	-2.5
3	-35.0	-5.1	-18.2	-2.6
4	-32.2	-4.7	-16.4	-2.4
5	-17.7	-2.6	-9.1	-1.3

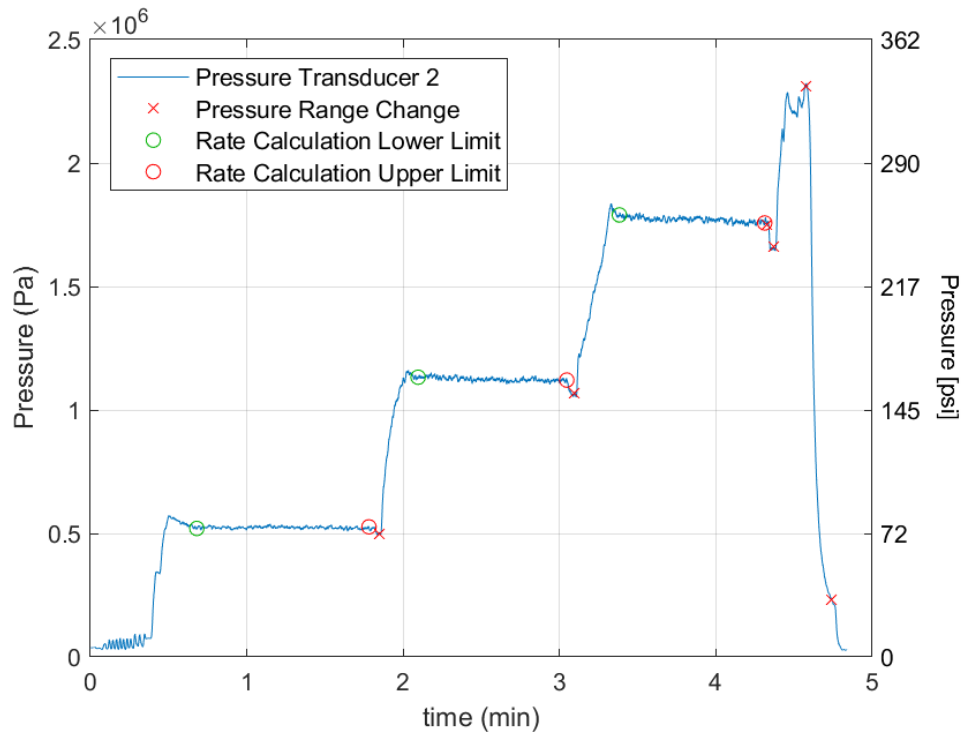


Figure G.8. Pressure vs. Time: Variant 1, 2 mm, Sample 1

Table G.15. Max Pressure: Variant 1, 2 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	2365.4	343.1	235	770
PT2	2323.4	337.0	231	756
Avg	2342.0	340.0	233	763

Table G.16. Pressure Loss: Variant 1, 2 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	6.5	0.9	5.9	0.9
2	-11.5	-1.7	-12.1	-1.8
3	-32.4	-4.7	-34.9	-5.1

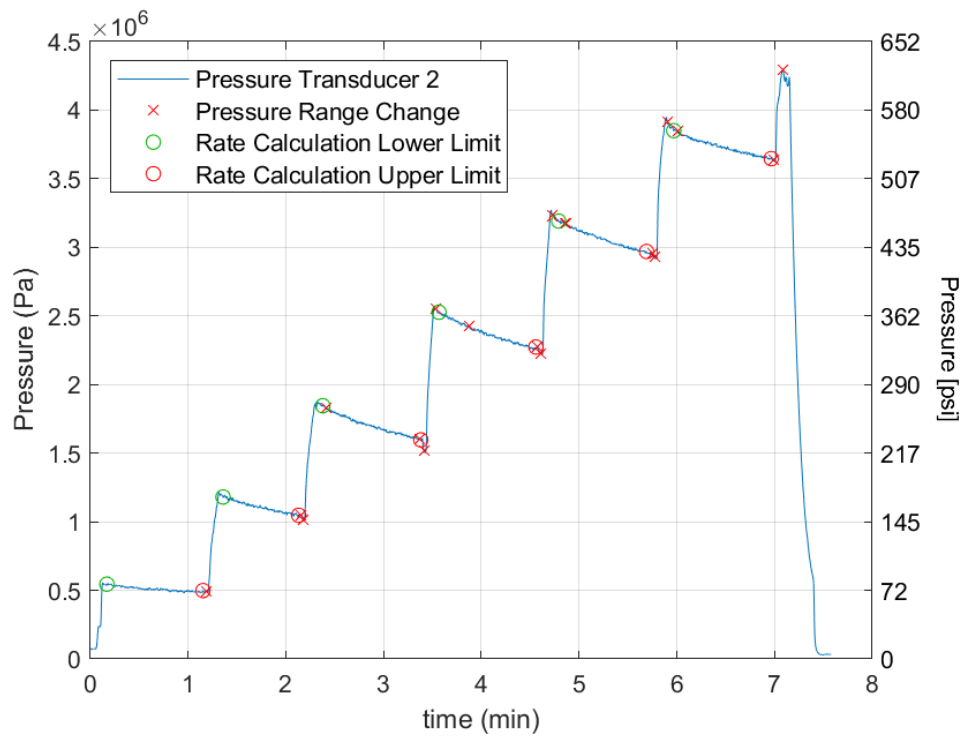


Figure G.9. Pressure vs. Time: Variant 1, 2 mm, Sample 2

Table G.17. Max Pressure: Variant 1, 2 mm, Sample 2

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4382.0	635.6	436	1427
PT2	4303.0	624.1	428	1401
Avg	4342.0	629.8	432	1414

Table G.18. Pressure Loss: Variant 1, 2 mm, Sample 2

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-47.0	-6.8	-47.8	-6.9
2	-133.1	-19.3	-171.9	-24.9
3	-248.8	-36.1	-248.1	-36.0
4	-253.6	-36.8	-255.8	-37.1
5	-221.4	-32.1	-246.1	-35.7
6	-204.3	-29.6	-204.8	-29.7

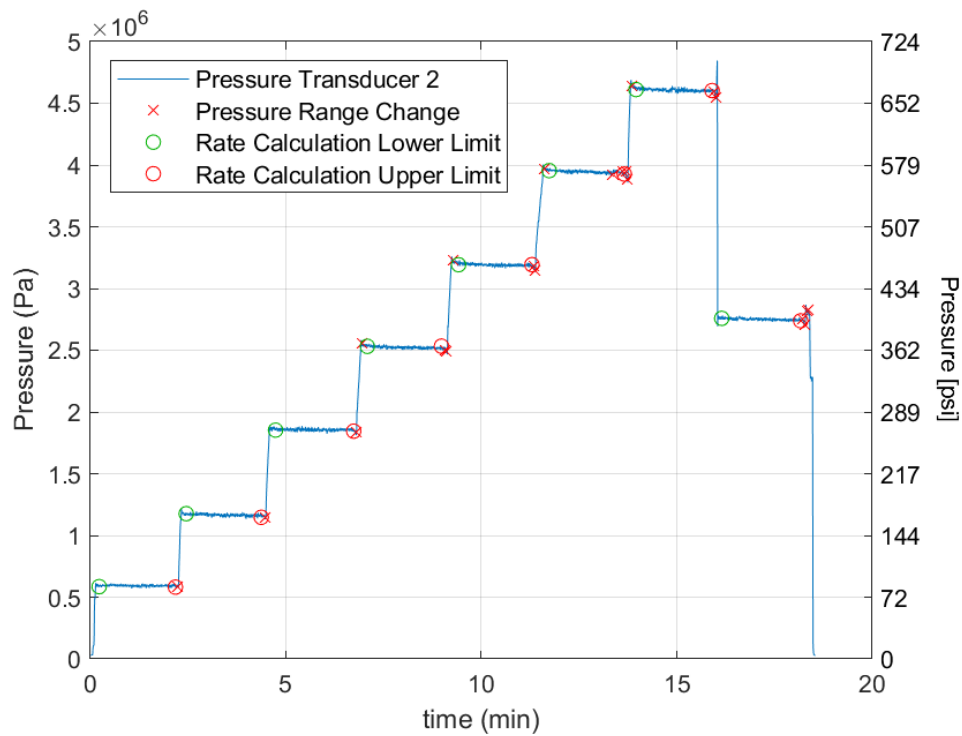


Figure G.10. Pressure vs. Time: Variant 1, 2 mm, Sample 3

Table G.19. Max Pressure: Variant 1, 2 mm, Sample 3

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4937.5	715.8	491	1607
PT2	4846.0	702.6	482	1577
Avg	4890.3	709.0	487	1592

Table G.20. Pressure Loss: Variant 1, 2 mm, Sample 3

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-3.8	-0.6	-2.0	-0.28
2	-29.6	-4.3	-15.4	-2.23
3	-8.4	-1.2	-4.2	-0.61
4	1.7	0.3	0.9	0.13
5	0.8	0.1	0.4	0.06
6	-27.7	-4.0	-14.4	-2.09
7	-8.1	-1.2	-4.2	-0.60
8	-19.7	-2.9	-9.7	-1.41
Avg	-10.7	-1.6	-5.5	-0.80

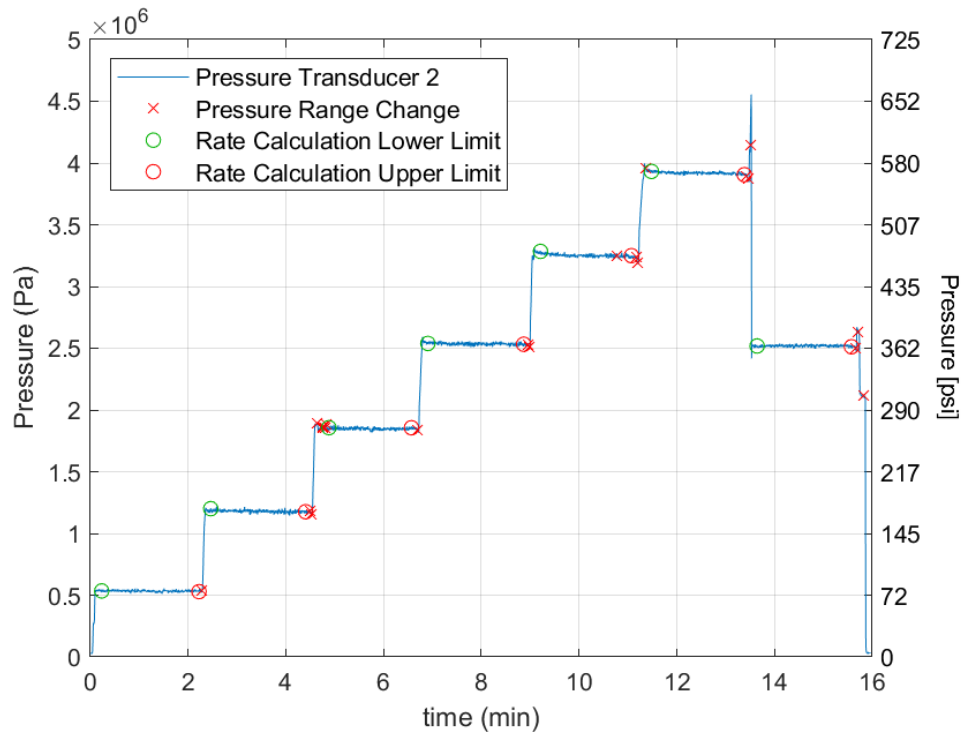


Figure G.11. Pressure vs. Time: Variant 1, 2 mm, Sample 4

Table G.21. Max Pressure: Variant 1, 2 mm, Sample 4

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4629.2	671.4	461	1507
PT2	4555.3	660.7	453	1483
Avg	4591.8	666.0	457	1495

Table G.22. Pressure Loss: Variant 1, 2 mm, Sample 4

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-5.7	-0.8	-2.9	-0.4
2	-24.8	-3.6	-12.7	-1.8
3	-0.8	-0.1	-0.5	-0.1
4	-6.1	-0.9	-3.1	-0.4
5	-34.5	-5.0	-18.5	-2.7
6	-26.5	-3.8	-13.9	-2.0
7	-6.3	-0.9	-3285.8	-0.5

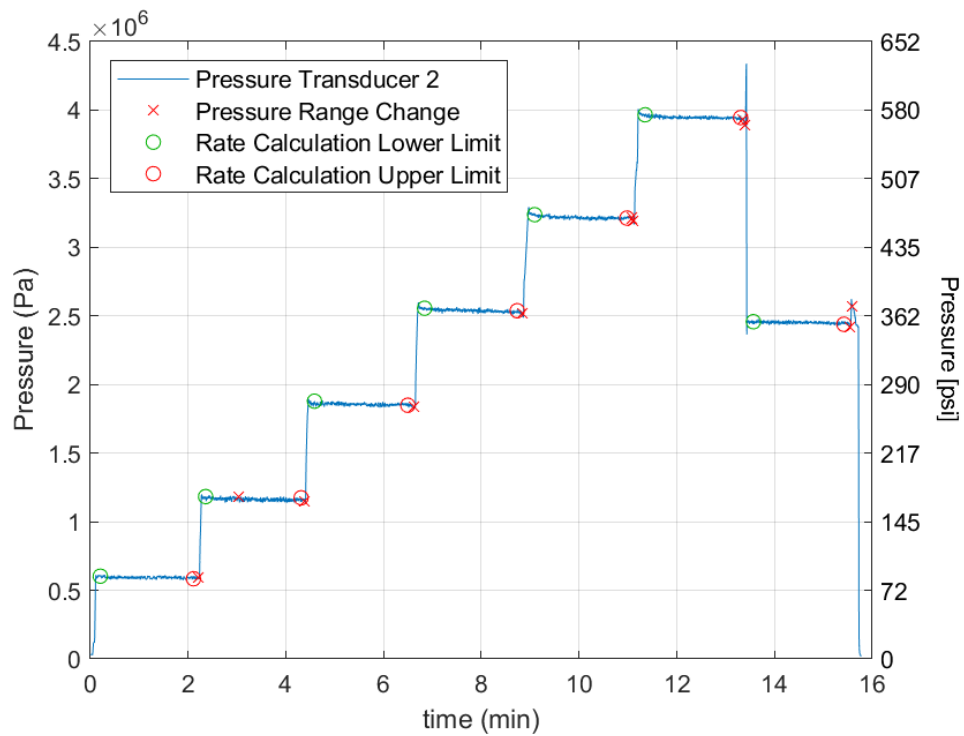


Figure G.12. Pressure vs. Time: Variant 1, 2 mm, Sample 5

Table G.23. Max Pressure: Variant 1, 2 mm, Sample 5

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4415.4	640.4	439	1438
PT2	4338.9	629.3	432	1413
Avg	4376.9	634.8	435	1425

Table G.24. Pressure Loss: Variant 1, 2 mm, Sample 5

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-18.8	-2.7	-9.9	-1.4
2	-8.8	-1.3	-4.5	-0.7
3	-29.6	-4.3	-15.5	-2.2
4	-18.0	-2.6	-9.5	-1.4
5	-24.9	-3.6	-13.2	-2.0
6	-20.4	-3.0	-10.5	-1.5
7	-18.6	-2.7	-10.0	-1.5

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APPENDIX H:

Experimental Data for Hollow Part Variant 2

Graphs of pressure vs. time and tables for max pressure and pressure loss are presented for each tested sample of buoyant part design variant 2.

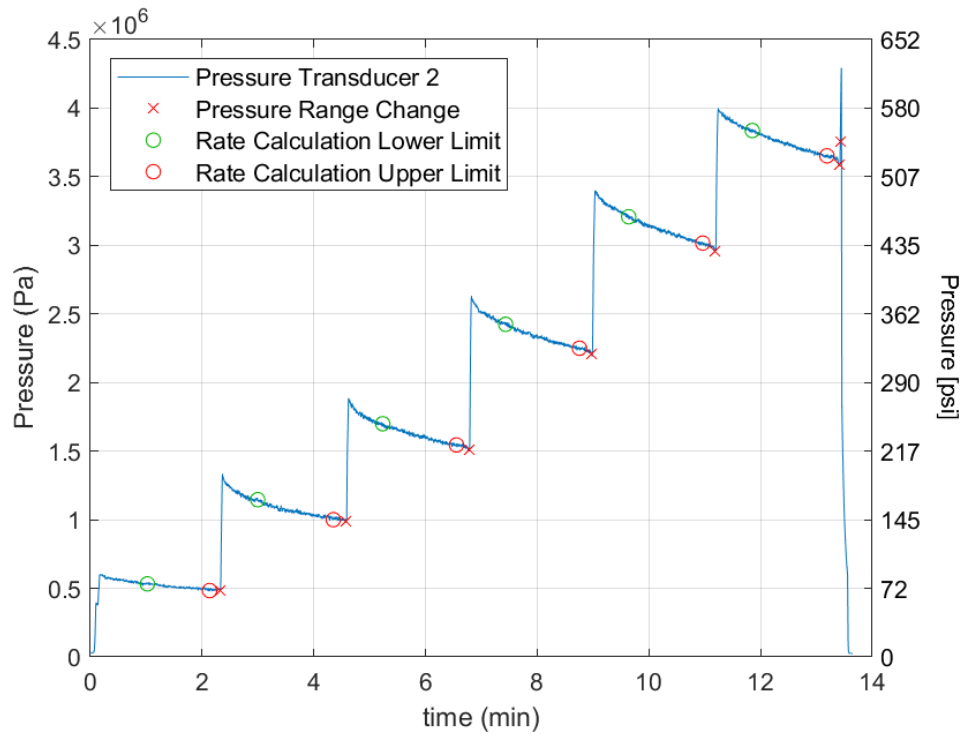


Figure H.1. Pressure vs. Time: Variant 2, 2 mm, Sample 1

Table H.1. Max Pressure: Variant 2, 2 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4373.5	634.3	435	1424
PT2	4294.0	622.8	427	1398
Avg	4333.5	628.5	431	1411

Table H.2. Pressure Loss: Variant 2, 2 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-48.9	-7.1	-44.4	-6.4
2	-145.7	-21.1	-107.7	-15.6
3	-154.7	-22.4	-117.1	-17.0
4	-174.2	-25.3	-131.9	-19.1
5	-192.0	-27.8	-144.8	-21.0
6	-184.4	-26.7	-138.1	-20.0

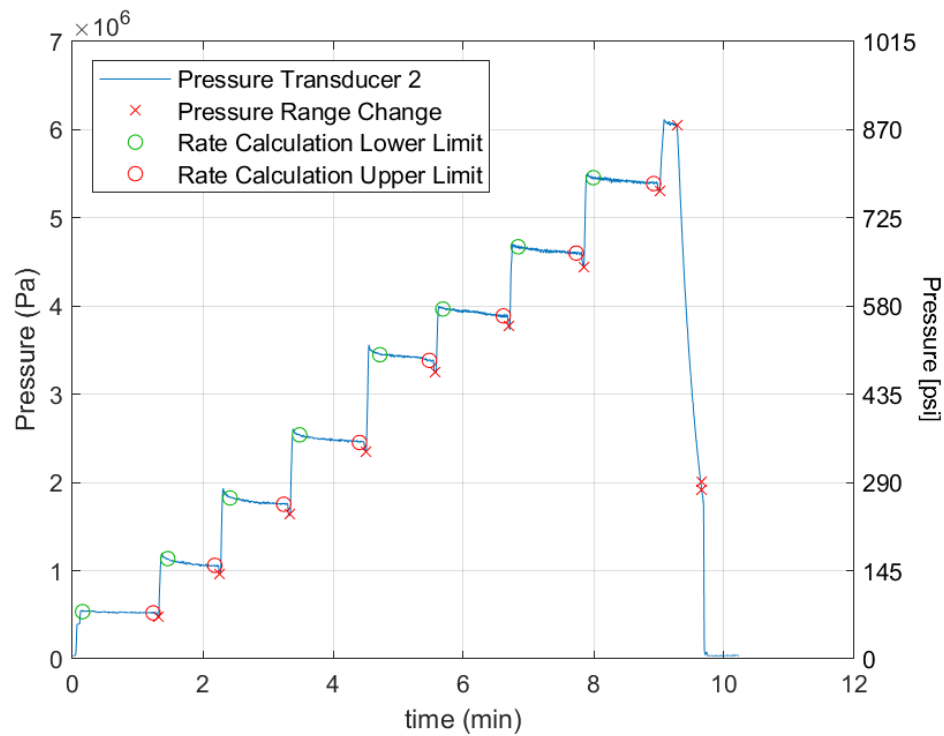


Figure H.2. Pressure vs. Time: Variant 2, 3 mm, Sample 1

Table H.3. Max Pressure: Variant 2, 3 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	6225.9	903.0	619	2027
PT2	6118.5	887.4	609	1992
Avg	6171.3	895.1	614	2009

Table H.4. Pressure Loss: Variant 2, 3 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-14.1	-2.0	-13.0	-1.9
2	-76.5	-11.1	-105.91	-15.4
3	-71.2	-10.3	-86.4	-12.5
4	-87.1	-12.6	-94.7	-13.7
5	-65.0	-9.4	-85.6	-12.4
6	-77.8	-11.3	-83.8	-12.2
7	-71.9	-10.4	-80.5	-11.7
8	-66.9	-9.7	-72.2	-10.5

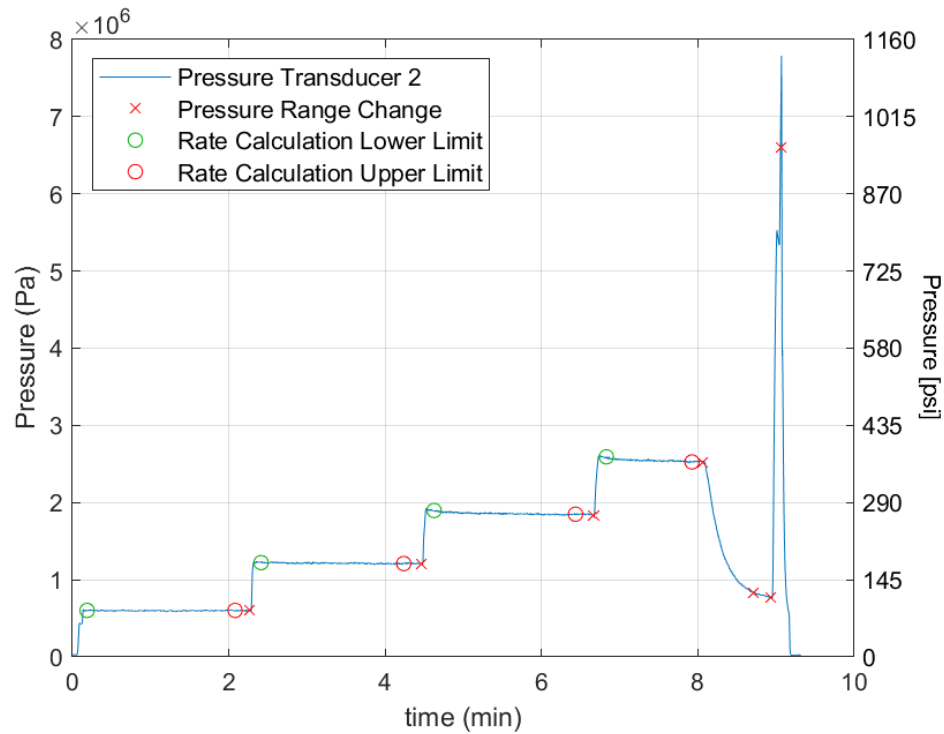


Figure H.3. Pressure vs. Time: Variant 2, 3 mm, Sample 2

Table H.5. Max Pressure: Variant 2, 3 mm, Sample 1

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	7883.7	1143.4	784	2567
PT2	7788.6	1129.6	775	2536
Avg	7836.1	1136.5	780	2551

Table H.6. Pressure Loss: Variant 2, 3 mm, Sample 1

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	0.3	0.0	0.2	0.0
2	-12.1	-1.8	-6.6	-1.0
3	-47.7	-6.9	-26.4	-3.8
4	-66.1	-9.6	-60.4	-8.8

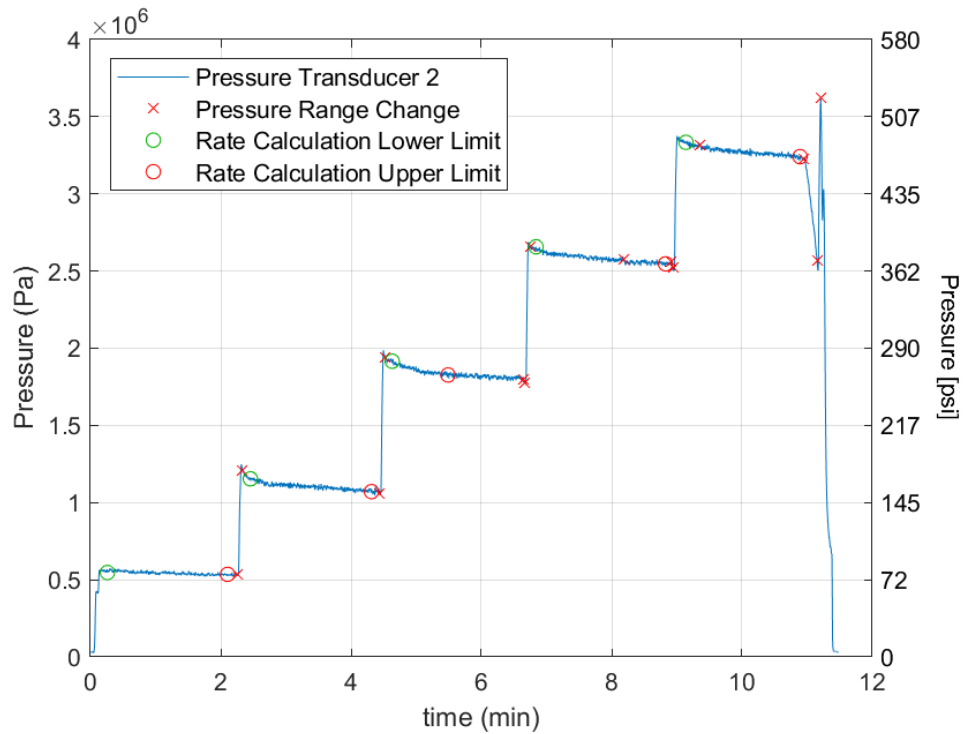


Figure H.4. Pressure vs. Time: Variant 2, 3 mm, Sample 3

Table H.7. Max Pressure: Variant 2, 3 mm, Sample 3

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	3708.5	537.9	369	1207
PT2	3626.0	525.9	361	1181
Avg	3667.1	531.9	365	1194

Table H.8. Pressure Loss: Variant 2, 3 mm, Sample 3

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-12.1	-1.8	-6.6	-1.0
2	-83.0	-12.0	-44.6	-6.5
3	-89.2	-12.9	-104.2	-15.1
4	-110.5	-16.0	-55.6	-8.1
5	-92.7	-13.5	-53.0	-7.7

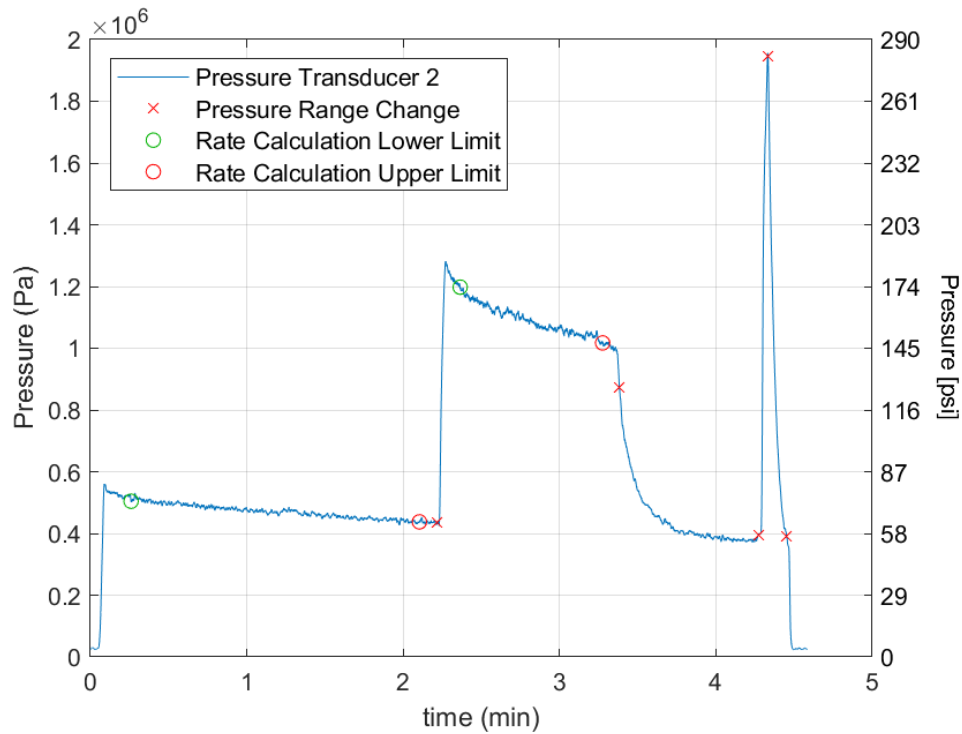


Figure H.5. Pressure vs. Time: Variant 2, 3 mm, Sample 4

Table H.9. Max Pressure: Variant 2, 3 mm, Sample 4

	Average Max Pressure		Average Depth	
	kPa	psi	Meters of Sea Water	Feet of Sea Water
PT1	1990.7	288.7	198	648
PT2	1956.5	283.8	195	637
Avg	1972.9	286.1	196	642

Table H.10. Pressure Loss: Variant 2, 3 mm, Sample 4

Interval	Average Pressure Loss		Average Loss Rate	
	kPa	psi	kPa / min	psi / min
1	-66.8	-9.7	-36.2	-5.3
2	-180.7	-26.2	-198.4	-28.8

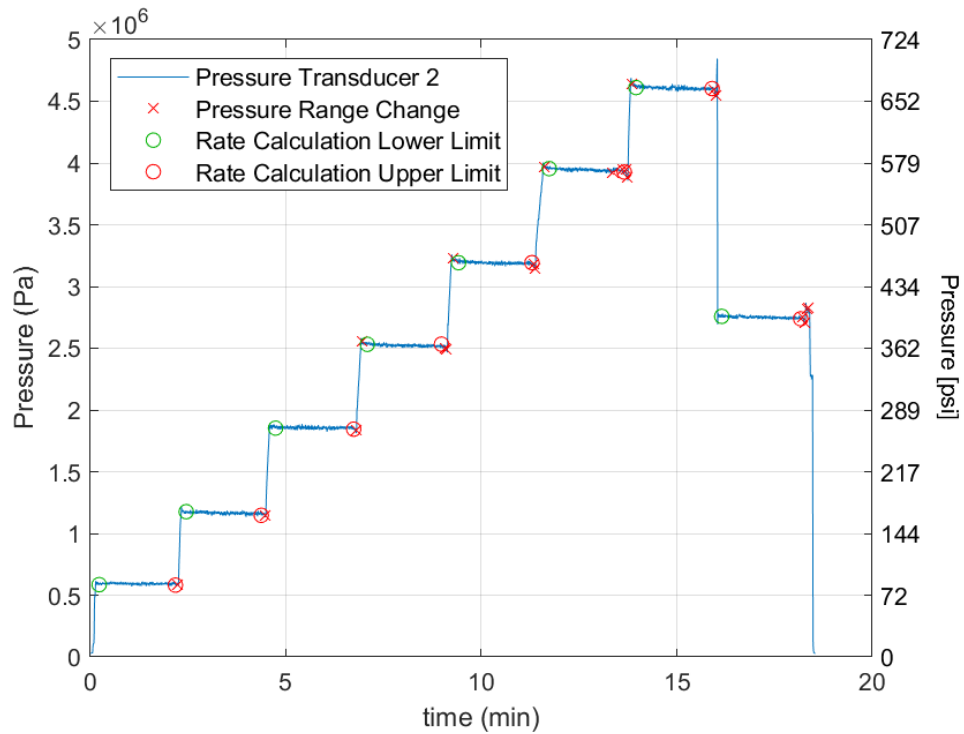


Figure H.6. Pressure vs. Time: Variant 2, 3 mm, Sample 5

Table H.11. Max Pressure: Variant 2, 3 mm, Sample 5

	Average Max Pressure		Average Depth	
	Pa	psi	Meters of Sea Water	Feet of Sea Water
PT1	4937482.30	715.81	491.20	1606.92
PT2	4846017.24	702.55	482.10	1577.15
Avg	4890275.19	708.97	486.51	1591.56

Table H.12. Pressure Loss: Variant 2, 3 mm, Sample 5

Interval	Average Pressure Loss		Average Loss Rate	
	Pa	psi	Pa / min	psi / min
1	-3818.98	-0.55	-1957.61	-0.28
2	-29563.63	-4.29	-15404.41	-2.23
3	-8415.14	-1.22	-4209.39	-0.61
4	1703.61	0.25	899.40	0.13
5	777.18	0.11	414.16	0.06
6	-27685.02	-4.01	-14418.03	-2.09
7	-8070.30	-1.17	-4150.11	-0.60
8	-19691.93	-2.85	-9746.07	-1.41
Avg	-10725.00	-1.55	-5546.6	-0.80

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APPENDIX I: Xerox ElemX SOP

Xerox ElemX SOP utilized by military personnel at the time of this thesis is presented in this appendix.

Xerox ElemX Operating Procedure

Naval Postgraduate School

Rev 004, SEPT 2022

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1 Preparing the Nozzle

1.1 Plasma Treating

- 1) Plasma treat the lower pump IAW the provided OI.

Plasma treating is essential to ensure proper wetting of the nozzle.

1.2 Pump Assembly

- 1) Verify onepiece and upper pump are free from debris. Clean gently with soft plastic.
- 2) Apply a thin layer of glue to the mating surface of the nozzle. Mate the nozzle to the upper pump and allow the glue to dry. Wipe off any excess glue

The onepiece and upper pump are fragile and care must be taken when cleaning to ensure the upper pump is not punctured and the onepiece is not cracked.

2 Start UP

2.1 Aligning Mainline Power & PC Start Up

- | | |
|--|---|
| 1) Take the power selector switch on the back of the Xerox ElemX to ON. | Aligns mainline power to the Xerox ElemX. |
| 2) Pull the emergency stop button out. | Resets the emergency shutdown interlock. |
| 3) Wait for the PC to start up then log in. Once logged in, the HMI should start up automatically. If not, select it from the desktop. | |
| 4) Once the HMI is open, select NO ACCESS in the top right corner and log in with the user password used to log in to the PC. | |
| 5) IF flashing, press the green START button on the OBP. | Resets the HMI. |
| 6) Press the JOG button on the OBP. | Shifts printer axis control to manual. |
| 7) Press and hold the RESET button on the OBP for 5 seconds. | Clears start up alerts and warnings. |
| 8) Press FEED START on OBP | Permits operation of motion controls |

2.2 Installing Pump

- | | |
|---|---|
| 1) Press the DOOR LOCK button on the OBP. | Disengages the front door interlock. |
| 2) Verify that the DOOR LOCK button is flashing green then open the front door. | Damage can occur to the door locking mechanism if the door interlock is not met. |
| 3) Place the pump assembly into the print head and ensure the following: <ul style="list-style-type: none"> a) thermocouple are properly routed, b) heater contacts are secure, c) shoulder bolts are flush and tight. | Print quality can be negatively affected if the upper pump assembly and associated instrumentation is not properly installed. Excess vibration of loose print head components can lead to damage of the print head and upper pump assembly. |
| 4) Select the METALS CONTROL tab on the HMI and verify the thermocouple position | The upper pump is delicate and can be punctured by the thermocouple. |

on the print head camera.

Verification of proper installation must be performed via the viewing camera to prevent damage to the thermocouple.

- 5) Place a catch tray for the Burn-In process in the center of the build plate.
- 6) Close the front door.
- 7) Verify the green START button is flashing then push the green START button.

2.3 Energizing Chiller Power & Aligning Argon

WARNING: DO NOT EXCEED 100 PSI ON THE EXTERNAL ARGON REGULATOR. EXCEEDING 100 PSI INTO THE XEROX ELEM X COULD RESULT IN INTERNAL REGULATOR DAMAGE.

- 1) Align external argon source to Xerox ElemX and adjust regulator pressure to 40 ± 10 psi.
- 2) Press the COOLANT ON button on the OBP.
- 3) Select the THERMAL CONTROLS tab on the HMI. If EMERGENCY CHILLER STOP is flashing red, select EMERGENCY CHILLER STOP.
- 4) Press the ARGON button on the OBP.
- 5) Observe the argon flow indication increase to 7 ± 0.15 L/min. If argon flow does not reach 7 ± 0.15 L/min, verify external argon regulator outlet pressure.

DO NOT EXCEED 100 PSI.

EMERGENCY CHILLER STOP may be flashing as result of the shutdown procedure.

2.4 Energizing Heaters

- 1) Press the UPPER HEATER ENABLE and BUILD HEATER ENABLE buttons on the OBP.
- 2) Select the THERMAL CONTROLS tab on

the HMI then select BED START HEATING and HEAD START HEATING on the THERMAL CONTROLS SCREEN. Observe both temperatures increase. If temperatures do not increase, verify proper logon to the HMI and proper argon flow.

- 3) Wait for head temperature to reach 825 °C before proceeding. If not using "Plug," turn off prefill plug in user diagnostics.

Attempting to fill the reservoir before print head temperature of 825 °C is reached WILL CAUSE DAMAGE TO THE UPPER PUMP.

2.5 Filling Reservoir

- 1) Select the METAL CONTROLS tab on the HMI.
- 2) Press the WIRE FEED button on the OBP.
- 3) Take the WIRE FEED knob to 80mm/min.
- 4) Select FILL MODE on the METAL CONTROLS screen and observe the filament move into the reservoir.
- 5) If the HMI switches from FILL MODE to MANUAL MODE, verify the reservoir level is above the set point of 121.0 mm then select FILL MODE to continue filling.
- 6) After at least 5 minutes of feeding at 80 mm/min, take the WIRE FEED knob to 150 mm/min.
- 7) Once the melt pool is visible in the print head camera, take the WIRE FEED knob to 200 mm/min.
- 8) When the reservoir level reaches 121.0 mm, observe the HMI shift from FILL MODE to HOLD MODE.
- 9) Once HOLD MODE is active, take the WIRE FEED knob to 100%.

As the filament is fed into the reservoir, the laser level sensor can erroneously detect level out of range and shift from automatic feeding to manual feeding.

2.6 Energizing the High Voltage Supply & Electronic Pulse Controller

- | | |
|--|--|
| <ol style="list-style-type: none"> 1) Press the HV ENABLE button on the OBP then wait 10 seconds. 2) Press the EPC ENABLE button on the OBP. | <p>A 10 second wait is required to ensure the current demand does not exceed the interlock to prevent simultaneous energization of HV and EPC.</p> |
|--|--|

2.7 Loading Pulse Width

- | | |
|---|----------|
| <ol style="list-style-type: none"> 1) Select the METAL CONTROLS tab on the HMI. 2) Select LOAD TIME on the METAL CONTROLS screen and observe the PULSE HIGH TIME read +0.1300 | <p>.</p> |
|---|----------|

2.8 Burn-In

- | | |
|---|--|
| <ol style="list-style-type: none"> 1) Verify work offsets are centered at 150X and 150Y by performing the following: <ol style="list-style-type: none"> a) Select the OPERATE tab on the HMI. b) Select the MACHINE icon on the OPERATE screen to display the MACHINE page. c) Select PARAMETERS then WORK OFFSETS. d) Select the G54...G57 icon and verify G56 and G57 values for X and Y are 150. f) Return to the main OPERATE page by selecting PARAMETERS then MACHINE. 2) On the OBP, press the RESET button. 3) Take the FREQUENCY knob to 100%. 4) Select the METAL CONTROLS tab on the HMI. 5) Select BURN-IN on the METAL CONTROLS screen. | <p>Not Necessary for Burn-In, but good to check before each print.</p> |
|---|--|

- Jetting should appear as slow moving, consistent drops. Spraying, inconsistent drop size, or excessive deviation from vertical jetting are indications of poor nozzle performance and a new nozzle should be considered.

2.9 Setting Drop Mass

-

flashing green then open the front door. 3) Place a zeroed collection tray under the print head and close the front door. 4) Verify the START button on the OBP is flashing green then press the START button. 5) Press the HV ENABLE button on the OBP then wait 10 seconds. 6) Press the EPC ENABLE button on the OBP. Confirm Build Plate Loaded on Operate Tab. 7) On the OBP, press the CYCLE STOP button then the RESET button. 8) Select the METAL CONTROLS tab on the HMI. 9) Select CALIBRATE on the METAL CONTROLS screen. 10) Take the FREQUENCY knob to 60% (300 Hz). 11) Press the CYCLE START button on the OBP. 12) When the cycle completes, press the DOOR LOCK button on the OBP. Verify that the DOOR LOCK button is flashing green then open the front door. 13) Remove the collection tray and weigh it. The mass should be 1.45 ± 0.05 grams. If not, adjust the PULSE HIGH TIME as follows: a) Increase/Decrease the PULSE HIGH TIME by 0.001 ms per 0.05 grams under/over 1.45 grams. 14) Once PULSE HIGH TIME has been adjusted, perform steps (11) through (13) until mass is 1.45 ± 0.05 grams.	Assess Jetting Quality Up drafts generated by the hot print material may cause lighter-than-actual weight readings. Allowing time for the material to cool down will reduce fluctuations in weight readings.
---	--

3 Printing

3.1 Selecting a Part to Print

- 1) Select the OPERATE tab on the HMI.
- 2) On the OPERATE tab, select PART FOR SUBMISSION. A file explorer will open. Navigate to where the part .MPF file is stored and copy the file to the local disc.
- 3) Once the part is copied to the local disc, select the desired print mode PRINT OPTIMIZATION or ERROR DIFFUSION/FILTERING ONLY.
- 4) Select SUBMIT PART.
- 5) Verify the correct part is loaded on the HMI.

.
While printing from a network or portable media device is possible, momentary network interruption or portable media device removal will result cycle failure and a failed print.

3.2 Printing Selected Part

- 1) Verify the desired part is loaded.
- 2) Verify the build plate is clear.
- 3) Take the FREQUENCY knob to 100%.
- 4) On the OBP, push the CYCLE START button.

.

3.3 Removing Printed Parts

- 1) When CYCLE COMPLETE is displayed on the OPERATE tab of the OBP, select BUILD PLATE LOAD/UNLOAD | PROCESS SHUTDOWN tab on the HMI.
- 2) Select UNLOAD PART. The print bed will move to the side door, and the side door will unlock.

.

WARNING: BUILD PLATE TEMPERATURE IS IN EXCESS OF 450 °C. EXERCISE EXTREME CARE WHILE HANDLING.

- | | |
|---|--|
| <ol style="list-style-type: none"> 3) Open the side door then, using the build plate cart, lift the build plate and remove it from the print bay. 4) Slide the side door shut then select UNDOCK CART on the HMI. | <p>.</p> <p>Revised with new build plate dunk cart. Door Shut -> Dunk part -> Reset 5sec -> Load build plate button -> open door -> Load plate in -> close door.</p> |
|---|--|

CAUTION: EXCESSIVE FORCE MAY DAMAGE PRINTED PART.

- | | |
|--|--|
| <ol style="list-style-type: none"> 5) Remove the part from the build plate by lowering the build plate into a pool of water or by spraying with a stream of water. Agitating the part with a rubber mallet may be necessary to remove the part from the print bed. 6) Once the part is removed from the build plate and the build plate is clear, dock the build plate cart to the Xerox ElemX and lower the build plate back onto the heating element. Ensure the build plate is properly aligned. 7) Select LOAD BUILD PLATE on the HMI and ensure the build plate latches properly grasp the build plate. 8) Select PLATE INSTALLED on the HMI then select the OPERATE tab and select BUILD PLATE LOADED. 9) To print additional parts, perform the following: <ol style="list-style-type: none"> a) Verify the START button on the OBP is flashing green then press the START button. b) On the OBP, press the HV ENABLE button then wait 10 seconds. c) On the OBP, press the EPC ENABLE button. d) proceed to section 3.1. | <p>.</p> <p>CAUTION: build plate may still be extremely hot.
Misalignment of the build plate can result in the build plate latch arms not seating correctly. Ensure proper alignment of the build plate.</p> |
|--|--|

button and observe the green light go out.

- 2) On the OBP, press the HV ENABLE button and observe the green light go out.

4.3 Print Head & Print Bed Cooldown

- 1) Select the THERMAL CONTROLS tab on the HMI then select HEAD STOP HEATING and BED STOP HEATING.

- 2) On the OBP, press the BUILD HEATER ENABLE button and observe green light go out.

- 3) When the print head temperature reads 50 °C and lowering, press the UPPER HEATER ENABLE button on the OBP and observe the green light go out.

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Can lead to unresponsiv BP Bug

Securing power to the upper heater before temperatures are reduced below 50 °C can induce cooldown rates that could damage the heater.

4.4 Securing the Chiller & Argon Flow

- 1) On the OBP, press the ARGON button and observe the green light go out.

- 2) Secure external argon to the Xerox ElemX.

- 3) Select the THERMAL CONTROLS tab on the HMI. When the print head temperature reached below 45 °C, select EMERGENCY CHILLER STOP.

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4.5 Securing the HMI & Xerox ElemX Power

- | | |
|--|-----------------------|
| 1) On the OBP, depress the red E-STOP button. Turn Feed Knob to 0 and Wire Feed to 50 (lowest) | . |
| 2) Select the EXIT tab on the HMI then select EXIT RUNTIME. | |
| 3) Shutdown the PC by selecting shutdown from the start menu. | Wait for black screen |
| 4) Take the power selector switch on the back of the Xerox ElemX to OFF. | |

5 Casualty Procedures

5.1 FIRE

- | | |
|---|---|
| 1) Depress the EMERGENCY STOP PUSH
BUTTON. | . |
| 2) Secure power to the Xerox ElemX by taking
the power selector switch on the back of
the Xerox ElemX to OFF. | |
| 3) Fight the fire. | |

5.2 Casualty TBD

- | | |
|---|---|
| 1) Depress the EMERGENCY STOP PUSH
BUTTON. | . |
| 2) ... | |
| 3) ... | |

5.3 Casualty TBD

- | | |
|---|---|
| 1) Depress the EMERGENCY STOP PUSH
BUTTON. | . |
| 2) ... | |
| 3) ... | |

5.4 Casualty TBD

- | | |
|---|---|
| 1) Depress the EMERGENCY STOP PUSH
BUTTON. | . |
| 2) ... | |
| 3) ... | |

6 Maintenance

6.1 Daily Maintenance

1) TBD		.
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6.2 Weekly Maintenance

1) TBD		.
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6.3 Monthly Maintenance

1) TBD		.
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6.4 Quarterly Maintenance

1) TBD		.
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6.5 Annual Maintenance

1) TBD		.
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6.6 As Required Maintenance

1) TBD		.
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