Kettering University

Climate Change Project Final Report

Annie Liang, Brenden Londeau, Matthew McMahon

MECH-322 - Fluid Mechanics

Dr. J. Berry

With an honorary mention to Dr. T. Atkinson

16 June 2023

Abstract

This report will explore the design process of a high pressure vessel using ASME standards whenever possible. This was done by considering the possible types of forces or pressures that could act upon the vessel and calculating the resulting stress from them. These stresses were then used to determine the material strength to vessel thickness ratio required to meet a factor of safety of 2.5. After designing the vessel, a fluid dynamics analysis was conducted for a hypothetical situation where the vessel is punctured. This analysis determined the resulting stress in the walls of the vessel when hydrogen gas is leaking out of the vessel due to the puncture and the wall stress as storage tank refills.

Introduction

There are two different categories of design requirements that the team had to take into account, which were initial vessel conditions and contextual factors. The initial design conditions for this vessel required it to hold radioactive hydrogen at 1500 bars in a spherical storage tank with an inner diameter of 0.3 meters. Additionally this hydrogen is held at 400°C and is used by a rocket propulsion company for advanced interstellar transport technology. The vessel must be designed to operate at these initial parameters following relevant ASME standards.

In addition to these conditions, the vessel was also expected to survive hurricane level natural disasters. This includes the winds resulting from these hurricanes along with the flying pieces of debris that are also common with high speed winds. By taking into account these initial vessel conditions along with the contextual factors, the team had to balance the relationship between the vessel's thickness and what material had the strength to hold the hydrogen.

After design was concluded, a fluid dynamic analysis was then conducted to analyze a scenario where during a hurricane the vessel was punctured by debris creating a 0.4 meters diameter hole in the storage tank.

$$V_e(t) = 20e^{-\alpha t}[\frac{m}{s}]$$

Equation 1: The given escape velocity of hydrogen.

The time it took for the pressure vessel to reach a critical pressure of 30 bars was then calculated by assuming the velocity of the escaping hydrogen was given by *Equation 1*, where α is a time constant with a value of 0.001, 0.005, 0.01, 0.1, or 0.5. Once the time to reach 30 bars was calculated, the team calculated the time it would take to refill the tank if the inlet had a diameter of 0.05 meters or 0.08 meters. The hydrogen gas was assumed to leak out at a constant velocity of $30\frac{m}{s}$ when all conditions were kept the same, including the outlet hole diameter and velocity. Additionally the team had to calculate the resulting wall stressing when the hydrogen is leaking out with no refilling. This must be done for each time constant at three different wall thicknesses based on three different wind speeds for category 2, 3, and 5 hurricanes.

Methodology

When conducting the initial research into ASME standards for pressure vessels, the team found that chapter 21 of *ASME Boiler and Pressure Vessel Code (BPVC)* layed out the design requirements for most standard storage tanks. However, while the team did manage to find equations to design thin-wall tanks in a companion guide to ASME standards, the actual standards cost around $500 to $1000, which is beyond the scope of this analysis. Additionally, the equations the team did find required information that were inaccessible. To simplify the problem so that analysis was do-able, idealized equations were used for all vessel analysis.

When considering what idealized model to use for the initial design of the vessel, it was important to consider the differences between a thin-walled model and a thick-walled model. A thin-walled model is one where the thickness of the walls is 10% or less than the diameter of the vessel. For this scenario that would mean a wall thickness of 0.03 meters, therefore this wall will have a much higher internal stress due to the high pressure. The required material strength in this case would result in difficulty finding the proper material, for an ideal strength and cost balance. The team decided that a thick-walled model for this vessel would therefore be ideal, as this would decrease the stress due to the internal pressure and allow for a more realistic material selection and overall design of such a high pressure vessel. The thick-walled model for a pressure vessel is defined by a wall thickness greater than 10% of the diameter, which means that the wall thickness used in further calculations would have to be greater than 0.03 meters.

After selecting what model to use, the team had to select what material the pressure vessel should be manufactured from. Again, the team did its best to refer to ASME standards and found *ASME BPVC* Section II Part D from the 2021 version of *ASME BPVC* that gave the yield strength of many materials used in pressure vessel construction at a range of temperatures.

However, this resource had its limitations as Section II is primarily for pressure vessels with pressures under 10 ksi whereas the team's 1500 bar vessel holds hydrogen at approximately 22 ksi. While Section III does contain details on materials for pressure vessels greater than 10 ksi, it costs hundreds to thousands of dollars and is therefore, out of the scope of this analysis. So, in order to follow ASME standards the team had to select materials from the available Section II of the ASME codes.

When considering the wind loads on the the vessel the speed of the wind, the value of C_d, and the density of the area were unknown values. During the team's research the team found that *International Build Code (IBC)* section 1609.3 described standard wind speed values when designing for wind load. In these standards the team found wind maps for the entire United States in four different categories. This wind maps are provided below:

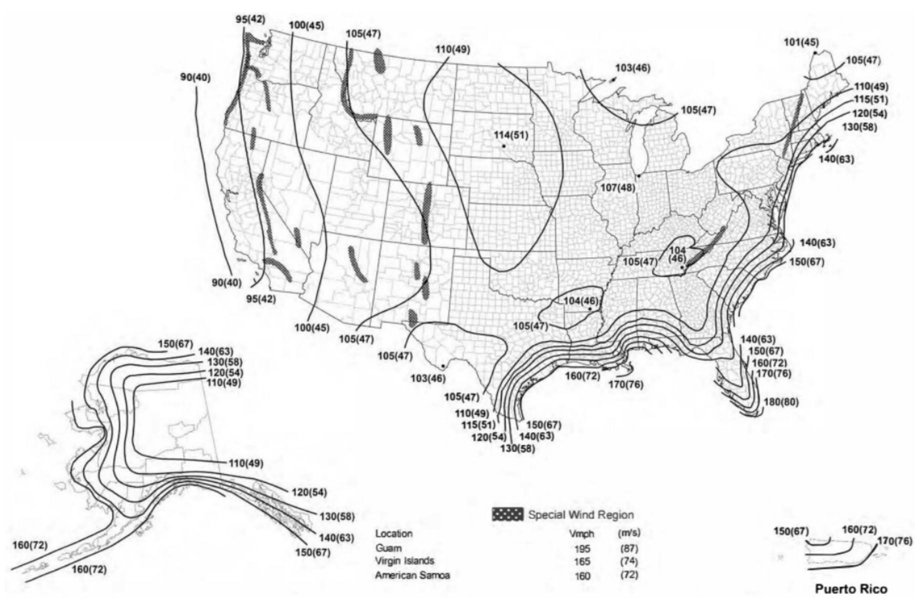


Figure 1: Basic Design Wind Speeds, V, For Risk Category II Structures

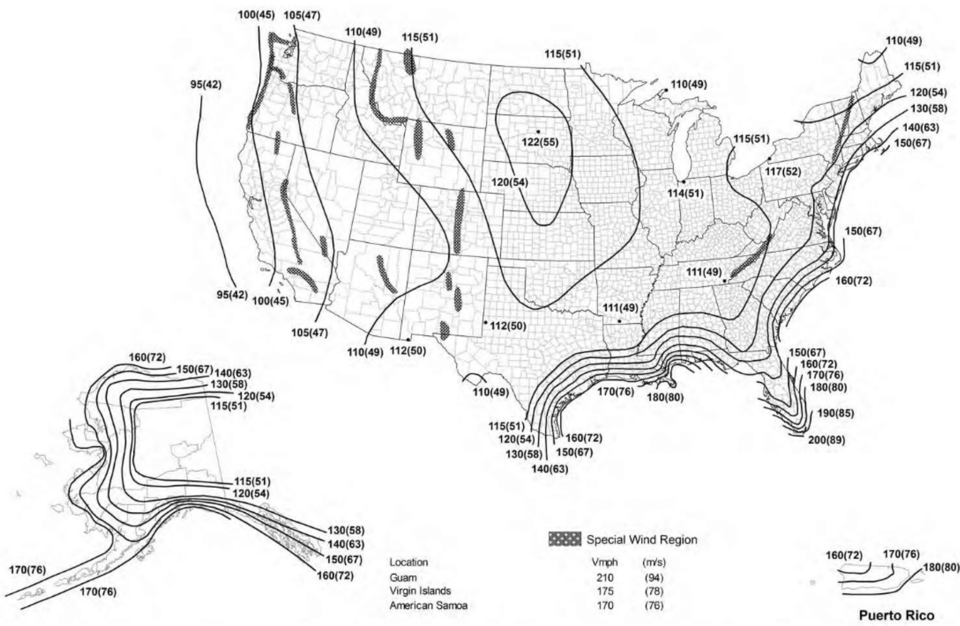


Figure 2: Basic Design Wind Speeds, V, For Risk Category III Structures

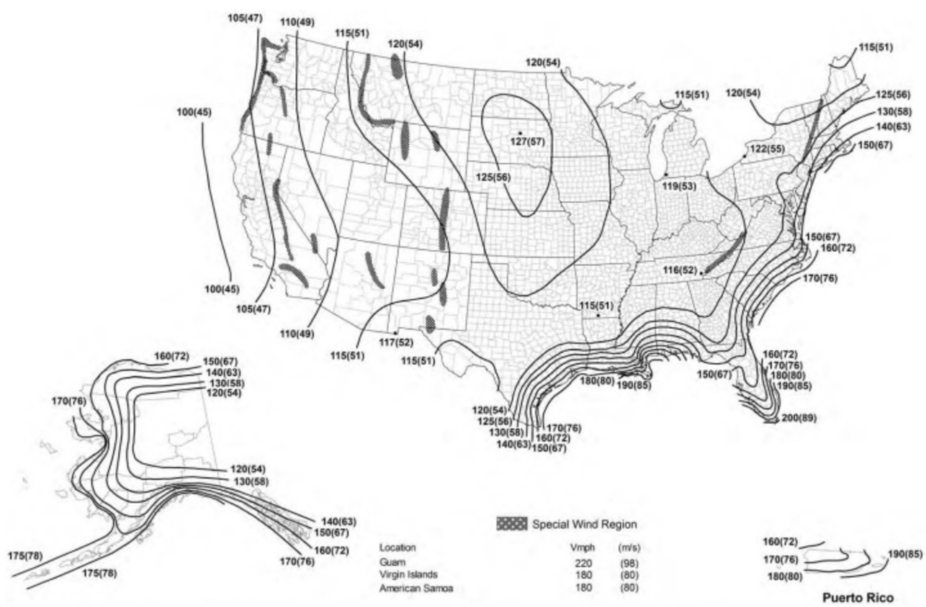


Figure 3: Basic Design Wind Speeds, V, For Risk Category III Structures

When considering the contextual factor of this being a rocket propulsion company, the team assumed that the tank would be located near a rocket launch site. In the US the two most well known rocket launching sites are NASA's Kennedy space center and SpaceX's Starbase, which when placed on these wind maps both fall into areas that contain the same wind speeds. Using these locations the team found that Category II structures should design to $150\frac{mi}{hr}$ ($67.06\frac{m}{s}$), Category III designs to $160\frac{mi}{hr}$ ($71.53\frac{m}{s}$), and Category VI designs to $170\frac{mi}{hr}$ ($76.00\frac{m}{s}$). The vessel design specification advises testing the vessels at category 2, 3, and 5 hurricane wind speeds, however, the group decided to analyze the vessel at IBC specifications in order to follow relevant standards.

In regards to the C_d value, the sphere was assumed to be perfect and this value was therefore assumed to be 0.5. Additionally, to make this analysis simpler, the team analyzed the wind loads where air is an incompressible fluid as the flow velocity is less than $100\frac{m}{s}$. This allowed the team to assume that the density of the air would remain constant at a value of 1.23 $\frac{kg}{m^3}$ for each of the different wind speeds.

Once the pressure from different properties of debris is calculated, the team analyzed how the pressure would affect the vessel walls. With assistance from Dr. T. Atkinson, the team decided that the effect that debris would have on the walls is a buckling effect. Buckling is defined as a sudden change in shape, or deformation, of a structural component under load.

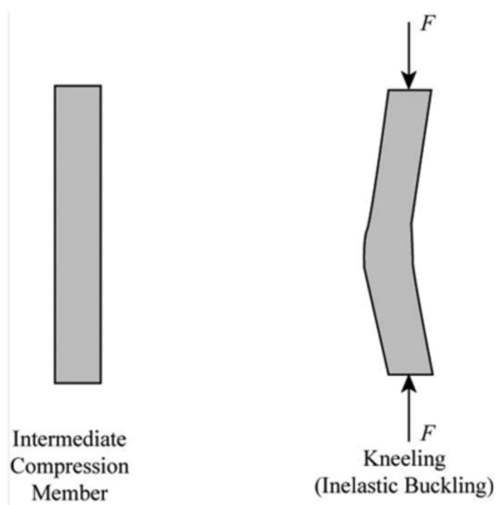


Figure 4: Diagram of Buckling on a Simple Beam

Originally, the team was given a set of thin wall equations to calculate the buckling load of the team's sphere from Dr. T. Atkinson. However, in order to account for a vessel with thicker walls, the team found research on a study done on deep sea pressure vessels, which was used to determine the buckling load for mid-thickness pressure vessels (Zhang et al. 2018). The equation derived from these experiments was taken and implemented into their analysis to create a model for a thick wall analysis.

Calculations

I. Wind Loads

The first step in finding the total stress in the vessel walls was to calculate the exterior load from different wind speeds. Based on the guidelines in the *International Build Code* Section 1609.3, the team calculated the applied load from different hurricane category wind speeds using the following equation:

$$F_{wind} = \frac{1}{2}\rho_{air} v_{wind}^2 A_{object} C_d$$

Equation 2: Formula to Calculate the Force due to Wind Speed

Equation 2 shows that the applied force is a function of the wind velocity, the density of air, the aerodynamic characteristics of the object being hit by the wind and its cross-sectional area. With the wind force calculated, the team then found and plotted the pressure on the exposed area of the spherical vessel. The results of this analysis are shown in *Figure 1*, shown below:

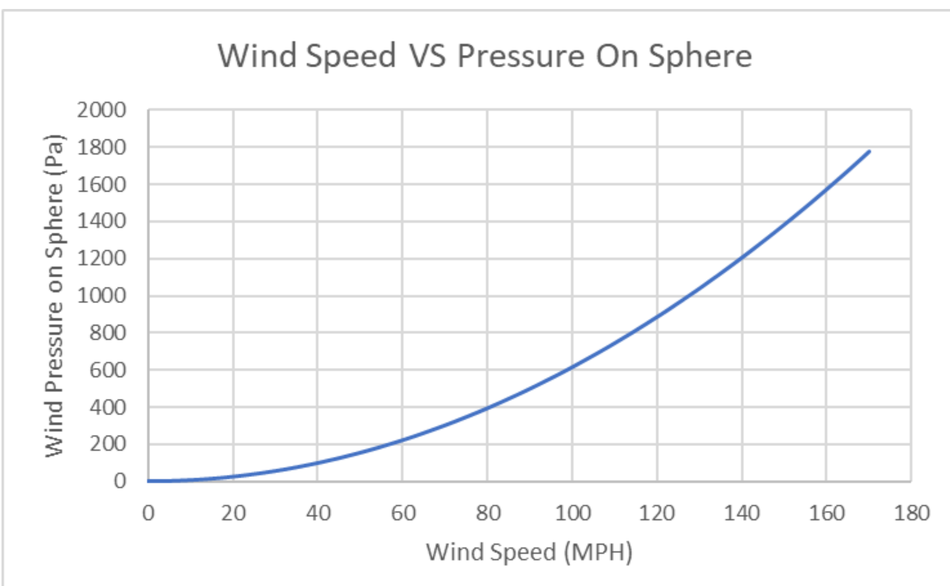


Figure 1: Graph of Wind Speed vs Pressure on Spherical Vessel

From this data the team noticed that, as expected, the wind pressure increases with increasing wind velocity. However the team also noticed that the wind pressure was simply in pascals, where the stress due to the internal pressure was in MPa. Therefore, moving forward, the team assumed that the wind force was negligible to the wall stress, though it was still included in calculations for accuracy.

II. Debris Loads

With high wind speeds during a hurricane, also comes fast moving flying debris. To account for the stress caused by any debris strikes, the team made some assumptions regarding the debris properties. The team assumed that the debris will have a circular contact area with the vessel, that the debris velocity will be the same as the wind velocity, and that the average impulse time of contact of debris strikes will be 0.005 seconds. With these assumptions the team were able to calculate the applied pressure for different debris diameters and masses, as well as the resulting buckling stress inflicted upon the vessel walls. The equation(s) for the stress caused by flying debris is shown below:

$$M_{momentum} = V_{velocity} m_{mass}$$

Equation 3: General momentum equation.

$$F_{force} = \frac{M_{momentum}}{\Delta t_{time}}$$

Equation 4: General force equation with respect to momentum.

$$\sigma_{stress} = \frac{F_{force}}{A_{cross-sectional\ area}}$$

Equation 5: General stress equation.

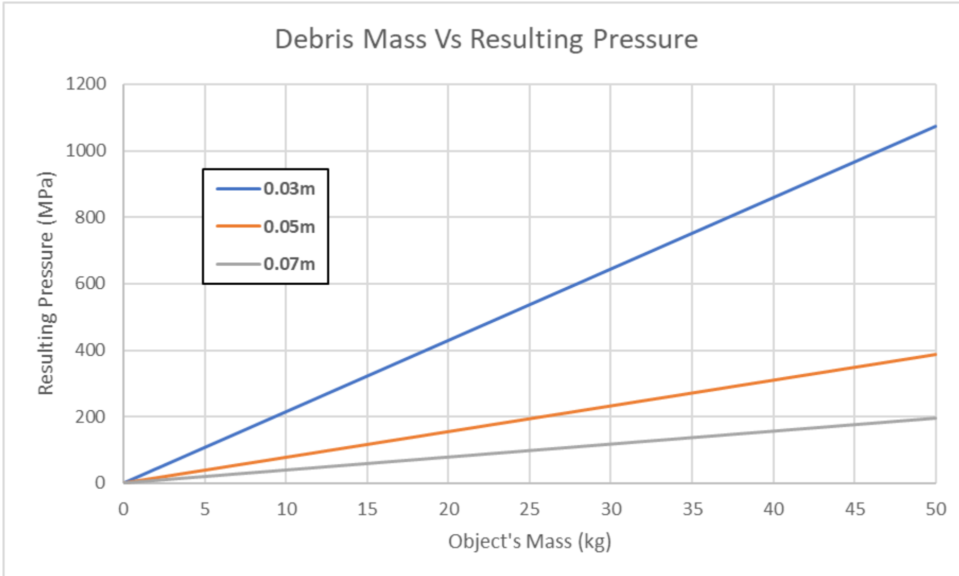


Figure 2: Graph of Debris Mass vs Resulting Wall Pressure

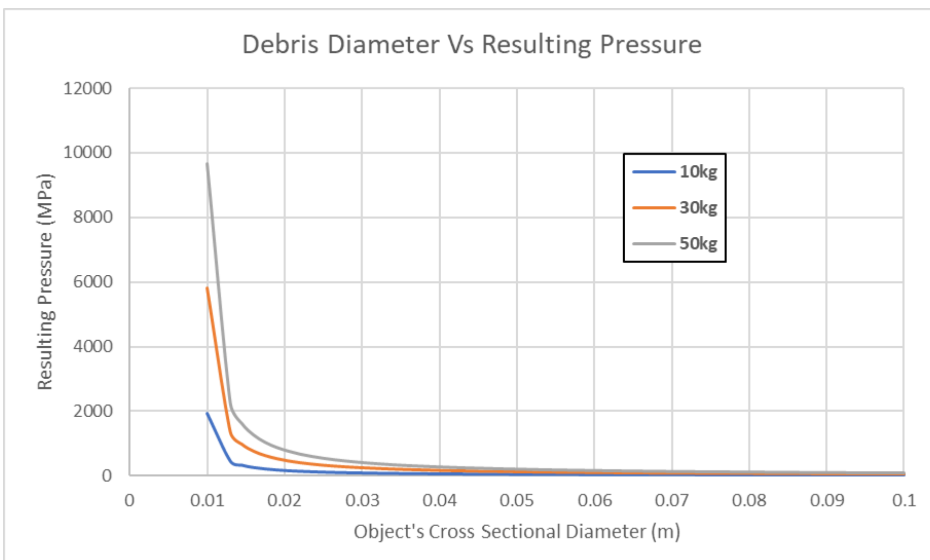


Figure 3: Graph of Debris Diameter/Cross-Sectional Area vs Resulting Wall Pressure

Using *Equations 3, 4, and 5* with varying debris mass and diameter resulted in the stress distributions shown in *Figures 2 and 3*.

The buckling load, or allowable load before an object buckles, of a spherical structural component is a function of the material elastic modulus, the thickness of the walls, the median radius of the sphere, and the poisson ratio of the material. The buckling load is therefore given by the equation:

$$p_{m-t} = \frac{2Et}{r(1-v^2)}\left[\sqrt{\frac{(1-v^2)}{3}\frac{t}{r}} - \frac{vt^2}{2r^2}\right]$$

Equation 6: Buckling of a Spherical Component under Load

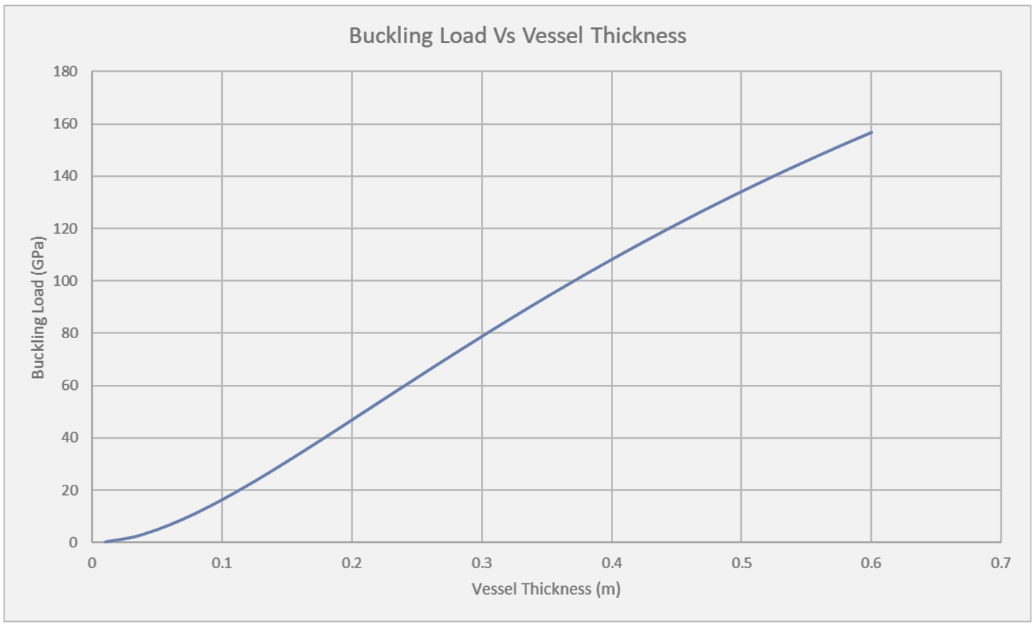


Figure 4: Graph of Wall Thickness vs Buckling Load

With this formula the team calculated and graphed the buckling load for a range of wall thicknesses. A thicker wall results in a higher buckling load, which means that the vessel can withstand a higher load before it begins to buckle. Extrapolating from the results, the team saw that the minimum wall thickness, accounting for a buckling strength of 10 GPa and a safety factor of 2.5, was approximately 0.16 meters. These results are shown in the graph above.

III. Mass Continuity

The final step of analyzing this pressure vessel is to find the time to reach the critical pressure once the exit hole is created by debris, to refill to the initial pressure, and then to calculate the varying wall stress over the time that it takes to reach the critical pressure. As a quick review, the vessel is ruptured by debris which results in an exit hole of 0.04 meters, which causes the vessel to leak its contents until reaching an internal pressure of 30 bar. The exit velocity of the gas is time varying, and is defined in *Equation 1*.

$$t = -\frac{ln[ln(\frac{P_f}{P_i})*(\frac{\alpha A}{20A}+1)]}{\alpha}$$

Equation 7: Time to Reach Critical Pressure.

Using the principles of mass continuity taught in fluid mechanics, the team found the time to reach critical pressure shown in *Equation 7*. This equation is a function of the initial pressure P_i as well as the critical pressure P_f , α the time constant, and *A* the area of the exit hole. The time to reach critical pressure was plotted against different time constant values, which resulted in the curve shown in *Figure 5*.

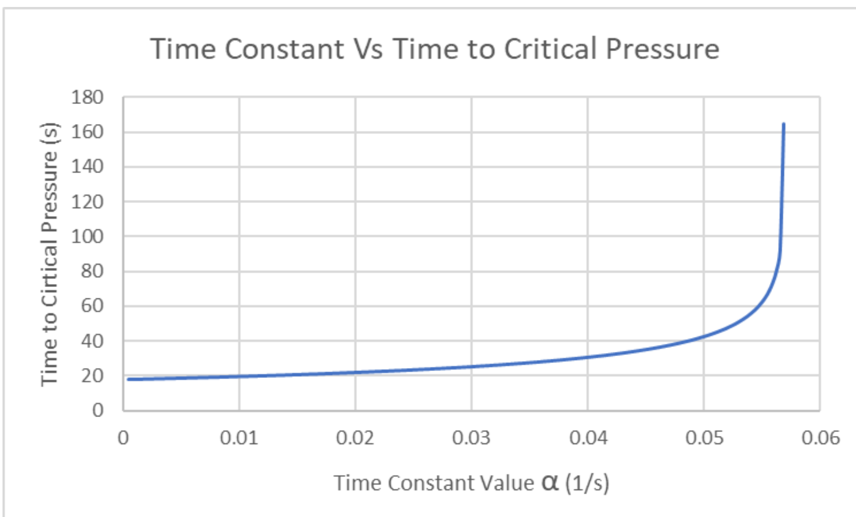


Figure 5: Time Constant Value vs Time to Reach Critical Pressure

Once the time to reach critical pressure was calculated and plotted, the next step was to calculate the time it takes to refill the vessel back to its original pressure. This calculation is also done with the principles of mass continuity with the addition of an inlet mass flow, shown in *Equation 8*.

$$v ln(\frac{P(t)}{P_i}) = (A_{in} * V_{in} * t) + \frac{20A_{exit}}{\alpha}(e^{-\alpha t} - 1)$$

Equation 8: Time to Refill Vessel to Initial Pressure

This equation is a function of the vessel volume, ν , the critical pressure ($P(t)$), the initial condition pressure of 1500 bar (P_i), the inlet and outlet hole areas, the inlet velocity, and the time constant α. Due to the multiple occurrences of the variable for time, this equation must be solved iteratively.

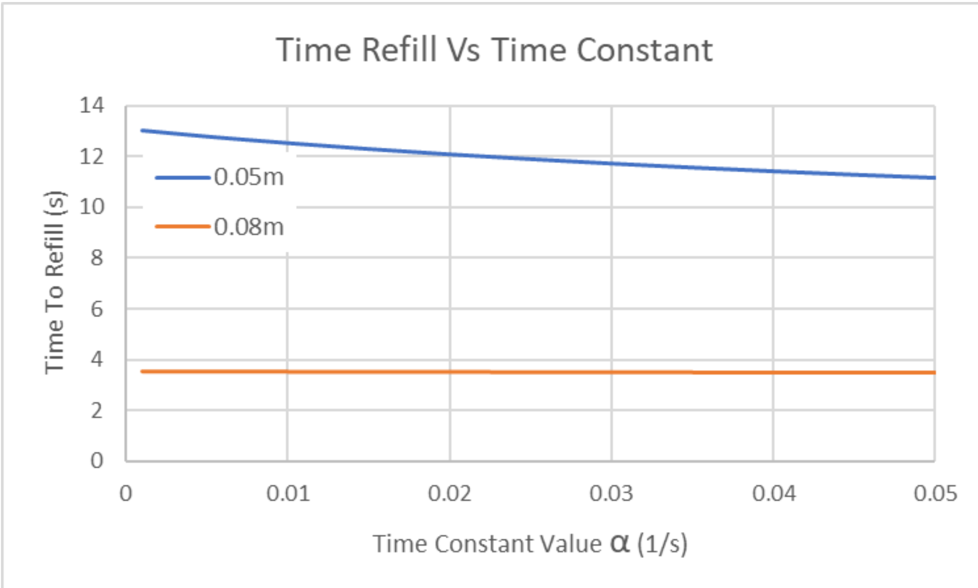
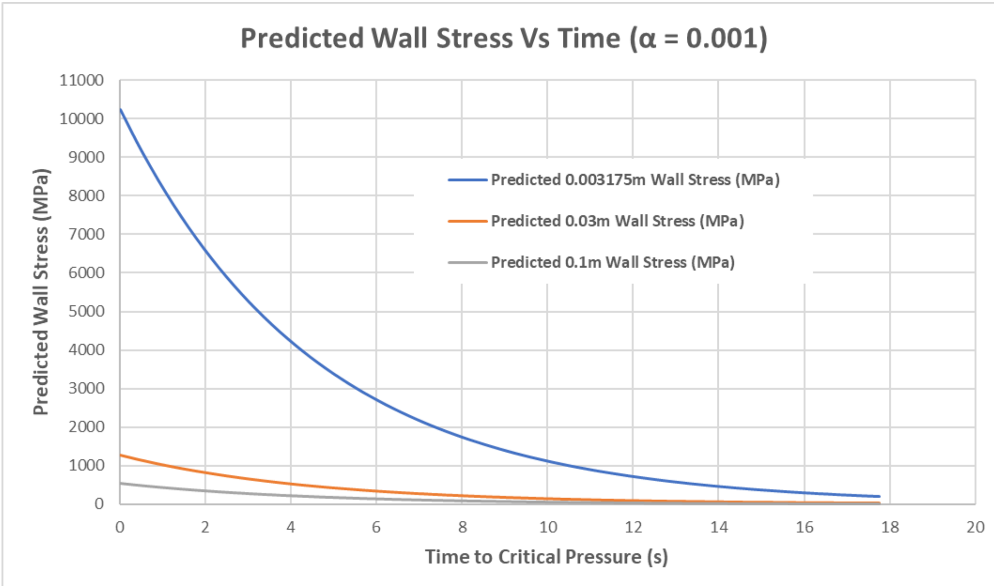


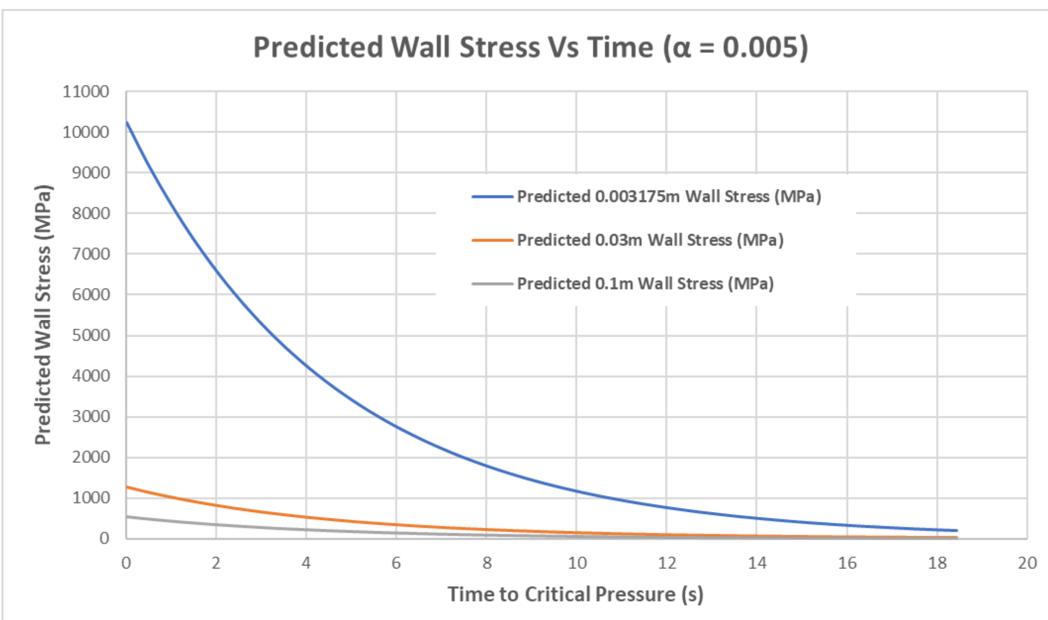
Figure 6: Graph of Time to Refill Vessel vs Time Constant Values

Once the equation is iteratively solved, the results were graphed as shown in *Figure 6* for a range of time constant values, as well as two different wall thicknesses of 0.05 meters and 0.08 meters.

IV. Wall Stress over Time

With the pressure models built, the final analysis to be done was to calculate the wall stress over time while the gas is escaping and the vessel is not being refilled. These calculations were done with three different wall thicknesses, time constants, and wind speeds to view how each variable affects the stress in the walls. The wind speeds and wall thicknesses were set to 150MPH and 0.003175 meters, 160MPH and 0.03 meters, and 170MPH and 0.10 meters. For the wall stress calculations, *Equation 7* is rearranged to solve for P(t), and is then plotted with time as the independent variable. Therefore, the equation for wall stress is a function of the same variables as for the time to reach critical pressure, plus time itself. The wall thicknesses of 0.003175 and 0.03 meters are both within the scope of a thin-walled vessel, and these values result in a wall stress greater than 1000MPa at peak pressure. This high stress is why the team decided to pursue the analysis within a thick-walled model. The 0.10 meter thickness results in a peak pressure of approximately 500MPa. The wall stress vs time graph for each different time constant is shown in *Figure 7,* along with a curve for each different thickness included in this calculation.





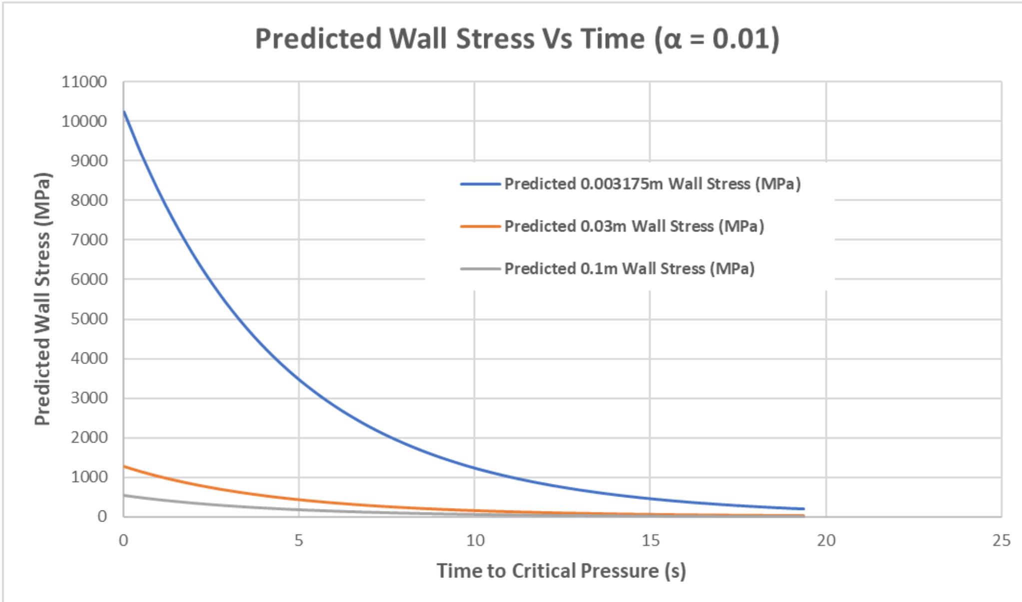


Figure 7: Graphs of Wall Stress vs Time - Three Thicknesses and Three Time Constants

Conclusion and Results Discussion

Once the model and calculations are complete, the results can be reviewed and discussed to decide on a material and how fluid mechanics can be beneficial to the creation of many machines.

The most important consideration for the material selection is the stress due to the internal pressure, followed by the buckling from flying debris. First, the stress from internal pressure up to a wall thickness of approximately 0.1 meters is more than 1000 MPa after accounting for the factor of safety of 2.5. This means that the vessel must have a wall thickness greater than 0.1 for a realistic material to be used. From the buckling calculations, the team found that the vessel must have a thickness of at least 0.16 meters to withstand the average load from flying debris. Inserting this thickness into the calculation for pressure stress, the resulting stress in the wall is approximately 1100MPa after multiplying by the safety factor.

When researching what materials are used to construct pressure vessels, the group found that carbon steels and stainless steel are very common due to their relatively high strength. However when looking in *ASME BPVC* Section II Part D the team found that on average stainless steels have yield strengths ranging 100 MPa - 150 MPa at 400°C while carbon steels range slightly higher at 150 MPa - 180 MPa at 400°C. Upon realizing that standard materials are not capable of handling the resulting stress the group realized that a more exotic material is required. After doing more research on the material yield strengths in the ASME material guidelines for pressure vessels, the team found that UNS N07718 was the closest to 1000 MPa yield strength at 400°C when compared to other code approved materials. UNS N07718, otherwise known as inconel 718, is an extremely strong material which has a yield strength of 922MPa at 400°C. To bring the stress in the vessel walls closer to this yield strength, the wall thickness was increased to 0.275m, or 92% of the diameter of the vessel. This high percent wall thickness is required because of the small size of the vessel combined with the high internal pressure. While inconel is a strong material that can withstand the usage described for this vessel, it is also expensive and difficult to manufacture. The cost of this material, however, can be justified by the situation and task the vessel must endure. The high pressure, radioactive hydrogen that is contained in the vessel would be extremely dangerous if the vessel were to fail and leak into the environment, therefore the high expense of inconel is more than justified to keep the environment and people safe. If the vessel were to rupture, however, the wall stress and pressure would still follow the same curve shown in *Figure 7* dispute the thicker walls. Additionally, the mass continuity calculations are not affected by the wall thickness.

Once the design concept is finished and a material is selected, the principles of fluid mechanics and mechanics of materials can be applied to the vessel operation and create safety measures. One of these safety measures can monitor for a leak in the vessel by utilizing the principles of mechanics of materials. As the team found from the fluid mechanics study, the stress in the walls will decrease as the internal pressure also decreases due to a leak. The stress in the walls can be measured and monitored using a strain gauge, which relates to the stress in the walls through Hooke's law. As the gauge sees a decrease in wall stress, it is correlated through the study done on mass continuity in the ruptured vessel to the internal pressure, and therefore can act as a monitor for a leak in the vessel. This combines principles of fluid mechanics and mechanics of materials to create a safe and effective system to contain the volatile materials for this application.

References

"2020 ICC 500 ICC/NSSA STANDARD for the DESIGN and CONSTRUCTION of STORM SHELTERS | ICC DIGITAL CODES." *Codes.iccsafe.org*, Dec. 2020, codes.iccsafe.org/content/ICC5002020P1/chapter-3-structural-design-and-testing-criteria.

"2021 INTERNATIONAL BUILDING CODE (IBC) | ICC DIGITAL CODES." *Codes.iccsafe.org*, 23 Oct. 2020, codes.iccsafe.org/content/IBC2021P1/chapter-16-structural-design#IBC2021P1_Ch16_Sec1609.

American Society of Mechanical Engineers. Companion Guide to the ASME Boiler & Pressure Vessel Code: Criteria and Commentary on Select Aspects of the Boiler & Pressure Vessel and Piping Codes. Vol. 2. 5th ed. American Society of Mechanical Engineers, 2017.

Bank, Eric. "Ontario Energy Board on How to Calculate Wind Loads from Wind Speeds." *Bing*, 27 May 2018, www.rds.oeb.ca/CMWebDrawer/Record/646733/File/document. Accessed 13 May 2023.

Holmes, John D. "Wind Loading of Structures." *Google Books*, 2015, books.google.com/books?hl=en&lr=&id=y4WbBgAAQBAJ&oi=fnd&pg=PP1&dq=%2Blargest%2Bload%2Bapplied%2Bby%2Bflying%2Bdebris%2Bduring%2Ba%2Bhurricane&ots=f_4RtfC8S2&sig=CEzjiInm19ZHk1vlXSdh-jCYnek#v=onepage&q&f=false

Johnson, Lee. "How to Calculate Wind Loads From Wind Speeds" sciencing.com, https://sciencing.com/calculate-wind-loads-wind-speeds-6104140.html. 12 May 2023.

Rabia, S. (2021). ASME BPVC Section II Materials Part A Ferrous Material Specifications (SA 451 to End). *www.academia.edu*. https://www.academia.edu/90708927/ASME_BPVC_Section_II_Materials_Part_A_Ferrous_Material_Specifications_SA_451_to_End_

Thick pressure vessels. Thick Pressure Vessels | Engineering Library. (n.d.). https://engineeringlibrary.org/reference/thick-pressure-vessels-air-force-stress-manual

Zhang, J., Zhang, R., Cui, W., Tang, W., Wang, F., & Pan, B. (2018). *Elastic-plastic buckling of deep sea spherical pressure hulls* (ISSN 0951-8339). Marine Structures. https://doi.org/10.1016/j.marstruc.2017.09.007