

A Case Study of Structural Industrial Pressure Vessel Under Wind Load

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Abstract

Tall structures that are in the open atmosphere are subjected to self-weight and wind loads. The investigation of an industrial pressure vessel can be vital from the security perspective based on its configuration such as its buttress at the base level and wind speeds. In this study, an industrial pressure vessel with a relatively low aspect ratio (height versus width) subjected to varying 'steady' wind loads has been analyzed to check for its dynamic stability. The pressure vessel in this current condition is standing on its three equidistant legs. The pressure vessel is assumed to be filled with material to simulate critical condition in the case of static and strength analysis. SolidWorks and ANSYS software were used to compare the results of this analysis. The analysis shows the support is sustainable under the loading as recommended with a factor of safety near 1.2 for a critical condition. The stability of the structure was investigated utilizing dynamic analysis. Various 'steady' wind speeds were investigated with emphasis on the maximum wind velocity recorded locally in the last 50 years of 120 mph. The vortex shedding frequencies were calculated for the structure at various wind speeds to determine whether a 'steady' wind would induce a resonance condition. The vortex shedding frequency depends on Strouhal and Reynold's numbers, 'steady' wind speed, and the diameter of the obstructing body in this case. Theoretical calculations and software generated results were utilized for this computation. Based on the results, the current pressure vessel design is found to be safe under operating conditions. A parametric study was performed with different design configurations with similar cost of pressure vessel support structures to improve the stiffness of the system. In the future, an experimental study of vibrational measurements will be performed on a scaled down pressure vessel model utilizing a wind tunnel. A specialized software designated MecaStack will be used for vortex shedding effect analysis.

Introduction

This report presents a mechanical analysis of a provided design for a pressure vessel from the Johnson and Matthey Process Technologies in Savannah, Georgia. Several models for the pressure vessel were devised and then compared by considering the cost, construction, accessibility, and sustainability. The analysis was focused on the strength of the support system as well as the

stability of the pressure vessel under severe conditions. The structure would be exposed to wind load due to it being installed in open air. For this reason, vibrational effects on this type of structure may be a concern for the designers. The pressure vessel rests on three load cells for accurate measurement of its slurry catalyst content. The first part of the analysis was based on the structural integrity of the supporting load cells under the fully loaded condition. This analysis process was carried out utilizing SolidWorks and ANSYS software. The software results were compared to illustrate consistency. The second part of this study focused on the dynamic stability of the of the pressure vessel under constant wind load. Air flowing past a body at a certain velocity will create vortices at the rear of that body initializing an oscillating flow. This oscillating flow depends on the size, shape and structure of the blunt body obstructing the flow of air.

The oscillating flow is known as vortex shedding and its frequency is known as the vortex shedding frequency ^[1] A resonating condition may arise resulting in significant damage as the vortex shedding frequency approaches the natural frequency of the structure^[2-4].

In the current research project, the natural frequency of the structure was estimated using software simulations. The natural frequency was compared with the possible vortex shedding frequency arising due to severe wind. In the end, a variety of leg cross-sections were analyzed to determine any notable development in the structure's natural frequency. Those results are also included in this report. In the future, further study into physical experiments is highly recommended.

Problem Definition & Scope

The structural analysis of a 200 CF Pressure Vessel designed for Johnson Matthey Process Technologies (JMPT) was the major focus of this project. The pressure vessel was designed with three W8x31 legs that are anchored to a ground structure using a structural assembly suitable to meet the structural loads experienced. The load cell is a part of a ground support assembly structure that anchors an 8.5 Ton Loader at 3 points in a circular arrangement. This anchoring is required to resist wind and seismic effects that may act on the structure specified at the installation location.

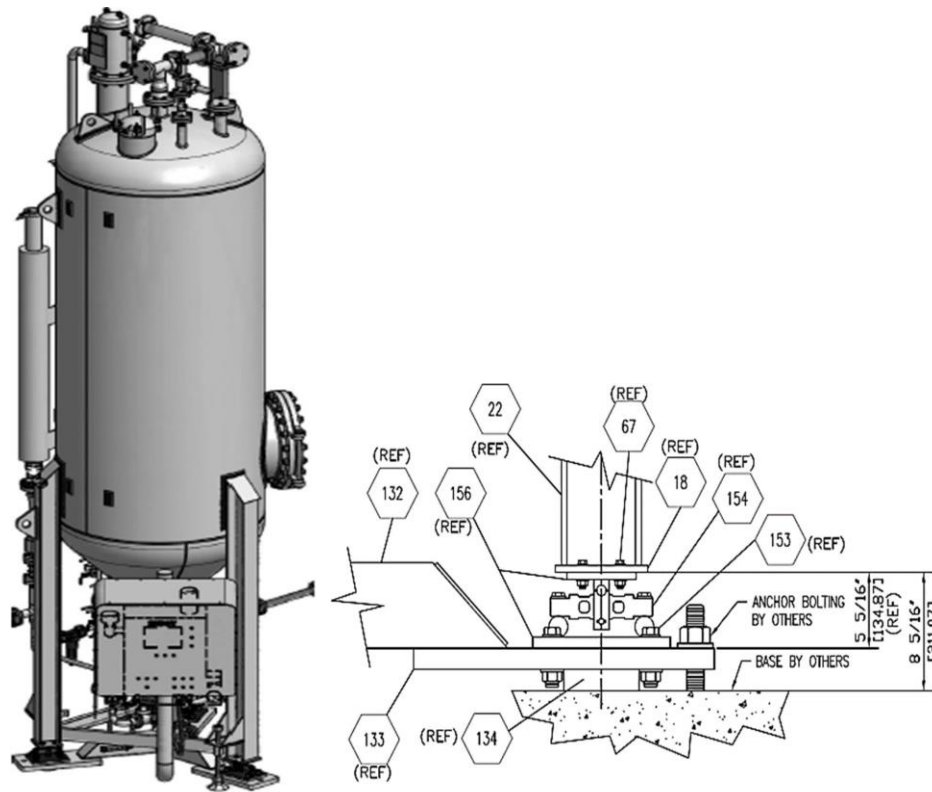


Fig 1. 200 GM Pressure Vessel Isometric View & Details of Load Cell Assembly

These W8x31 legs are welded to a 1/2 SA-516-70 plate on the other side; bolted with four 1/2-13 SA193- B7/SA194-2H bolts to the Load Cell Assembly EZ-Mount #17823 with the 10k Double Ended Beam Load cell. These load cells are bolted to the Skid frame with four 1"-8 SA193-B7/SA194-2H bolts which are bolted to the foundation via an embedded anchor bolt provided by the refinery. This configuration of structural connections is shown in Figure 1 and is itemized in Table 1 of this report. The critical elements of this design are items 154 and 156, i.e., the load cell components. These components have an individual structural qualification that meets the requirements of the design, but no analysis or testing has been carried out to assess the overall system structural capability [5].

Table 1. Components of the Ground Support Assembly Structure

REF No.	Components
67	Bolt; HH w/Nut:1/2"x2";SA193-B7/SA194-2H
153	Bolt; 1"; -8 UNC; 3 1/2" LG.; W/Nut; SA-193-B7/ SA-194-2H
154	Load Cell; 10 Klb.; ATEX
155	Dummy Load Cell; 5 to 20 Klb.
156	Load Cell Mount; 5 to 20 Klb.

From a model provided by JMPT, it is analyzed that the top and bottom faces of the bolts are fixed. There should be no penetration contact between the load cell and the rollers. The downward forces as the result of the weight of the tank body, water and catalyst would act on the top face of the mounting block as shown in Figure 2.

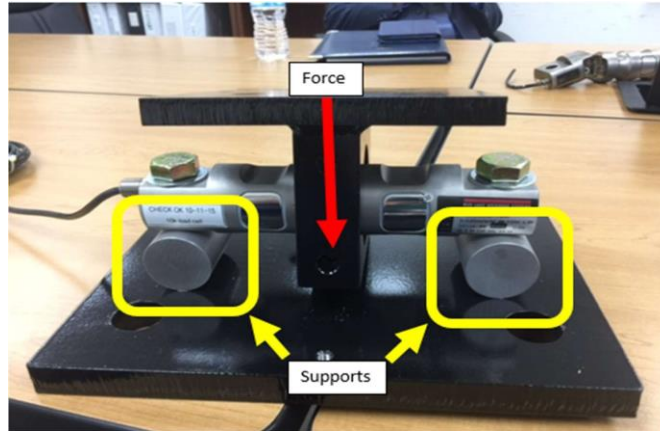


Fig 2. Force on Load Cell with EZ Mount 1

The structural integrity of the 200-GM-013 Loader was assessed by the Finite Element Method by two of the most popular software, ANSYS, and SolidWorks.

Table 2: EZ Mount 17823 Load Capacity According to JMPT

Load Direction	(lb.)
Shear/Side	25,950
Uplift	29,100
Overload	64,250

The major problem of this specific pressure vessel is its sustainability in the open air. As JMPT deals with catalysts needed in crude-oil refineries, most of the time this kind of pressure vessel is required to be installed in the open air.

Wind load analysis is one of the most critical factors to be considered to ensure the structural safety of this pressure vessel. The probable impairment, inconvenience or aids, and results from wind can be predicted from the wind load analysis ^[6].

The static analysis due to vertical loads and the dynamic effects of wind loads on the pressure vessel was the primary focus of this study. The impact of varying loads on the load cells should be known to have a clear concept about the static effects of various loads on the pressure vessel. The dynamic effects are essential for large, moderately tall and high aspect ratio structures as the wind is less affected by the terrain roughness above the earth surface. In upper altitude, wind gusts create a varying dynamism on tall structures which induces vibrations and oscillations. Also, fluctuating crosswind forces can be induced by vortex shedding with a discrete frequency depending on the shape and size of the structure ^[7-8]. At natural frequency, a system will oscillate by itself without a constant outside stimulating factor.

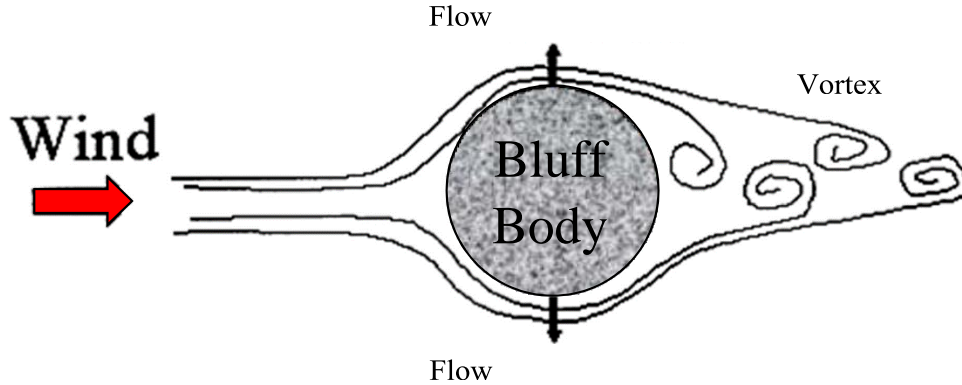


Fig 4. Vortex Shedding Frequency

Vortex shedding is a phenomenon of oscillating flow according to fluid dynamics. For any outdoor structure, air interacts with the solid body at different velocities depending on the shape of the body, altitude, weather condition, geographical position of the structure, etc. At the time of interaction, circular motion of air flows past the blunt body. This unstable separation of flow creates downstream vortices at the rear of the solid structure which then detaches intermittently.

This flow creates a low-pressure zone at the rear of the object. The object tends to move towards that low-pressure zone naturally. This phenomenon is defined as vortex shedding ^[9-10]. If the vortex shedding frequency coincides with the natural frequency of the assembly, it may augment the oscillation or vibration and causes failure or severe damage to the structure. For that, a thorough study on the natural frequency and vortex shedding is a mandatory thing to do for structures exposed in the atmosphere. Authors were charged to study this particular pressure vessel used by the JMPT under severe static loading and high wind conditions.

Design Description, Calculation & Analysis

The static analysis was done using SolidWorks and ANSYS to check the stability of the load cells under varying loads. Pre-established equations were used to calculate the vortex shedding frequency for the varying wind speeds. ANSYS analysis was used to determine the possible natural frequency of the pressure vessel under varying wind speeds and support types. Natural frequencies and vortex shedding frequencies were then compared.

Static Analysis of the Load Cell

According to the dimension and specifications provided by JMPT, the load cell is designed and analyzed using SolidWorks & ANSYS Workbench. In this project, the structural integrity analysis is done using both the software for consistency. The load cell itself is a Double-ended Beam made of Stainless Steel, NTEP Certified 1:5000 Class III/1:10,000 IIIL Multiple Cell, IP67. Static

Analysis of the load cell is done based on its Load Rating provided by JMPT. According to the data, with all the piping and accessories,

1. Load provided by the Empty Vessel is 9683.6 lb.
2. Load provided by a Water Filled Vessel is 24600 lb.
3. Load provided by a Catalyst Filled Vessel is 21623 lb.

These loads are applied during different calculations and analysis throughout the whole project.

SolidWorks & ANSYS Analyses

A downward force is applied to the top face of the mounting block. A fine mesh control is used to the exterior cutouts of the load cell where the most stress was expected. Load Cell Assembly Analysis, analysis with 10,000 lb. Load, analysis of the Empty Condition, Catalyst Filled Condition & Water Filled Condition using SolidWorks was done. The model was exported from SolidWorks as a STEP file and imported into ANSYS. The top of the bolts were made fixed, and a force was applied to the top of the mounting block. A patch conforming mesh was used on the main body of the load cell, and a refined mesh was used for the external cutouts. Same conditions were used in ANSYS, and the analysis was done.

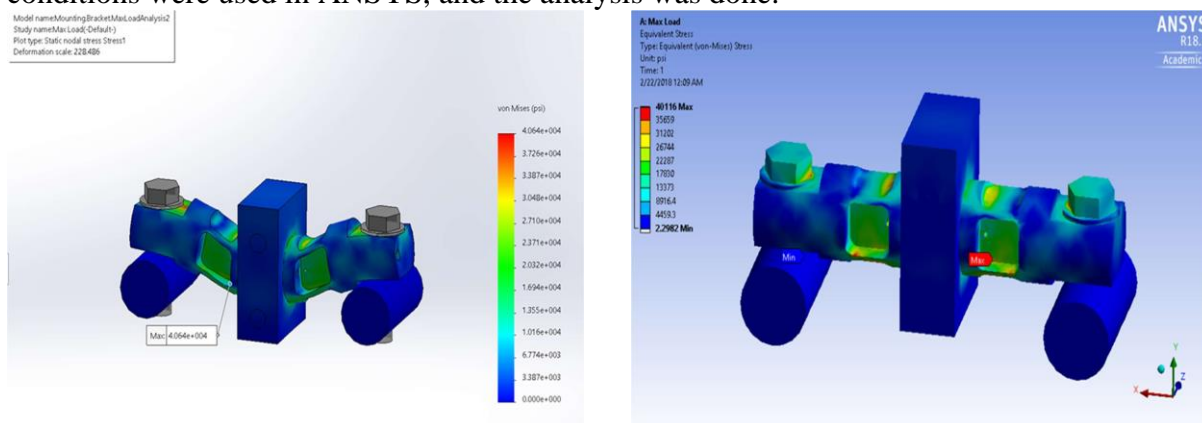


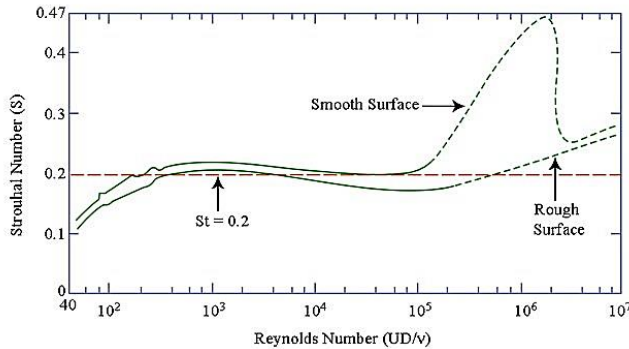
Fig 7. Load Cell Analysis with 10,000 lb. using SolidWorks & ANSYS

Table 3: Comparisons between the SolidWorks & ANSYS Analysis

Load Conditions	SolidWorks Analysis Results Von Mises Stress (Psi)	ANSYS Analysis Results Von Mises Stress (Psi)
Empty Condition	13120	12964
Catalyst Filled Condition	29300	28928
Water Filled Condition	33300	32981
10000 lb. (Rated load)	40640	40116

Calculation of Vortex Shedding Frequency

The vortex shedding frequency that was calculated for comparisons was determined with the following equations. Eq. 1 determines the Reynolds number. The Strouhal number is a dimensionless number representing the oscillating flow mechanisms ^[11]. This number is calculated with Eq. 2. Finally, the vortex shedding frequency is calculated with Eq. 3 for comparison with simulation results.



$$Re = \frac{U_0 D}{\nu} \tag{1}$$

$$St = 0.198 \left(1 - \frac{19.7}{Re} \right) \tag{2}$$

$$f_s = \frac{St U_0}{D} \tag{3}$$

Fig 8. Correlation of Strouhal Number and Reynolds Number ^[12]

The Strouhal Number has a crucial influence on the frequency at which vortex shedding occurs. Again, the Strouhal Number is dependent on Reynolds Number. The relation between these two dimensionless numbers can be represented by the graph obtained from MIT OCW. Data is taken from Lienhard (1966) and Achenbach and Heinecke (1981) ^[12]. The Strouhal Number remains nearly constant at approximately 0.2 unrelated to the geometry of any blunt body over a broad range of Reynolds Numbers ^[13]. The vortex shedding frequency is calculated for wind speeds of 30 mph and 120 mph. These wind speeds represent the highest wind speed in the normal range and the highest wind speed recorded in Savannah area in the last fifty years, respectively.

Taking, $St = 0.22$, $U_0 = 30 \text{ mph} = 13.4 \frac{m}{s}$, and $D = 5 \text{ ft} = 1.524 \text{ m}$, the Vortex Shedding Frequency

$$f_s = \frac{St U_0}{D} = \frac{0.22 \times 13.41 \text{ ms}^{-1}}{1.524 \text{ m}} = 1.94 \text{ Hz}$$

The Strouhal Number for Reynolds Number, 6.78×10^5 (for temp 20°C) will become approximately, $St = 0.3$.

Taking, $U_0 = 120 \text{ mph} = 53.64 \text{ m/s}$, the Vortex Shedding Frequency,

$$f_s = \frac{St U_0}{D} = \frac{0.3 \times 53.64 \text{ ms}^{-1}}{1.524 \text{ m}} = 10.56 \text{ Hz}$$

Natural Frequency Analysis of the Pressure Vessel

The natural frequency analysis of the pressure vessel with the provided design specification was completed in ANSYS workbench. The analysis was done considering three possible assemblies. They are-

1. Pressure Vessel with three legs, without any supports (without Cross & Parallel Bars)
2. Pressure Vessel with three legs, with Parallel Bars
3. Pressure Vessel with three legs, with Cross & Parallel Bars

These three support systems are analyzed two times using different conditions. Fixed-fixed support on the joining section of the legs with the tank and load cells. Another is fixed support on the joining part of the legs and the tank and pinned supports on the joining portion of the legs and load cells. The second one seems to be more practical due to the consideration of the wind load analysis on the tank body.

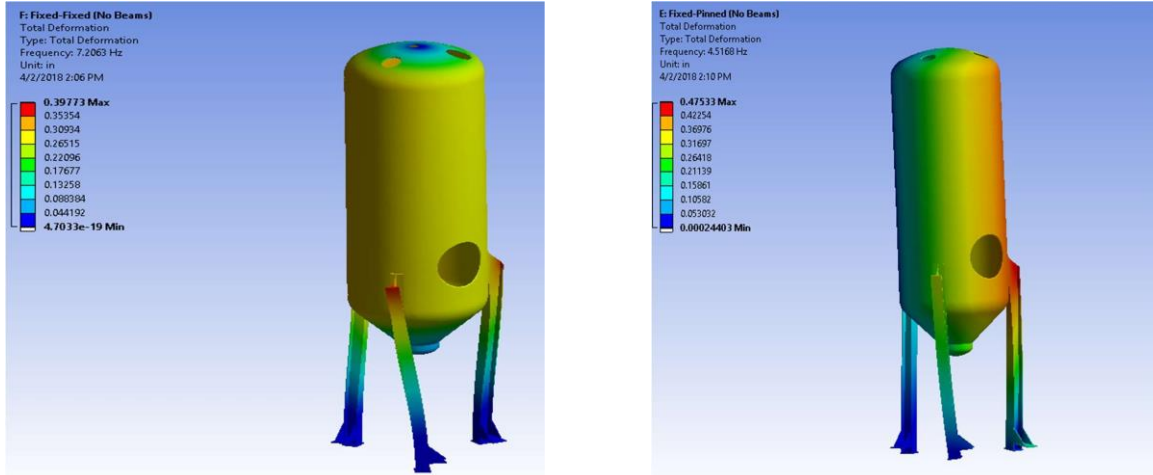


Fig 9. Pressure Vessel without Cross and Parallel Bars (Fixed-Fixed & Fixed-Pinned Support)

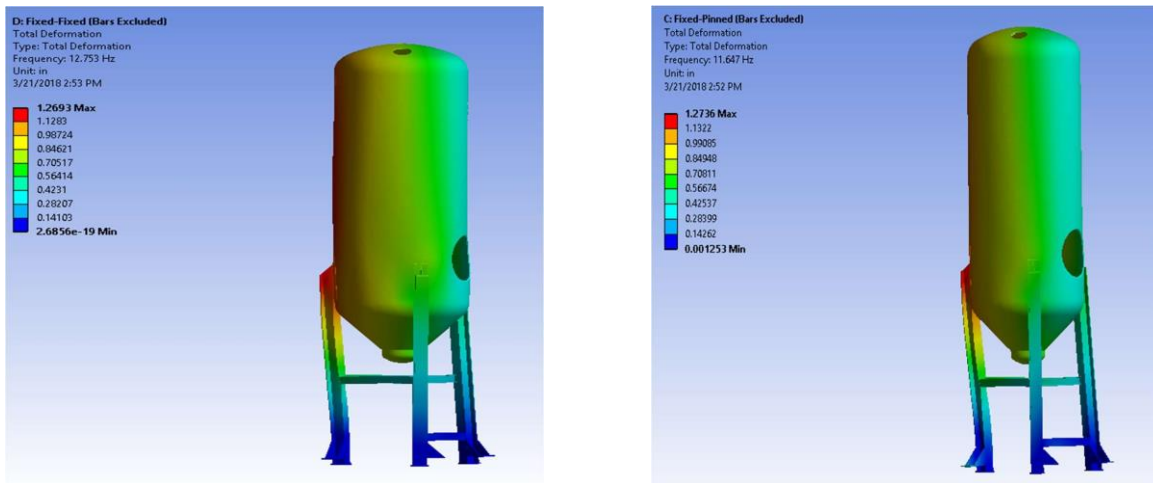


Fig 10. Pressure Vessel with Parallel bars (Fixed-Fixed Support & Fixed-Pinned Support)

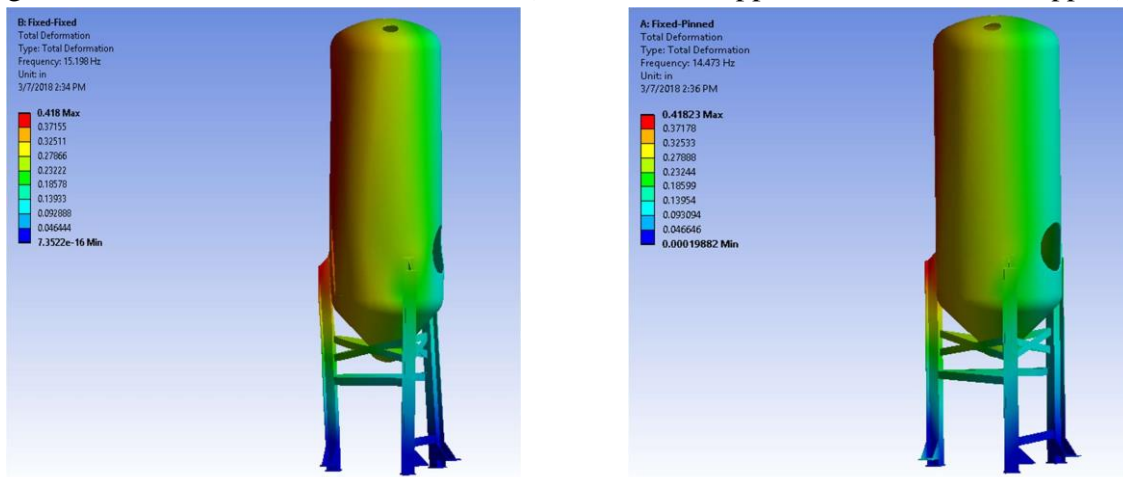


Fig 11. Pressure Vessel with Parallel and Crossbars (Fixed-Fixed Support & Fixed-Pinned Support)

Redesign of the Cross Section of the Parallel Bars

The objective of this section is a parametric study of the effect of the cross section of the beam on the natural frequency of the system. The analysis performed utilizing ANSYS Workbench 18.1 showed insignificant differences in natural frequencies of redesigned cross-sectional geometries implemented into the pressure vessel system. The largest frequency difference determined was 0.522 Hz.

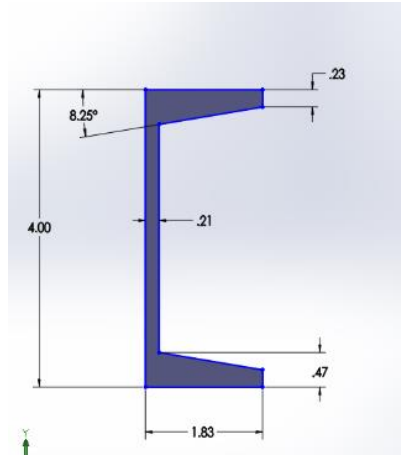


Fig 12. Original cross-sectional geometry with dimensions

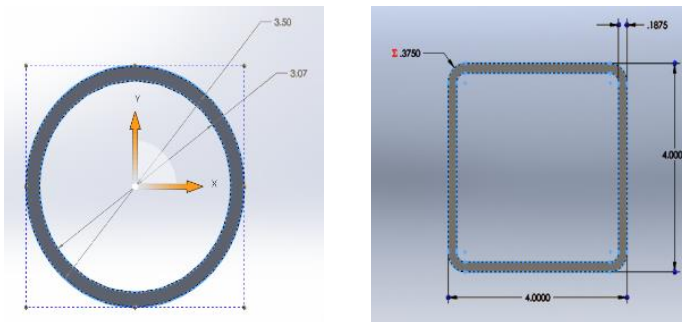


Fig 13. Circular Tubular beam cross-section & Fig Square Tubular beam cross-section

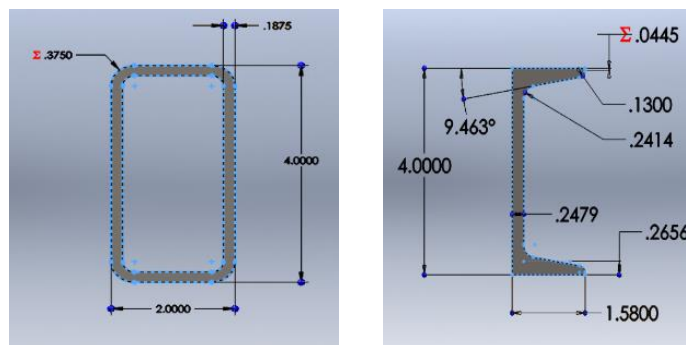


Fig 14. Rectangular Tubular beam cross-section & C-C channel beam cross-section

The Original cross-sectional geometry was found piercing one of the legs. This contributed to the high natural frequency attributed to the model using ANSYS Workbench 18.1. The parallel support bar and leg were being read as a single part due to this intersection, and this increased the stiffness of the model. Using a 4 inches x 2 inches C beam found in SolidWorks Toolbox, the Original cross-sectional geometry was replaced and properly sized. The C beam provided what would normally be industry standard, removing the sharp corners which introduce stress concentrations. The drop of 0.522 Hz during testing was unexpected. This prompted the inquisition as to whether modifying the cross-sectional geometry would produce a significant enough change to warrant replacing the current C beam.

The 3 new designs were circular tubular and two variations of square tubular. The two varieties were a difference in outer dimensions from 4 inches x 4 inches and 4 inches x 2 inches. We chose a variation of 4 inches x 2 inches because the dimensions are similar to the previously evaluated dimensions of the C Beam. The new designs were compared to the C Beam parallel bars with corrected lengths. Cut-Extrudes were made at the end of each beam to accommodate the 30 degrees angled surface of the leg. The beams are then mated to the same positions as the original designs to produce comparable results. The new geometries were then imported into ANSYS Workbench 18.1 for Modal analysis. Modal analysis determines the vibration characteristics of a structure. This analysis was utilized to determine the natural frequencies of each system redesign. The Patch Conforming mesh method was implemented for the cylinder, top, and bottom geometry bodies. The Automatic mesh method was utilized for the three legs of the system. The base of the legs were fixed by applying joints from body to ground. Contact regions were applied to one end of each beam. Fixed joints were employed at the other end of each beam. The Modal analysis simulation was performed to produce natural frequencies for each redesign system.

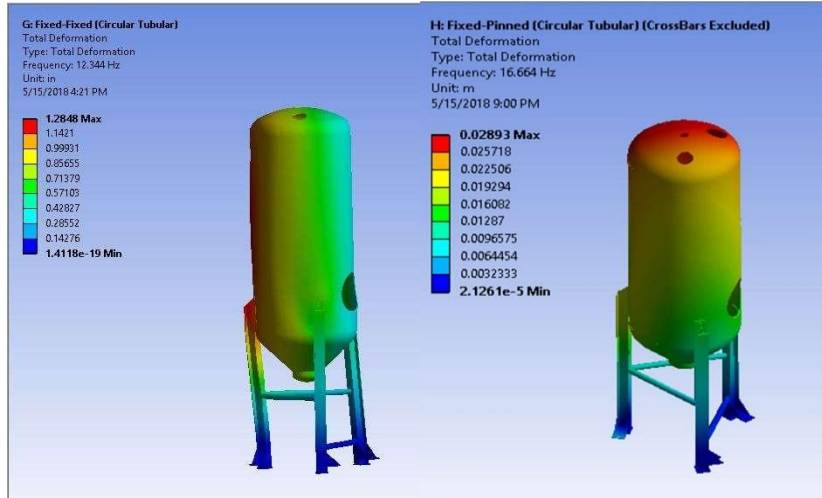


Fig 15. Circular section bars without Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

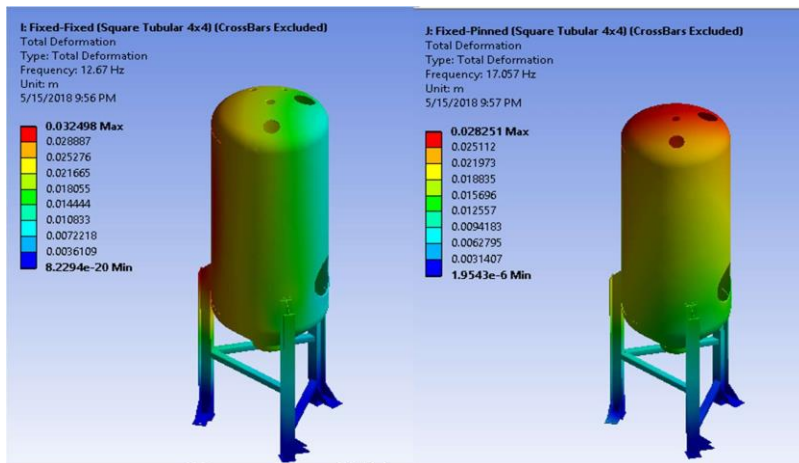


Fig 16. Square 4x4 Bars without Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

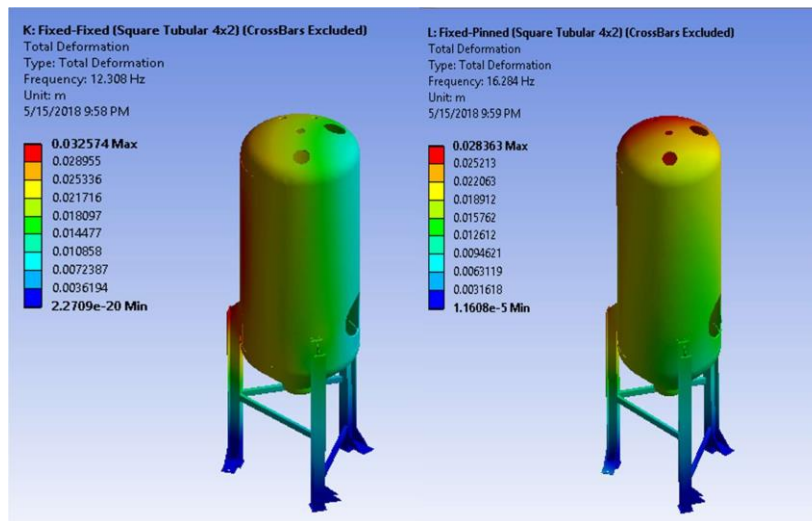


Fig 17. Square 4x2 Bars without Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

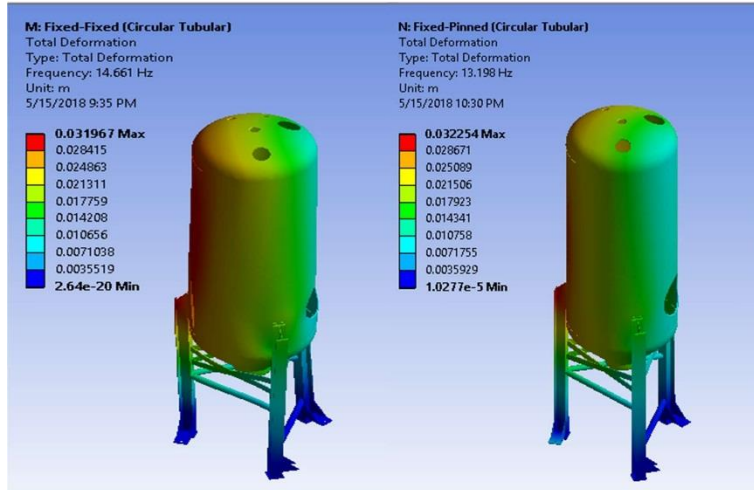


Fig 18. Circular Bars with Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

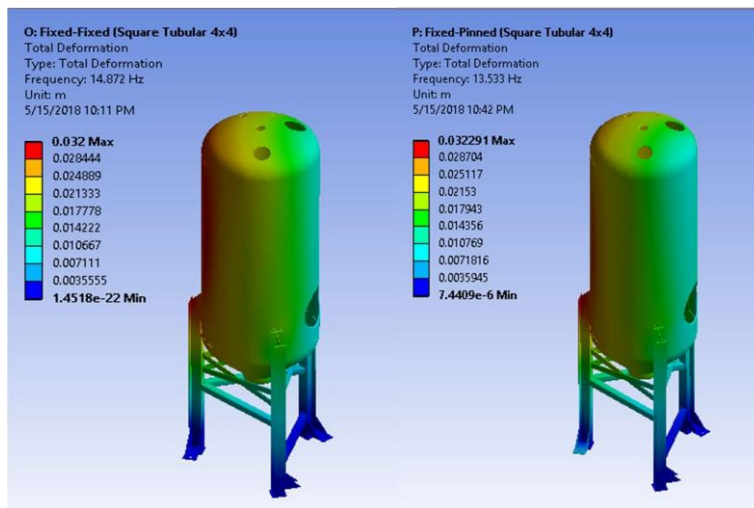


Fig 19. Square 4x4 Bars with Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

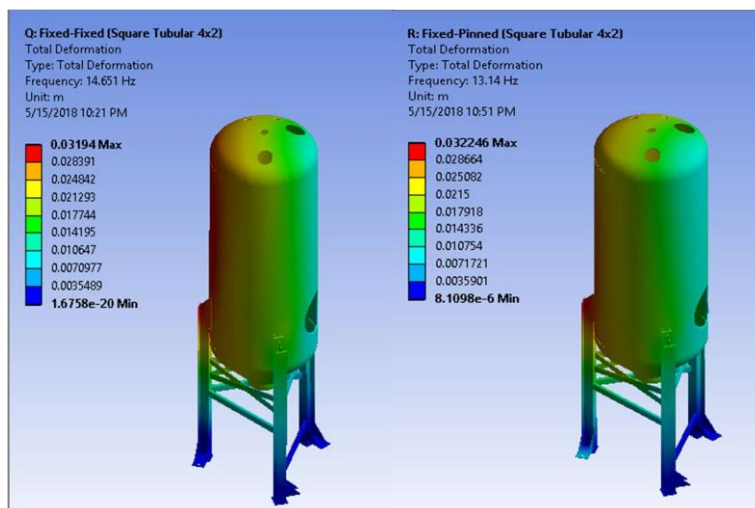


Fig 20. Square 4x2 Bars with Cross Bars (Fixed-Fixed & Fixed-Pinned Support)

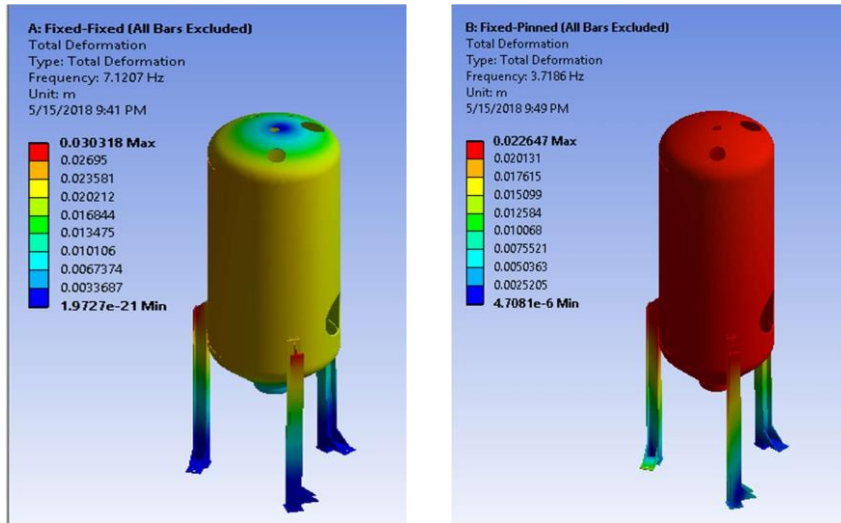


Fig 21. Pressure Vessel without Bars (Fixed-Fixed Support & Fixed-Pinned Support)

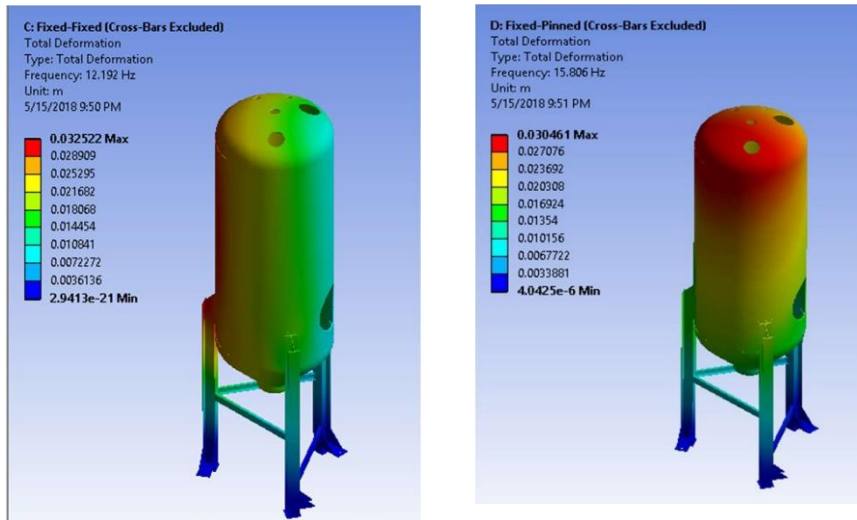


Fig 22. With C-Channel Parallel bars (Fixed-Fixed Support & Fixed-Pinned Support)

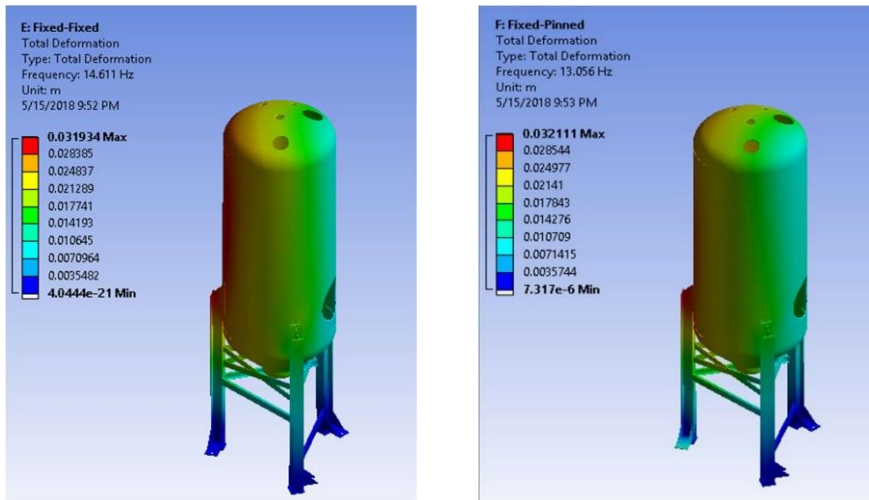


Fig 23. With C-Channel Parallel bars & Crossbars (Fixed-Fixed & Fixed-Pinned Support)

Results

Table 4. Natural Frequencies Compared to Original Design and Vortex Shedding Frequency

Geometry	Lowest Natural Frequency (Hz)	Percent Difference from Original Design (%)	Percent Difference from Vortex Shedding (%)
Unsupported	7.121		-32.57
Original	12.753		20.77
C-Channel	12.231	-4.09	15.82
Circular	12.344	-3.21	16.89
Square	12.670	-0.65	19.98
Rectangular	12.308	-3.49	16.55
Vortex Shedding Frequency (Hz)	10.52		

From the analysis, the most significant difference was a 0.522 Hz decrease resulting from the C-Channel beam design. The C-Channel beam was chosen for the next steps of the analysis as it mimicked most closely what was used on the prototype.

Table 5. Outcomes of the Analysis using ANSYS

Design Types of the Pressure Vessel	Types of the Leg Supports	Calculated Vortex Shedding Frequency with Varying Wind Speeds		Natural Frequency (Hz)
		30 mph	120 mph	
Provided Pressure Vessel without Supports	Fixed-Fixed	1.94	10.56	7.2063
	Fixed-Pinned	1.94	10.56	4.517
Provided Pressure Vessel with Parallel Bars	Fixed-Fixed	1.94	10.56	12.753
	Fixed-Pinned	1.94	10.56	11.647
Provided Pressure Vessel with Parallel & Cross Bars	Fixed-Fixed	1.94	10.56	15.198
	Fixed-Pinned	1.94	10.56	14.473
Redesigned Pressure Vessel without Supports	Fixed-Fixed	1.94	10.56	7.121
	Fixed-Pinned	1.94	10.56	3.719
Redesigned Pressure Vessel with C-Channel Parallel Bars	Fixed-Fixed	1.94	10.56	12.192
	Fixed-Pinned	1.94	10.56	15.806
Redesigned Pressure Vessel with C-Channel Parallel Bars & Cross Bars	Fixed-Fixed	1.94	10.56	14.611
	Fixed-Pinned	1.94	10.56	13.056

Discussion

A new approach was considered to improve the natural frequency once the static analysis of the load cells and modal analysis of the established design were performed. The cross-sectional geometry of the parallel supports was varied, with the new beams analyzed in the same position as the Original cross-sectional design. Each of the redesigns were expected to produce higher values of natural frequencies. That was not the case. The most considerable difference from the Original cross-section design analysis was the C-Channel design of 0.522 Hz. With the C-Channel cross section being established as the actual parallel bar geometry used by JMT, the three designs of circular, square, and rectangular cross-sectional geometry were tested against the C-Channel. While there was an increase of 0.439 Hz from using the square cross-sectional geometry, this difference is not weighty enough to recommend a change to any of the new models. The C-Channel, even with the decrease in natural frequency, exceeds the vortex shedding frequency imposed by 120 mph winds. Performing physical experiments will be required to validate the effects produced in ANSYS Workbench 18.1. Manufacturing an appropriately scaled model will be essential to provide meaningful data that is compatible with the simulation data.

Conclusion

The current design was checked and found to be sufficient for structural integrity. A factor of safety of 1.2 was observed based on the load rating of the load cell under the maximum possible static loading conditions. Vibration analysis was performed with the assumption of an empty vessel. Any liquid contained within the vessel would act as a damper due to the viscous effect. The current design was sufficient if parallel bars were introduced in between the legs. Additional cross bars would provide extra strength and stability to the structure. However, the cross bars may deter the accessibility to the measurement panel and can be eliminated. Also, this recommendation is primarily based on the 120 mph wind speed which is a rarity by itself.

Future Work

Future work will involve experimentation with a scaled down model of the pressure vessel using the existing wind tunnel facility at the Mechanical Engineering Department of Georgia Southern University. JMPT has supplied model of the pressure vessel. Currently, the team is studying the feasibility of using that model to carry out the experimentation. Also, optimizing the position of the beams would be the next step in the analysis with SolidWorks and ANSYS. Testing the beams at various positions along the legs may produce a varying degree of natural frequencies.

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Sanjida Haque is currently a graduate student in the Department of Mechanical Engineering at Georgia Southern University with an expected graduation date of December 7, 2018. Haque likes to work as a problem solver with engineering knowledge and worked in several projects, as a result published four research papers as a first author. Haque is looking forward to establishing a promising career as a dynamic and proficient engineer.

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Dr. Mosfequr Rahman is an Associate Professor with 20 years' experience in academia. He has published over 90 papers; and established the Georgia Southern Wind Energy Lab (GSWEL), and Advanced Materials Lab at Georgia Southern University. He has advised 16 Master's theses and projects in the past 12 years. His research was supported by NSF, NASA and Mechanical Engineering Department of Georgia Southern.

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