Me 421 - Fall 2021

Pressure Vessel

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Professor Simmons

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We have adhered to the Duke Community Standard in completing this assignment. We understand that a violation of the Standard can result in failure of this assignment, failure of this course, and/or suspension from Duke University.

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1 Executive Summary

The purpose of this memo is to provide an update on the design of a liquid pressure vessel to be used in a laboratory setting. The team, tasked with developing the final design of the pressure vessel, was working under several constraints. The vessel must be designed to hold between 50-60 milliliters of liquid, which has been pressurized to 750 psig. Additionally, the design must include an o-ring seal, a threaded connection, and must use a 1/4" NPT connection for filling and emptying the vessel. There are no manufacturing considerations beyond machining as this design is custom and will not be built at a large scale.

The final vessel, constructed from aluminum, costs less than $75 to source the parts and was designed with a factor of safety of 8.75. The vessel has an outer diameter of 3" and is split into two parts: the cap and the body. The cap, which includes the NPT connection, has a thickness of 3/4" while the body has a length of 5.0". Four steel hex head screws are utilized as a threaded connection between the cap and body. The vessel can hold 55 milliliters of liquid and is sealed with a -323 rubber O-ring. The design will take about two hours of machining to be fully constructed, a 50 percent decrease from estimates that used steel. Further design decisions, safety calculations, and cost considerations are described below.

2 Pressure Vessel Design

2.1 Material Selection

The pressure vessel was constructed from 6061 aluminum stock, which was chosen for a variety of reasons. This multipurpose aluminum is widely used in engineering projects, from pipe fittings to aerospace parts. 6061 aluminum is relatively strong, is corrosion resistant, is easily machinable, and is cheap. Although this material is weaker than several steels considered, the ease of machining and low cost were ultimately deemed as more important for implementation in the pressure vessel design. Thicker walls were used in this design in order to overcome the lower strength of aluminum, when compared to steel. This design choice saves the company money during manufacturing as less time is spent on machining, which is quite expensive. The team estimates that using aluminium, rather than a harder steel, cuts machining time by 50 percent, decreasing the machining time from 4 hours to 2 hours.

2.2 Component Selection

The list of the components chosen by the team can be found in Table 1. The team decided to use a 6061 Aluminum stock with a 3" diameter and 6" length. The stock is a 6061 Aluminum shaft. The reasons for choosing this material have been discussed in the previous section.

In order to prevent leaks and to properly seal the body of the pressure vessel and its lid, an oil resistant Buna-N O-Ring was used. The O-ring has an inner diameter of 1.287", an outer diameter of 1.707", and a thickness of 0.21". The team found these dimensions for the O-ring reasonable as this O-ring size provides sufficient sealing to prevent leaks and does not need an extremely large groove to be machined into the body of the pressure vessel, thus saving machining time and reducing any stress concentrations on the face of the pressure vessel body. Furthermore, this O-ring is very cheap and can be replaced easily if it happens to fail.

In order to secure the body and the lid of the pressure vessel together and satisfy the threaded connection requirement, 1.5" Grade 8 Hex Head Screws with a thread size of 1/4"-20 were used. These are steel bolts with a black-oxide finish. The bolts have a high tensile strength of 150,000 psi. On doing stress analysis, it turned out that the tensile failure factor of safety for the bolts was 8.75. This seemed to be a sufficiently high factor of safety for the design. Furthermore, the 1/4" thread diameter doesn't require drilling large holes into the body of the pressure vessel and the lid, thus saving machining time and reducing stress concentrations.

Lastly, 18-8 stainless steel washers were used in the assembly. These washers are meant for a 1/4" screw and are very cheap. The washers help distribute the load from the bolts evenly and thus this component was selected.

2.3 Failure Mode Calculations

2.3.1 Bolt Failure

There are three primary concerns for the failure of the bolt. Tensile yielding, shear failure on the threads of the bolt and shear failure on the threads of the housing. Normally, shear failure of the bolt threads is all that is considered, but because aluminum (the housing material) has a lower yield stress than steel (the bolt material), both sets of threads must be checked.

Considering the tensile strength of the bolts, the amount of force the bolts are holding must first be considered with the following calculation:

$$F = PA_s \tag{1}$$

where F is the resulting force, P is the pressure (750 psi) and A_s is the surface area (1.133 in^2) the pressure is acting upon. This equation shows that there is 849.75 lbs acting on the four bolts, and thus, 212.4375 lbs on each individual bolt. The stress the bolts are under is then found with the following formula:

$$\sigma_y = F/A_p \tag{2}$$

where σ is the resultant tensile stress, F is the force calculated above and A_p is the cross sectional area of each bolt (0.0372 in^2), calculated using a pitch diameter of 0.2175 in. This results in a tensile stress of 5,710.685 psi. Steel has a yield strength of 50,000 psi, thus the Factor of Safety for the tensile failure mode of the bolts is **8.75**.

The next failure mode to examine is the shear failure of the bolt threads. The force the bolts are each individually subject to has already been calculated at 212.4375 lbs, and the surface area of the bolts under shear (or the circumference calculated using the minor diameter times the length of the threaded connection) is 0.442 in^2. The shear stress each bolt is under can then be calculated with the following formula:

$$\sigma_s = F/A_s \tag{3}$$

where σ_s is the shear stress and A_s is the shear area discussed above. Ultimately, this gives a shear stress of 480.63 psi. Steel has a shear strength of about 25,000 psi, thus the Factor of Safety for this failure mode is **52.0**.

Finally, the shear failure of the housing threads must be considered. The force on these threads is the same, as well as the length of the threaded connection. The only change in this case from the bolt case is the material and the shear area, which in this case is calculated with the major diameter (0.25 in) and is 0.589 in^2. Thus the shear stress, calculated with equation (3), is 360.67 psi. Aluminum has a shear strength of about 20,000 which makes the Factor of Safety of this failure mode **55.5**.

2.3.2 Bolt Pre-load

One important specification that must be denoted is bolt pre-load. Pre-loading the bolt ensures that the alternating stress it undergoes remains at a minimum throughout repeated loading and unloading cycles and thus increases the lifetime of the bolt. While the team was unaware whether or not this pressure vessel would undergo repeated use, pre-load will be specified and should be used in the event that multiple pressurization and depressurization cycles are expected. If these pre-load specifications are followed, fatigue is not a worry in the case of the bolt. Ultimately, to effectively alleviate the fatigue effects of pressurization and depressurization, it is required that the initial stress in the bolt is slightly above that of the maximum stress it will experience through the loading cycles. This maximum stress is 5,710 psi. Thus an initial stress of 5,750 will be enough of a pre-load for the bolts. In order to achieve this pre-load, the bolt should be tightened with a torque wrench to 24.2 ft lb_f.

2.3.3 Tangential and Radial Stresses

For thick-walled pressure vessels where the external pressure (P_o) is 0, the following relationships can be used to calculate the radial and tangential stresses:

$$\sigma_t = \frac{r_i^2 p_i}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right) \tag{4}$$

$$\sigma_r = \frac{r_i^2 p_i}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right) \tag{5}$$

Where σ_t is the tangential stress and σ_r is the radial stress; $r_o = 1.5$”, $r_i = 0.5625$”, $P_i = 750 psig$

$$\sigma_t = 1140.63 psi \tag{6}$$

$$\sigma_r = 859.38 psi \tag{7}$$

6061 Aluminum has a yield strength of approximately 40,000 psi. This gives a Factor of Safety of **35.07** for failure from tangential stress and a Factor of Safety of **46.55** from the radial stress for the pressure vessel.

2.4 Other Design Considerations

Some of the design decisions made by the group in regard to the pressure vessel may seem unnecessary, but in reality are grounded in cost and lead time considerations. The main design decision that follows this trend is the seemingly over-engineering of the diameter of the vessel. When considering the threaded connection portion of the design requirements, the group initially considered pipe threading a plug made from the same aluminum stock as the vessel. Ultimately though, this would have increased machining time and costs and potentially called for custom threading which would have made the design far more expensive to machine than necessary.

The team then turned toward bolts as fulfillment to the threaded connection requirement. While examining how to implement the bolts, the team was worried about drilling into the walls of the pressure vessel housing because of the decrease in radius it would add to those locations (and thus the increase in stress) as well as leaving room for the O-Ring groove to be machined. To avoid this, the team came up with the idea to extend the initial diameter of the pressure vessel, which was calculated with a factor of safety to **2.5** in order to machine a flange onto the top of the vessel housing to avoid the previously discussed reduction in effective diameter. Once the flange was considered, the team then decided to purchase larger stock and avoid using the flange, thus decreasing the machining time required to turn down a majority of the stock, and instead drill and tap holes into this larger diameter stock outside the O-Ring groove. This ultimately keeps the radius at a safe size, bringing the vessel to a factor of safety of **8.75**, eliminates the need to order nuts, and decreases the likelihood of shear failure of the bolts by increasing the contact length of the bolt from the size of the nut that would have been used to 0.75", or the length of the threaded hole in the housing. Ultimately, the team saw this design decision as a safer and more effective one, while also remaining under the material budget and decreasing machining cost.

3 Additional Concerns

3.1 Expected Failure

Ultimately, the lowest factor of safety in the entire vessel is the bolt tensile yielding failure mode, with a factor of safety of **8.75**. This is desired because, given the amount of machining necessary and the cost due to this machining, to create the pressure vessel, as well as the price of the other components in the assembly, it is desired to have the bolts fail first. The failure of the bolts would not damage any of the other components in the assembly and would result in only having to replace the bolts, which are cheap, standardized, and require no machining. While the vessel is designed to specifications for the bolts not to fail to begin with, they do have the lowest factor of safety which is most advantageous for cost and lead time purposes.

Another potential failure mode is due to O-Ring plastic deformation. This type of failure is common when using O-Rings at high pressure or for many cycles. This failure mode would present itself as a leak through the face connection of the lid and housing. In addition though, O-Rings wear out through many cycles and given that the O-Rings used in this vessel were a pack of 25, it is not a worry to simply replace the O-Ring. In addition, O-Rings are relatively cheap and require no machining, thus this failure mode has little effect on cost and lead time required to repair the vessel.

4 Cost Analysis

4.0.1 Bill of Materials

Table 1: Bill of Materials

Part	**Description**	**Qty**	**Unit Cost ($)**	**McMaster-Carr Part #**
Stock	6061 Aluminum stock, 6" long with 3" diameter	1	45.68	8974K82
O-ring	Buna-N 323 O-ring with 3/16 fractional width	1	15.48	9452K403
Bolts	2" grade 8 steel hex head screws, 1/4"-20	1	6.12	91268A506
General Washers	18-8 Stainless steel washer for 1/4" screw	1	3.47	92141A029
			($): 74.10	

References

[1] Avallone, E. A., and Baumeister, T. (1996). Mark's Standard Handbook for Mechanical Engineers: Revised by a staff of specialists. Mc Graw Hill.

[2] Budynas, R. G., Nisbett, J. K., and Shigley, J. E. (2015). Shigley's Mechanical Engineering Design. McGraw-Hill Education.

5 Appendix

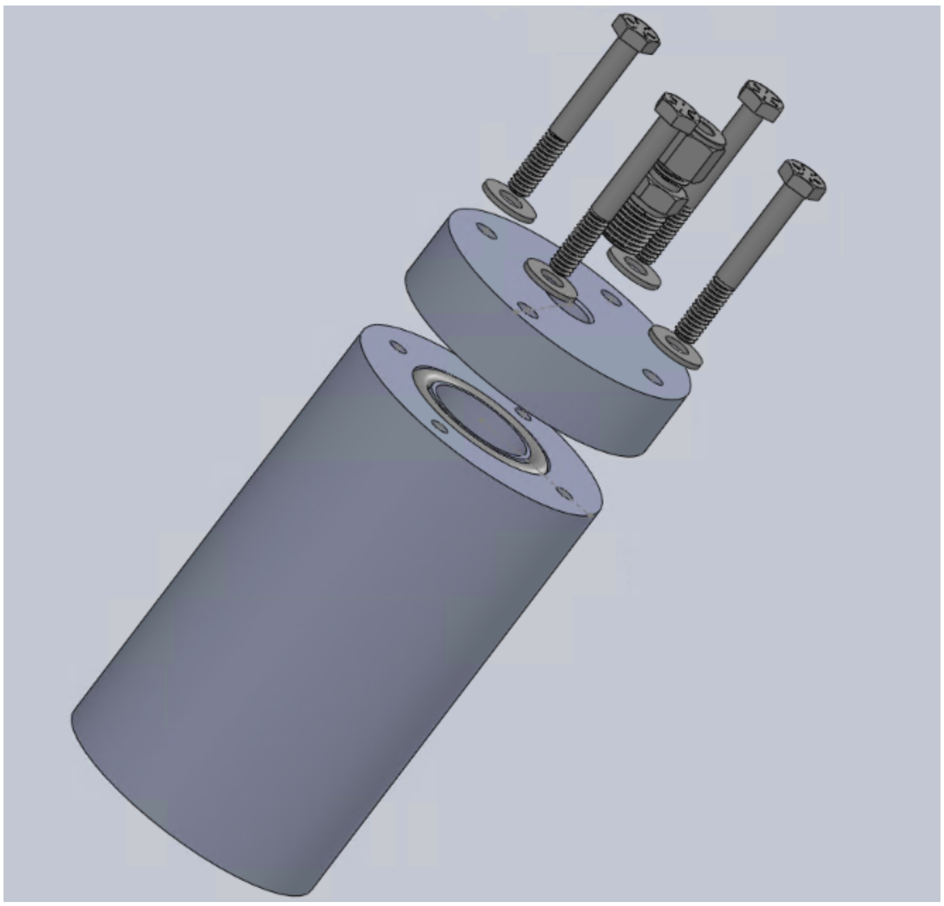

Figure 1: Exploded View of Assembly

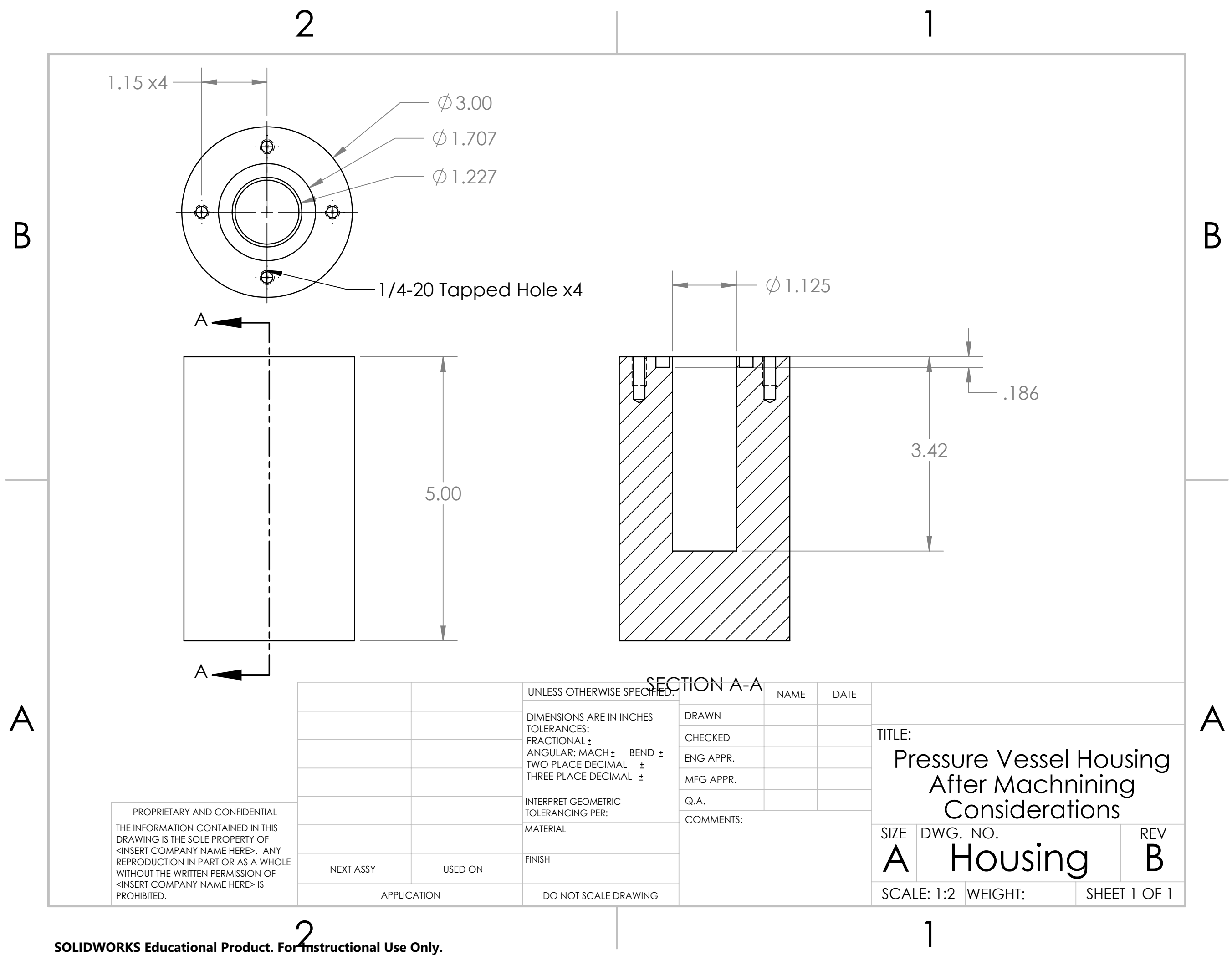

Figure 2: Pressure Vessel Housing Engineering Drawing

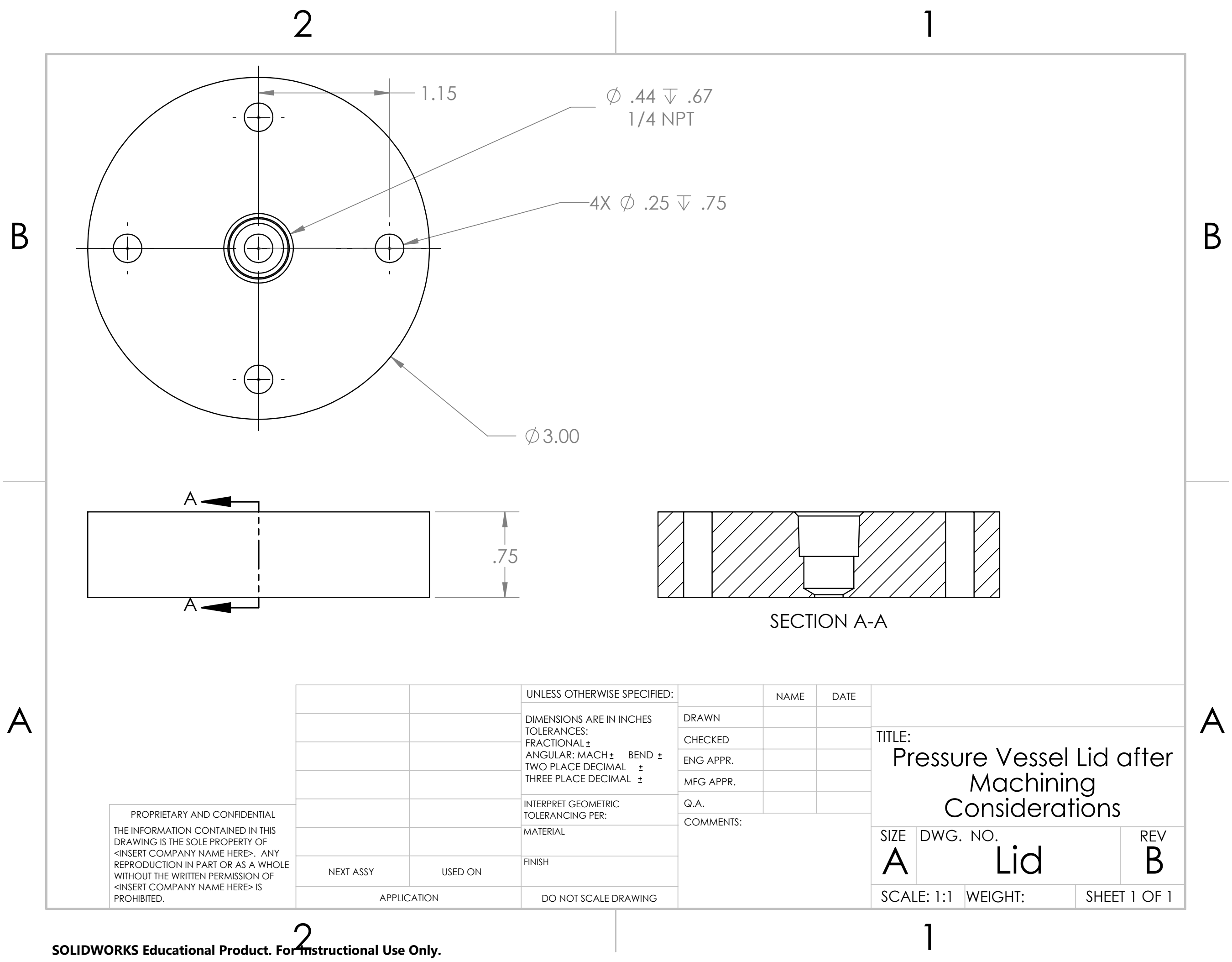

Figure 3: Pressure Vessel Lid Engineering Drawing

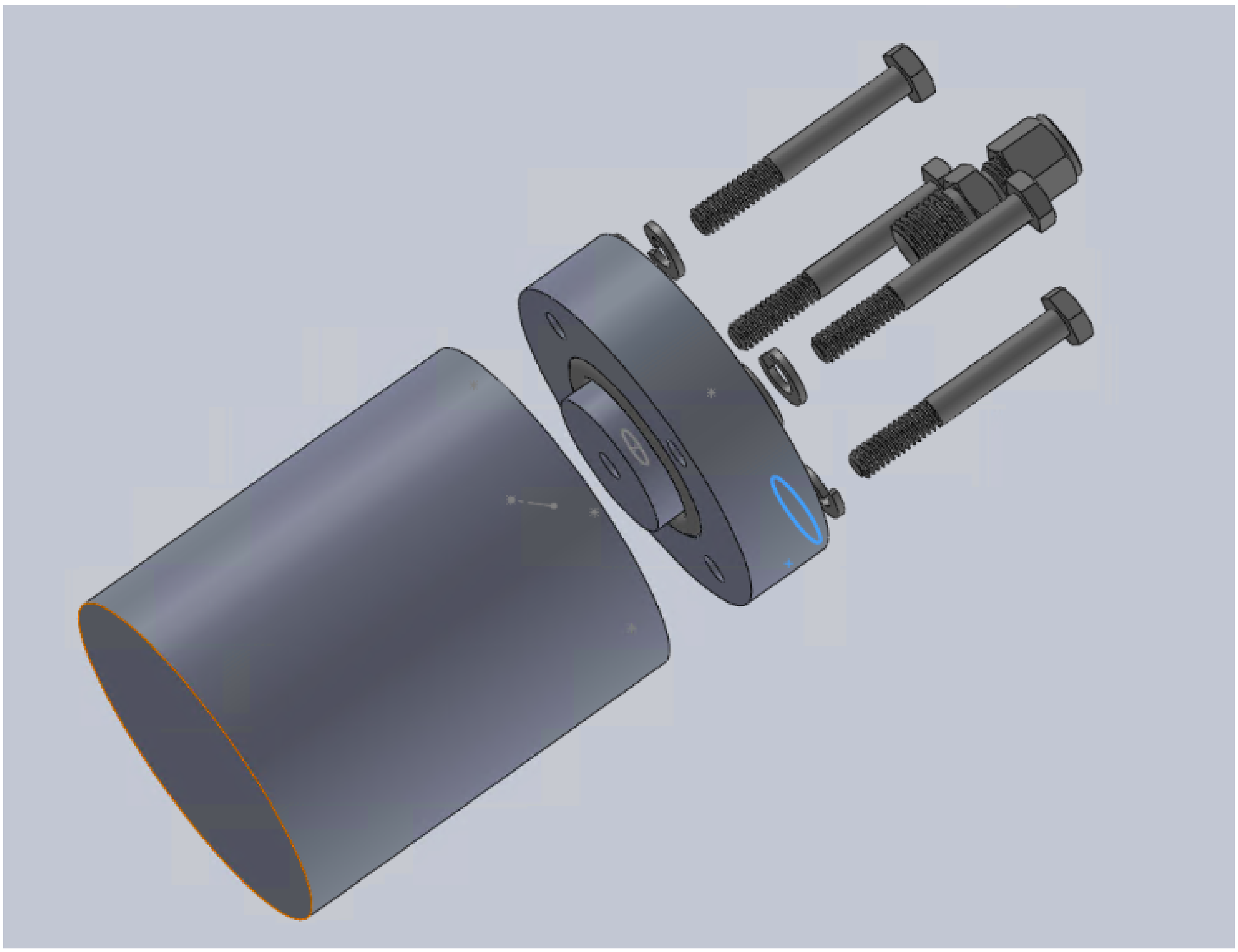

Figure 4: Pressure Vessel Initial Design Exploded View

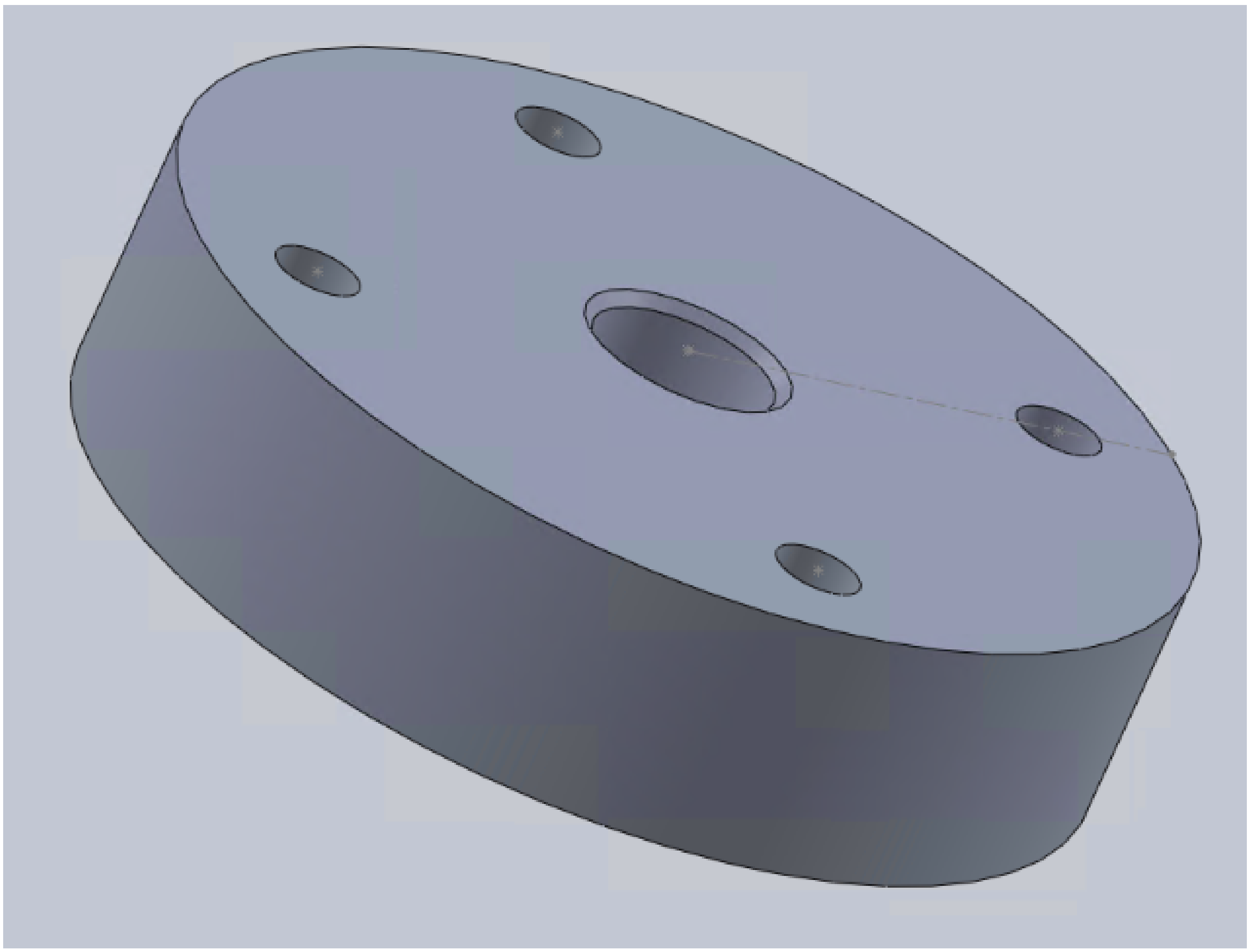

Figure 5: Pressure Vessel Lid Isometric View

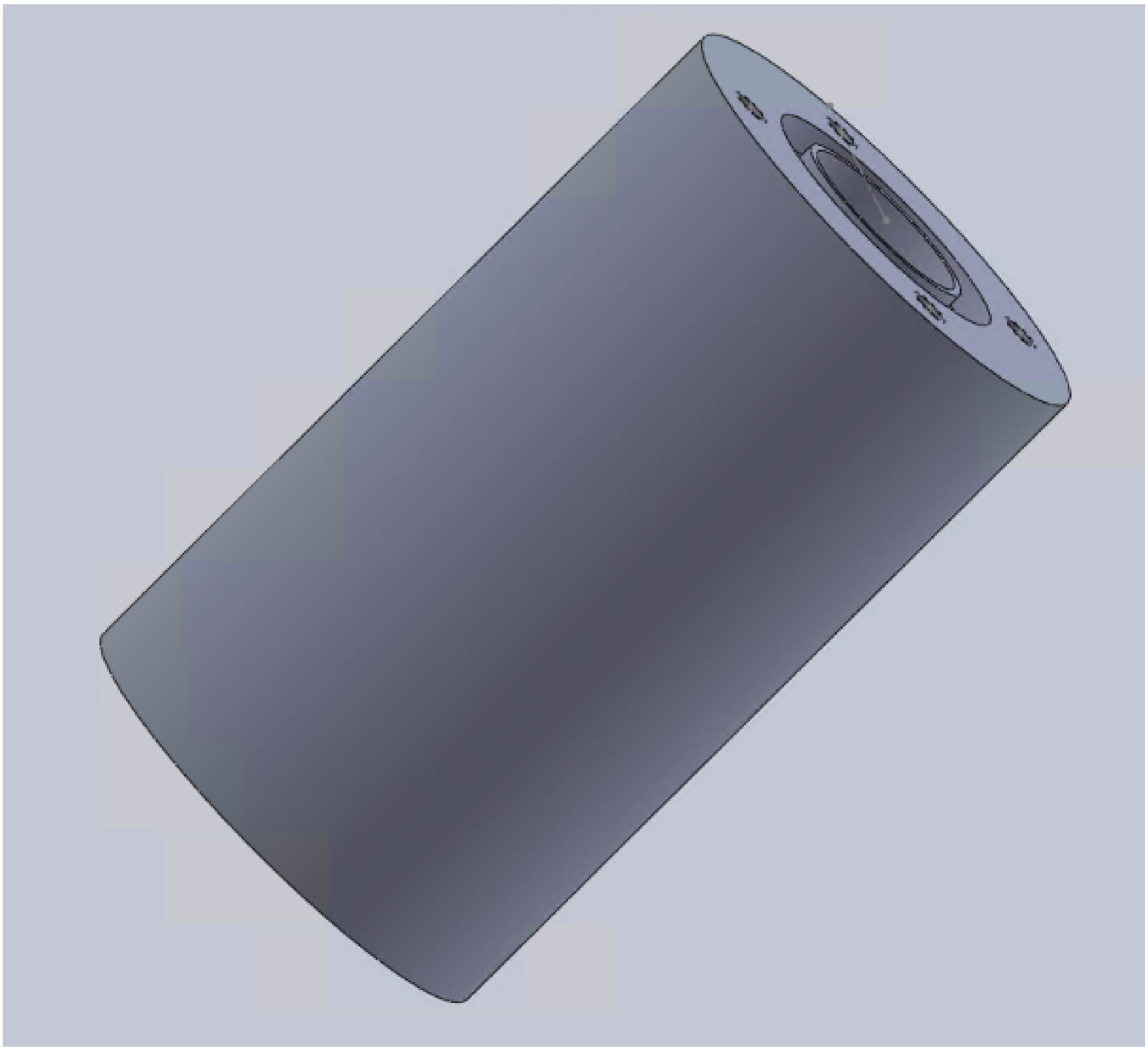

Figure 6: Pressure Vessel Housing Isometric View

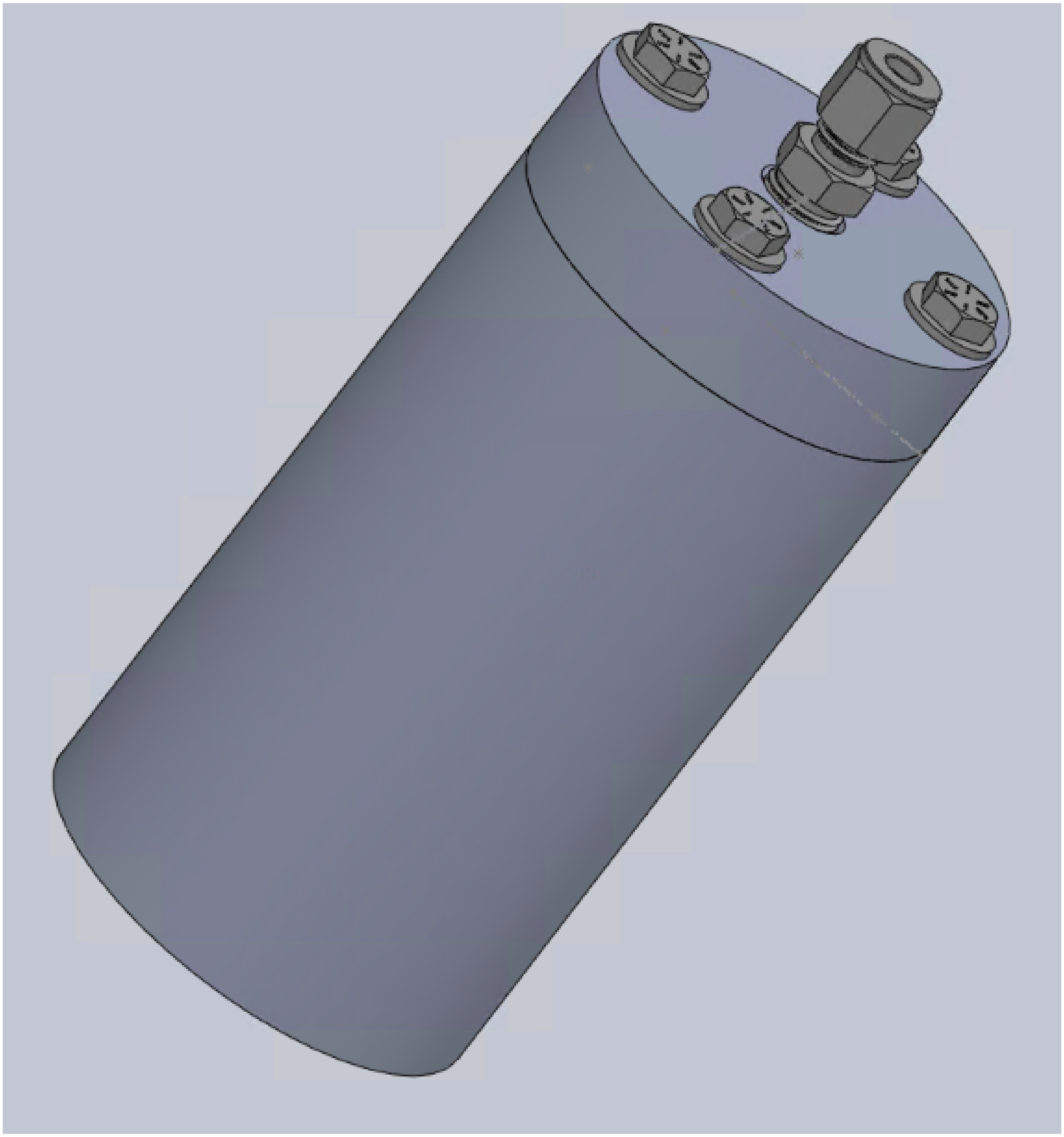

Figure 7: Pressure Vessel Assembled View