MECH 4240 Midterm Design Review: Variable Volume Pressure Vessel

Corp 11 – AMRDEC

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ABSTRACT

The Aviation and Missile Research, Development and Engineering Center (AMRDEC) has a hydrostatic pressure testing system that uses a PI controller to control the amount of water inside the test subject and therefore control the pressure. This PI controller is not operating to the standards required by AMRDEC and needs to be adjusted. To calculate what the ideal values for the constants in the controller are, data for many sizes of pressure vessels is needed. A design for a variable volume pressure vessel was then requested. The design process and the final design for this vessel are outlined in this report.

The final design operates on the method of water displacement. Nine concentric cylinders will displace water inside the vessel, allowing volumes starting at half a gallon and stepping up by half a gallon up to 25 gallons. There will be spaces in between the cylinders to allow the water to flow between them, equalizing the pressure on all sides of the cylinders. The end caps of the pressure vessel will be removable to allow the cylinders to be arranged as needed.

The body of the vessel will be constructed of 7075-T6 aluminum, with an outer radius of 14", an inner radius of 10" and 6.5' long. This will weigh roughly 650 lb. with the end caps.

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INTRODUCTION

The system at the Aviation and Missile Research, Development and Engineering Center (AMRDEC) is designed to test the pressure limits of composite missiles. A concrete room contains the vessels as they are tested to failure. The vessels have a compressor and intensifier pumping water into them with the flow rate controlled by a PI controller. The intensifier system is described below in Fig. 1.

The PI controller is explained in the following block diagram in Fig. 2. The pressure data is read by three pressure transducers and sent back to the controller, which adjusts the valve it controls as needed to pump the appropriate amount of water in the system.

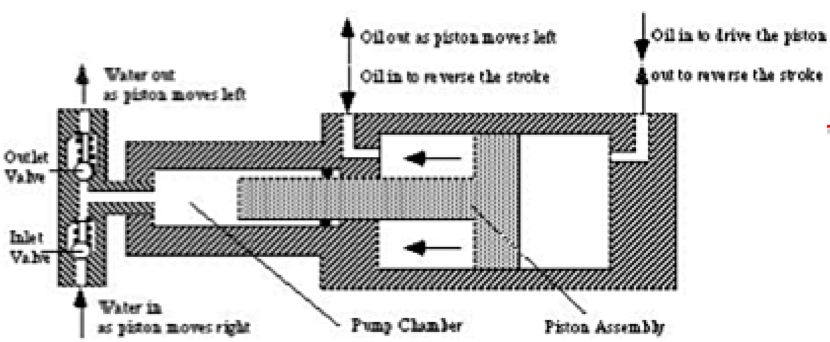


Figure 1: Basic Intensifier Schematic

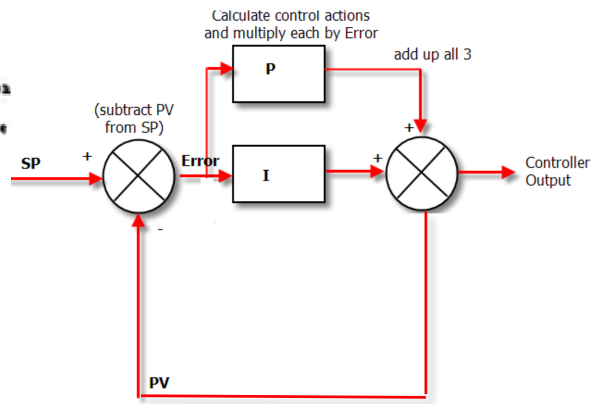


Figure 2: PI Block Diagram

The issue was that the PI controller that controlled the pressure testing at the facilities at AMRDEC did not meet their expectations for ramp rate and pressure holding. To study this problem, they required data on a multitude of pressure vessel sizes and a variety of pressures. To gather this data, a variable volume pressure vessel is to be designed and constructed.

The different designs the engineering team deliberated on will be discussed in a later section. The final chosen design is a cylinder of 7075-T6 aluminum with smaller, non-load bearing, plastic cylinders placed inside it to displace discrete amounts of water which changes the volume. The following schematic details how the design will fit into the system.

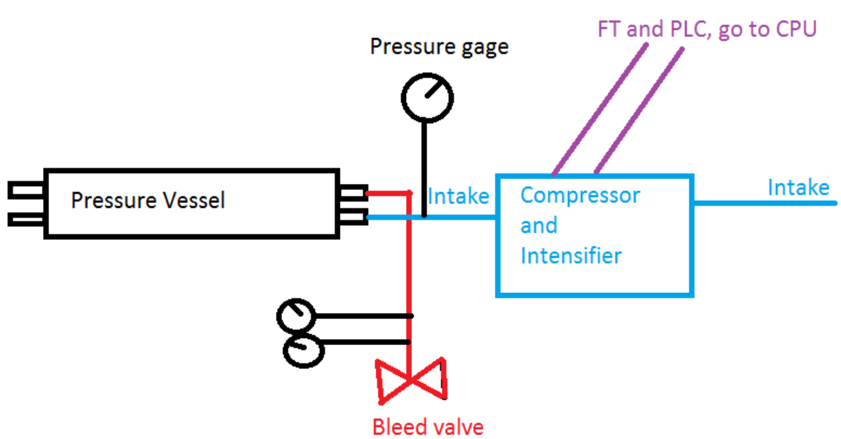


Figure 3: System Schematic

DESIGN DISCUSSION

Mission Objective

The overall objective of this project is to create a variable volume pressure vessel to perform hydrostatic pressure tests on and gather the test data to refine the existing pressure testing controls.

Architectural Design Development

Brainstorming

The first step the engineering team took to create a working design was to look at the desired requirements for the system. The requirements are listed below in Table 1.

Table 1: Design Requirements

Design Requirements
Must accommodate volumes from 0.5 gallons to 25 gallons
Must withstand 15000 psi with a failure factor of safety of 1.5
Must accommodate pressure sensor port
Must attach to the existing pump fittings (1/4” NPT)
Must last through 50+ testing sessions

The initial brainstorming sessions yielded designs that fell into two categories: a piston design or a displacement design. The piston designs changed the volume continuously as the piston moved while the displacement designs changed the volume in discrete amounts by placing objects in the vessel. Some of the preliminary designs are shown below in Fig. 3.

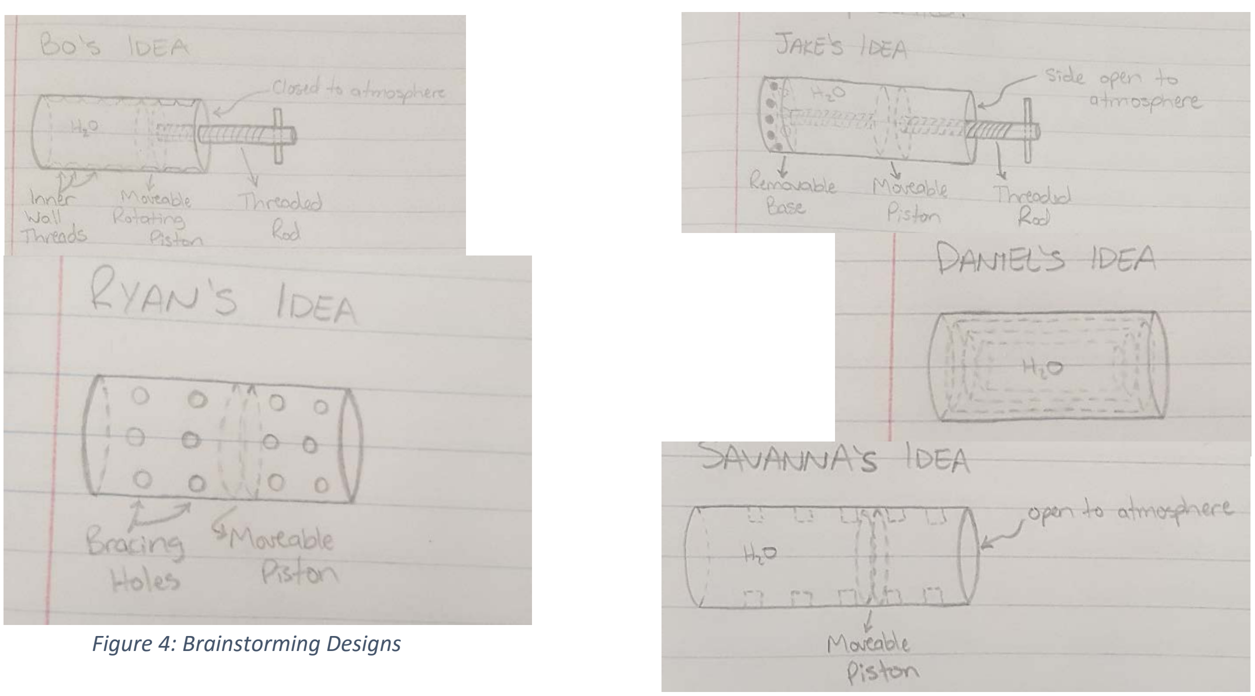


Figure 4: Brainstorming Designs

The piston ideas worked on several different principles. One idea was to have a type of O-ring system around the piston to seal it as it moved and a brake system to keep it in place. Another proposed threading the inside of the cylinder and threading the outside of the piston so that the piston moved like a screw.

The displacement idea was to place several concentric cylinders inside the outer cylindrical pressure vessel to displace water. The cylinders themselves would not attach to the pressure vessel, allowing the water to move throughout the vessel. This would mean that the cylinders are experiencing balanced pressure on all sides and therefore very little stress besides compression.

Design Decision

During further discussion, the piston ideas were narrowed to a threaded design idea, as a braking system seemed unnecessarily complicated and would take more time to design and execute than a threaded system. Basic analysis was done on each design to determine if either could be easily eliminated. The shear force on the threads was analyzed using an estimated radius of 5 inches and the following formula (Eqn. (1)) from *Machinery's Handbooks* and it was determined that a 5-2 Acme thread would not result in any shearing failure, i.e. stripping, of the threads.

$$\text{ShearAreaPerInch} = \pi * \text{MaxMinorBoxDia} * \left[\frac{1}{2} + \left(\frac{1}{P} * \tan(14.5^{\circ})\right) * (\text{MinPitchDiaPin} - \text{MaxMinorBoxDia})\right] (\text{in}) \quad (1)$$

The main issue with the threaded design was how to seal the piston to make the chamber waterproof. An NPT fitting was considered, but would only work for about ten tests before the fitting becomes deformed and stops sealing. O-rings would also be difficult to seal over a threaded opening.

The displacement design was based on the idea that the equivalent pressures on all sides of the objects inside the cylinder would not cause any damage to the objects. Since there would be nine cylinders inside the main cylindrical pressure vessel for a half gallon volume, there was some concern that perhaps the pressure would cause the cylinders to press into one another or be pushed against the side of the pressure vessel and cause more stress that was not accounted for. After consulting with Dr. Richard Williams, this was determined not to be the case. Once the system was pressurized, the cylinders would not push against anything. As long as a gap of a few millimeters allowed water to flow between the cylinders the design would fulfill its requirements. However, depending on their material they might shrink under compression so an additional constraint to that design would be to make the cylinders out of a material that could withstand the pressure without compressing. PVC was considered, as it would be inexpensive and light. A PVC cylinder was tested in SolidWorks under 15,000 psi and it resulted in a deformation of roughly 0.003 inches on the faces of the cylinder. This results in a 0% change in the volume of the vessel.

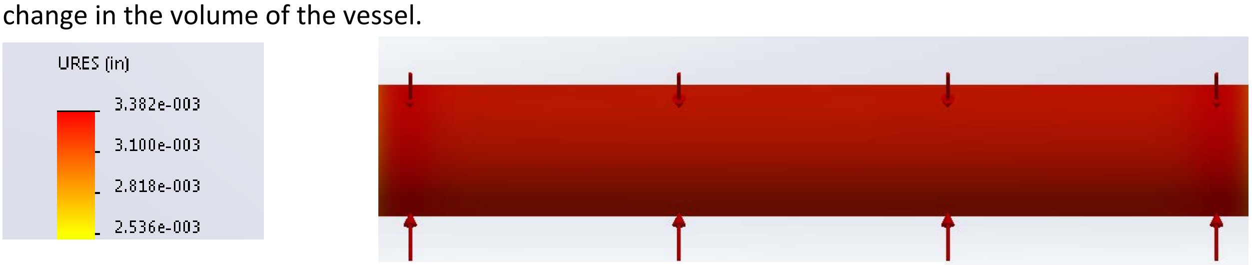


Figure 5: Cylinder Deformation

After considering the viability of these two ideas, a decision matrix was created with point values on a scale of 1-3 assigned to several different categories, which is shown on the next page in Table 2.

Table 2: Decision Matrix

	Ease of Construction	Meets Volume Requirements	Meets Pressure Requirements	Sealing Capabilities	Volume Change Speed	Weight	Cost	**Total**
Threaded	1	3	3	2	3	2	2	**16**
Displacement	3	2	3	3	2	3	3	**19**

The results are added into the final column, which reveals that the displacement design is a more viable design. It is much easier to construct and seal than the threaded design.

With this final design being chosen, basic SolidWorks drawings were created to give a more complete understanding of how the system would function. The following figures demonstrate how the vessel would accomplish its requirement of varying the volume of the pressure vessel.

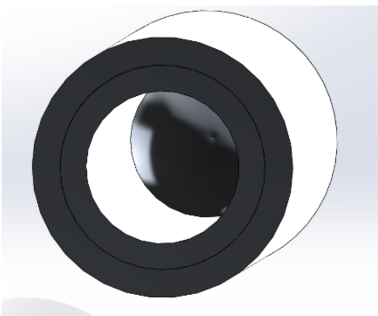


Figure 6: 15 Gallon Volume

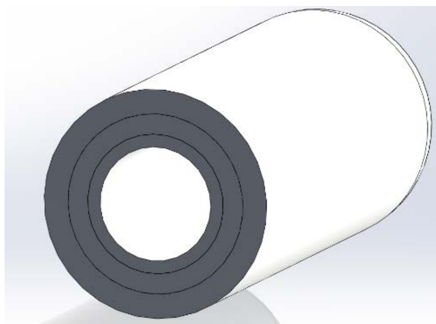


Figure 8: 10 Gallon Volume

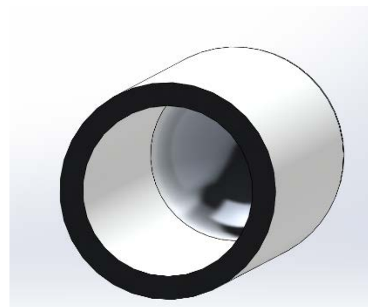


Figure 7: 25 Gallon Volume

The vessel is shown in Fig.3-5 in a mid-section view to show how the cylinders would nest inside the larger pressure vessel. There is a small gap between the inside cylinders and the pressure vessel itself of a few millimeters to allow the water to flow in between the cylinders. The entire assembly is shown below with the end cap attached.

Figure 9: Assembly

Requirements

With the final design chosen, some requirement specific to this design become apparent. As mentioned earlier, the compression of the cylindrical material would need to be negligible. One end of the pressure vessel would also need to be easily removed to be able to change the volume between tests. An outline of all the requirement is shown below.

System Requirements

- Must accommodate volumes from 0.5 gallons to 25 gallons
    - The cylinders to adjust the volume must be incompressible
        - The end of the pressure vessel must be removable to adjust the cylinders
- Must withstand 15000 psi with a failure factor of safety of 1.5
- Must accommodate pressure sensor port
    - The outermost cylinder must not interfere with the pressure sensor
- Must attach to the existing pump fittings (1/4” NPT)
- Must last through 50+ testing sessions
    - The cylinders must also last through testing as well as the pressure vessel

Concept of Operations

To fulfill these requirements, the system will follow these steps. Initially, the pressure vessel end cap will be taken off and the number of cylinders needed will be inserted. The end cap will then be bolted back on. The intake and bleed valve will be attached to the fittings on the end. The pressure vessel will then be filled with water and pressurized to the necessary pressure for that test. Once the test is finished, the system will depressurize using the bleed valve. The end cap can then be removed and the volume can be adjusted for the next test.

System Verification and Validation

To test the volume requirement, the cylinders can be added to a discrete volume. The same volume of water can be added to check that the calculated volumes match the amount of water that fills the system.

The pressure sensor can be verified by simply installing it at the pressure sensor port and running the sensor to check that it is functioning properly. Similarly, the fittings can be checked by testing them with 1/4” NPT and seeing if they attach properly.

The bolts for the end plates can be tested by subjecting them to the previously calculated force that they will be holding.

To initially test the sealing of this model, the vessel can be filled with unpressurized water to check for any leaks. Once this is complete, the design can be tested at the AMRDEC facility to the maximum pressure to ensure the system does not leak. This will also ensure that the cylinders do not compress, since if the volume changed the pressure would decrease inside the vessel and this would be shown in the plot of the pressurization. Testing at the AMRDEC facility will also validate that the system functions as needed.

Interfaces

The mechanical interfaces of the pressure vessel are the ¼” NPT fittings where the water is pumped in and drained out. The only possible electrical component on the vessel is the pressure sensor, which would need to be sealed. However, pressure sensors that have a dial output are being considered to minimize the need for sealing any ports in the vessel.

Mission Environment

The mission environment is the pressure chamber testing room at the AMRDEC facility. It has concrete block walls to protect from testing the missile casings up to failure. This room is also covered in plastic tiles to accommodate the water when a missile casing pressurized with water fails. This environment is ideal for testing a pressure vessel. Because of this, there are not many concerns. The main factor is fitting the vessel in the room. However, this only limits our design to being shorter than about twelve feet. Another possible concern would be the weight of the vessel. The facility has equipment to move the vessel into the room so that is not a limitation.

Risk Management

The obvious risk in constructing a pressure vessel is catastrophic failure. While the mission environment does mean that the vessel is behind concrete block walls when it is being tested, a mode of failure that does not pose a threat should be determined. The ASME Boiler and Pressure Vessel Code requires a leak-before-burst method so that the vessel does not catastrophically fail. Pop-off valves are being considered to release the vessel pressure safely if it gets above a set pressure.

Configuration Management and Documentation

The team has a Dropbox folder that contains the code and calculations done so far, sorted by either threaded or displacement design.

SUBSYSTEMS DESIGN ENGINEERING

Displacement Mechanism

The displacement mechanism in this design are concentric cylinders. To calculate the size of the cylinders needed, the following formula was used. The variables are defined below the formula. This calculates the volume displaced by a certain cylinder. Using this formula, several sizes of cylinders can be obtained to create discrete volumes stepping by a half gallon from 0.5 gallons to 25 gallons.

$$v = \pi * l * (r_o^2 - r_i^2) \tag{2}$$
$$t = r_o - r_i$$

v = volume displaced
l = length of tube
r_o = outer radius
r_i = inner radius
t = thickness

The sizes of the cylinders was also calculated in Matlab and the results are shown to the right in Figure 10.

```
Selecting the 5" radius pipe gives a length of 6.1275'
and thickness of 1.357"
         RingNo      RO      Thickness     Volume
           1          5        1.127         10
           2        3.873      0.7107         5
           3        3.1623     0.9262         5
           4        2.2361     0.23606        1
           5          2        0.26794        1
           6        1.7321     0.31783        1
           7        1.4143     0.41419        1
           8        1.0001     0.29286       0.5
           9        0.7072     0.69531       0.5
```

Figure 10: Cylinder Sizes

Sealing System

The end cap must be removable to provide access to the cylinders. The initial proposal was to pin the end cap on with several metal pins parallel to the back face going through the sides into the back face. This concept is explained in Figure 6.

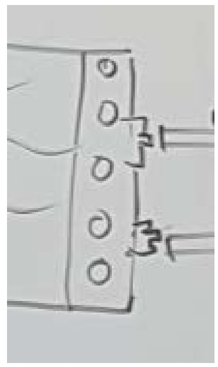


Figure 11: Initial End Cap Design

The pins would go into the holes shown above. The fittings for the intake and bleed valve would go through the back plate as shown. When calculations of the shear force on each pin were done, the number of pins needed was calculated to be eight if the pins were found to need to be three inches in diameter to hold the plate on, which would then compromise the integrity of the end with the holes for the pins. It was clear from this that a fixture not in shear would be a better option. A bolted face plate was then proposed with welded flanges holding the bolts in place. The force on each bolt was calculated for sixteen bolts to be 1,178,100 lb. and 1,767,100 lb. with a factor of safety against yield of 1.5. Several bolts satisfy this requirement, one of which is a 1 3/8" A139 Grade B7. A quote is being procured for a gasket to seal the end cap.

Vessel Material and Size

The size of the vessel to fulfill the requirements was determined. Code was written that can be found in Appendix A to determine the thickness needed for the vessel. This was calculated for different radii and the resulting length and thickness was determined. The initial material researched was A36 steel, which would satisfy the requirements and has a fracture toughness of 45.5 ksi/in. However, the vessel would be over 2300 lb. To minimize weight, 7075-T651 aluminum was then used in the code. This also satisfied the requirements with a weight of 570 lb but had a minimum fracture toughness of 18 ksi/in, which only gives a factor of safety against crack propagating of 1.2. It is lower than the value for steel but still strong enough to not crack under the maximum pressure.

In researching material costs, it was found that no pipe large enough for our design is prefabricated. A solid rod would have to be purchased and then machined, which incurs an addition cost in the budget. A less expensive aluminum, 6061, was researched but its yield strength of 18 ksi meant that it would not satisfy the requirement of holding 15 ksi with a 1.5 factor of safety against yield. The current design is therefore still 7075-T651 aluminum with the higher costs. This will be outlined in the budget in the project management section (pg. 12). On the following page are the results from the three materials considered. A summary of each material, the thickness needed, and its cost per cubic foot is also provided on the following page of the results.

10000 psi					15000 psi				
Inner Radius (in)	Length (ft)	Thickness (in)	Diameter (in)	Weight (lbs)	Inner Radius (in)	Length (ft)	Thickness (in)	Diameter (in)	Weight (lbs)
4	9.574165	0.927054	9.854108	304.681	4	9.574165	1.500675	11.00135	524.896
4.5	7.564772	1.042936	11.08587	304.681	4.5	7.564772	1.688259	12.37652	524.896
5	6.127465	1.158818	12.31764	304.681	5	6.127465	1.875844	13.75169	524.896
5.5	5.064021	1.274699	13.5494	304.681	5.5	5.064021	2.063428	15.12686	524.896
6	4.255184	1.390581	14.78116	304.681	6	4.255184	2.251013	16.50203	524.896
6.5	3.625719	1.506463	16.01293	304.681	6.5	3.625719	2.438597	17.87719	524.896
7	3.126258	1.622345	17.24469	304.681	7	3.126258	2.626181	19.25236	524.896
7.5	2.723318	1.738226	18.47645	304.681	7.5	2.723318	2.813766	20.62753	524.896
8	2.393541	1.854108	19.70822	304.681	8	2.393541	3.00135	22.0027	524.896

Table 3: 7075-T651 Results

Table 4: A36 Steel Results

10000 psi					15000 psi				
Inner Radius (in)	Length (ft)	Thickness (in)	Diameter (in)	Weight (lbs)	Inner Radius (in)	Length (ft)	Thickness (in)	Diameter (in)	Weight (lbs)
4	9.574165	2.23355	12.4671	2343	4	9.574165	4.326664	16.65333	5467
4.5	7.564772	2.512744	14.02549	2343	4.5	7.564772	4.867497	18.73499	5467
5	6.127465	2.791937	15.58387	2343	5	6.127465	5.40833	20.81666	5467
5.5	5.064021	3.071131	17.14226	2343	5.5	5.064021	5.949163	22.89833	5467
6	4.255184	3.350325	18.70065	2343	6	4.255184	6.489996	24.97999	5467
6.5	3.625719	3.629518	20.25904	2343	6.5	3.625719	7.030829	27.06166	5467
7	3.126258	3.908712	21.81742	2343	7	3.126258	7.571662	29.14332	5467
7.5	2.723318	4.187906	23.37581	2343	7.5	2.723318	8.112495	31.22499	5467
8	2.393541	4.4671	24.9342	2343	8	2.393541	8.653328	33.30666	5467

Table 5: 6061 Aluminum Results

10000 psi				
Inner Radius (in)	Length (ft)	Thickness (in)	Diameter (in)	Weight (lbs)
4	9.574165	9.266499	26.533	5659.5
4.5	7.564772	10.42481	29.84962	5659.5
5	6.127465	11.58312	33.16625	5659.5
5.5	5.064021	12.74144	36.48287	5659.5
6	4.255184	13.89975	39.7995	5659.5
6.5	3.625719	15.05806	43.11612	5659.5
7	3.126258	16.21637	46.43275	5659.5
7.5	2.723318	17.37469	49.74937	5659.5
8	2.393541	18.533	53.066	5659.5

Table 6: Summary of Materials Study

Material	Yield Strength (psi)	Weight for 10 ksi. Max Pressure (lb.)	Weight for 15 ksi. Max Pressure (lb.)	Wall Thickness for 5 in. Inner Diameter (in.)	Cost/cubic foot ($/ft^3)
A36 Steel	36000	2343	5467	2.7919	1837.86
6061 Aluminum	18000	5660	n/a	11.583	1072.68
7075-T6 Aluminum	69000	327.25	504.31	2.014	2031.55

PROJECT MANAGEMENT

The project leader is Savanna Earnest and the scribe is Jake Hamlett. During design development, the team was divided up into two teams to work on the two different designs. An outline of the team's work so far and future planning is provided on the following page in Table 3.

Table 7: Project Milestone Outline

Past			
May 30	**May 31**	**June 6**	**June 10**
Met to plan questions for AMRDEC on conference call	Conference call with AMRDEC	Discussion of materials and basic concepts	Visit to AMRDEC facilities
June 13	**June 15**	**June 16**	**June 19**
System design meeting	Preliminary brainstorming	System design meeting	Final brainstorming meeting
June 21	**June 22**	**June 23**	
Detail design of vessel size	Final design chosen	Contacting vendors for pricing SolidWorks outline created	
Future			
July 6	**July 8**	**July 11**	**July 20**
Finalize detail design	Early Prototyping	Parts ordered	Testing
July 21	**July 25**		
Adjust design based on test results if needed	Finish detail design		

For the project's budget, the largest expense is the cylindrical vessel. A preliminary estimate of the budget is shown below in Table 4. Quotes can be found in Appendix B.

Table 8: Preliminary Budget

Material	Cost
7075-T6 Al Tube, 14” OD, 10”ID, 6.5’ long	8288
Pressure Transducer	390
7075-T6 End Cap Material	455
Total	**9,133**

CONCLUSIONS

In summary, two different general designs were considered: a threaded piston and vessel and a displacement of water inside the vessel. The threaded design presented an interesting engineering challenge but the displacement design was simpler and therefore had fewer locations for issues in the design. The displacement design was chosen and deciding the details of the design is in progress. The next step will be to finalize all the details of the design and begin testing prototypes and the final model to ensure it meets requirements.

APPENDIX A

Code to determine vessel size

```
clc
clear all
%knowns
po = 0; %psi
p = [5000,10000,15000]; %psi
gal = 25;
v = 231*gal; %in3
ri = 4:.5:8; %in
a = pi*ri.^2; %in2
l = v./a; %in
s_steel = 73000; %psi
sf = 1.5;
p_steel = .098; %lb/in3 %.284 steel .098 al
%thin walled: stress = p*r/t
%t_thin = p.*ri/s_steel; %in
%thick walled
syms t
for o = 1:length(p)
pin = p(o);
for n = 1:length(ri)
eqn = (pin*(ri(n))^2-po*(t+ri(n))^2)/((ri(n)+t)^2-ri(n)^2)...
+(pin-po)*(((ri(n)+t)^2)*ri(n)^2)/(((ri(n)+t)^2-ri(n)^2)*(ri(n))^2) ==
s_steel/sf;
thickness(n,:) = solve(eqn,t);
end
treal = double(thickness(thickness>0));
matrix(:,1) = ri';
matrix(:,2) = l'/12;
matrix(:,3) = treal;
matrix(:,4) = (matrix(:,1)+matrix(:,3))*2;
matrix2(:,1) = (pi.*l.*((treal'+ri).^2-ri.^2))'; %volume steel thick
%matrix2(:,2) = (pi.*l.*((t_thin+ri).^2-ri.^2))'; %volume steel thin
matrix(:,5) = matrix2(:,1)*p_steel; %weight thick
%matrix2(:,3) = matrix2(:,2)*p_steel; %weight thin
fprintf('For a pressure of %5.0f psi and a volume of %2.0f
gallons\n',pin,gal)
disp(' RI(in) Length(ft) t(thick) OD(in) W(lbs)');
disp(matrix);
subplot(length(p),length(p_steel),o)
plot(ri,l/12,'b',ri,treal,'g');
legend('Length,ft','T(Thick),in');
title(pin)
xlabel('Inside Radius, in');
end
% for m = 1:length(matrix)
% fprintf(' %4.1f %8.3f %11.3f %12.3f %8.3f %8.3f\n',...
% matrix(m,1),matrix(m,2),matrix(m,3),matrix(m,4));
% end
%
% plot(ri,l/12,'b',ri,treal,'g',ri,t_thin,'r');
% legend('Length,ft','T(Thick),in','T(Thin),in');
% xlabel('Inside Radius, in');
```

```
% % figure
% % plot(ri,matrix2(:,3),'k',ri,matrix2(:,4),'y')
% disp('');
disp('Selecting the 5" radius pipe gives a length of 6.1275''')
disp('and thickness of 1.876"');
% for pipe with following
% radius length thick
% 5.0000 6.1275 1.876
% 1 gal = 231 in^3
r1 = 5; %in
t0 = 1.876; %in
l = 6.1275*12; %in
vtot = (pi*l*r1^2)/231; %gal
syms t1
eq1 = (pi*l*(r1^2-(r1-t1)^2))/231 == 10;
t1 = double(solve(eq1,t1));
r2 = r1-t1(1);
syms t2
eq2 = (pi*l*(r2^2-(r2-t2)^2))/231 == 5;
t2 = double(solve(eq2,t2));
r3 = r2-t2(1);
syms t3
eq3 = (pi*l*(r3^2-(r3-t3)^2))/231 == 5;
t3 = double(solve(eq3,t3));
r4 = r3-t3(1);
syms t4
eq4 = (pi*l*(r4^2-(r4-t4)^2))/231 == 1;
t4 = double(solve(eq4,t4));
r5 = r4-t4(1);
syms t5
eq5 = (pi*l*(r5^2-(r5-t5)^2))/231 == 1;
t5 = double(solve(eq5,t5));
r6 = r5-t5(1);
syms t6
eq6 = (pi*l*(r6^2-(r6-t6)^2))/231 == 1;
t6 = double(solve(eq6,t6));
r7 = r6-t6(1);
syms t7
eq7 = (pi*l*(r7^2-(r7-t7)^2))/231 == 1;
t7 = double(solve(eq7,t7));
r8 = r7-t7(1);
syms t8
eq8 = (pi*l*(r8^2-(r8-t8)^2))/231 == .5;
t8 = double(solve(eq8,t8));
r9 = r8-t8(1);
syms t9
eq9 = (pi*l*(r9^2-(r9-t9)^2))/231 == .5;
t9 = double(solve(eq9,t9));
r10 = r9-t9(1);
matrixnew(1,:) = [r1+t0,r1,r2,r3,r4,r5,r6,r7,r8,r9,r10];
matrixnew(2,:) =
[t0,t1(1),t2(1),t3(1),t4(1),t5(1),t6(1),t7(1),t8(1),t9(1),r10];
disp(' RO Thickness');
disp(matrixnew');
extra_room = r1-t1(1)-t2(1)-t3(1)-t4(1)-t5(1)-t6(1)-t7(1)-t8(1)-t9(1);
figure
phi = 0:.01:2*pi;
```

```
plot(r1*cos(phi),r1*sin(phi),'r',r2*cos(phi),r2*sin(phi),'k',...
r3*cos(phi),r3*sin(phi),'r',r4*cos(phi),r4*sin(phi),'k',...
r5*cos(phi),r5*sin(phi),'r',r6*cos(phi),r6*sin(phi),'k',...
r7*cos(phi),r7*sin(phi),'r',r8*cos(phi),r8*sin(phi),'k',...
r9*cos(phi),r9*sin(phi),'r');
```

Code to determine cylinder size

```
clc
clear all
format short g

disp('Selecting the 5" radius pipe gives a length of 6.1275''')
disp('and thickness of 1.357"');

% for pipe with following
% radius    length    thick
% 5.0000    6.1275    1.3739
% 1 gal = 231 in^3

r1 = 5; %in
t0 = 1.3739; %in
l = 6.1275*12; %in

vtot = (pi*l*r1^2)/231; %gal

syms t1
eq1 = (pi*l*(r1^2-(r1-t1)^2))/231 == 10;
t1 = double(solve(eq1,t1));
r2 = r1-t1(2);

syms t2
eq2 = (pi*l*(r2^2-(r2-t2)^2))/231 == 5;
t2 = double(solve(eq2,t2));
r3 = r2-t2(2);

syms t3
eq3 = (pi*l*(r3^2-(r3-t3)^2))/231 == 5;
t3 = double(solve(eq3,t3));
r4 = r3-t3(2);

syms t4
eq4 = (pi*l*(r4^2-(r4-t4)^2))/231 == 1;
t4 = double(solve(eq4,t4));
r5 = r4-t4(2);

syms t5
eq5 = (pi*l*(r5^2-(r5-t5)^2))/231 == 1;
t5 = double(solve(eq5,t5));
r6 = r5-t5(2);

syms t6
```

```
eq6 = (pi*l*(r6^2-(r6-t6)^2))/231 == 1;
t6 = double(solve(eq6,t6));
r7 = r6-t6(2);

syms t7
eq7 = (pi*l*(r7^2-(r7-t7)^2))/231 == 1;
t7 = double(solve(eq7,t7));
r8 = r7-t7(2);

syms t8
eq8 = (pi*l*(r8^2-(r8-t8)^2))/231 == .5;
t8 = double(solve(eq8,t8));
r9 = r8-t8(2);

syms t9
eq9 = (pi*l*(r9^2-(r9-t9)^2))/231 == .5;
t9 = double(solve(eq9,t9));
r10 = r9-t9(2);

matrixnew(1,:) = [1,2,3,4,5,6,7,8,9];
matrixnew(2,:) = [r1,r2,r3,r4,r5,r6,r7,r8,r9];
matrixnew(3,:) = [t1(2),t2(2),t3(2),t4(2),t5(2),t6(2),t7(2),t8(2),t9(2)];
matrixnew(4,:) = [l*pi*(r1^2-r2^2)/231,l*pi*(r2^2-r3^2)/231,l*pi*(r3^2-
r4^2)/231,l*pi*(r4^2-r5^2)/231,l*pi*(r5^2-r6^2)/231,l*pi*(r6^2-
r7^2)/231,l*pi*(r7^2-r8^2)/231,l*pi*(r8^2-r9^2)/231,l*pi*(r9^2-r10^2)/231];
disp('         RingNo      RO       Thickness       Volume');
disp(matrixnew');
```

APPENDIX B

MATERIAL QUOTATION

FUTURE ALLOYS, INC.

20151 Bahama Street, Chatsworth, CA 91311 PH: 818-701-1144 FAX: 818-701-6182

SOLD TO:
ARMDEC
575 Shelton Road, #4503
Auburn, AL 36830
(256)665-6024

SHIP TO:

DATE: June 29, 2016
BUYER: Chris Boehme
SELLER: Joe Jansen
P.O. #:

F.O.B. Auburn, AL SHIP VIA: Trucking co. TERMS: Check in advance. RESALE/TAXABLE:

QTY.	ALLOY/TEMPER	SIZE	FORM	UNIT PRICE	APPROX. WT.
1 pc.	7075 T6	14.000" OD x 10.000" ID x 78.000"		$8,288.00 each	
		ID tolerance = + 0.000", - 0.250"			

Price is delivered to you in Auburn, AL.

Lead time: delivered 2 weeks after receipt of order & payment.

All material is subject to prior sale.

COMMENTS:
Price above is discounted for check or bank wire payment in advance. Add 3% for payment by Visa or Mastercard.

SHIPPING INSTRUCTIONS:

If you would like to order this exactly as quoted, please sign, date and fax or email back. ______________________
Prices are valid for 10 days. SUBJECT TO PRIOR SALE. *Thank you for the opportunity to quote on your material requirements.*

THANK YOU

THIN-FILM TRANSDUCERS FOR GAGE, VACUUM, OR COMPOUND RANGES

RUGGED STAINLESS STEEL CONSTRUCTION

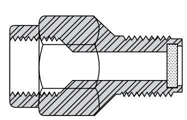
Vacuum to 20,000 psi (1380 bar)

PX603/PX613 Series

1 YEAR WARRANTY

- Excellent Long-Term Stability
- NEMA 4 (IP65) Cable or Connector Models
- 0.4% Accuracy
- Thin-Film Design for High Reliability
- Ideal for Liquid or Gas
- Zero and Span Adjustments

Metric thread adaptors available, see omega.com



¼ NPT Pressure Snubbers:
PS-4G = Gas
PS-4E = Lt Oil
PS-4D = Dense Lq

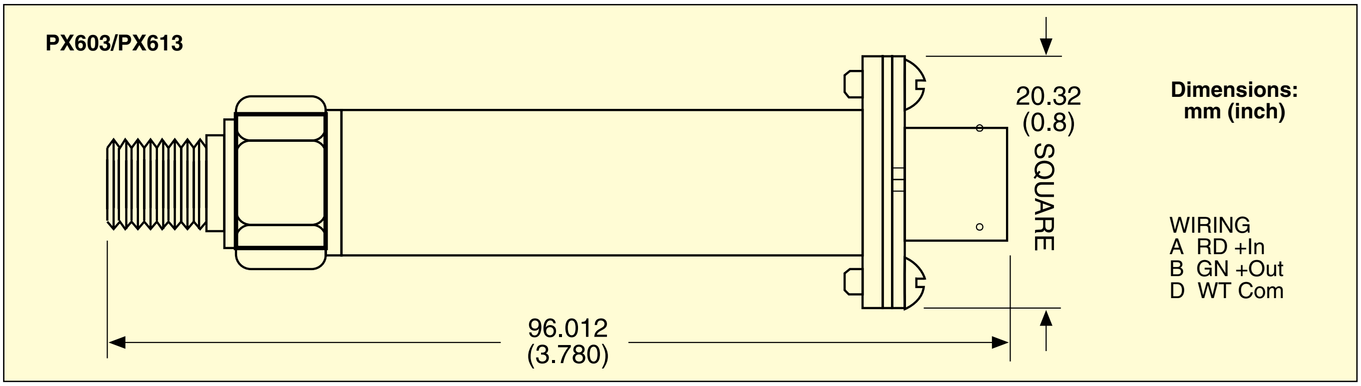
SPECIFICATIONS

Excitation: 10 to 30 Vdc unregulated
Output: 1 to 5 Vdc (3 wire)
Supply Current: <3.0 mA
Insulation Resistance: 100 MΩ @ 50V
Accuracy: ±0.4% BFSL
Hysteresis: ±0.2% full scale
Repeatability: ±0.05% full scale
Stability: ±1.0%/year
Durability: 100 million cycles
Operating Temperature: -48 to 90°C (-55 to 195°F)
Compensated Temperature: -29 to 82°C (-20 to 180°F)
Thermal Zero Effect: ±0.07% full scale/°C
Thermal Span Effect: ±0.07% full scale/°C
Proof Pressure:
- **15 to 2000 psi** = 200% full scale
- **3000 to 5000 psi** = 150% full scale
- **7500 to 20,000 psi** = 120% full scale

Burst Pressure:
- **15 to 2000 psi** = 800% full scale
- **3000 to 5000 psi** = 500% full scale
- **7500 to 20,000 psi** = 500% full scale

Gages: Thin-film polysilicone
Diaphragm: 17-4 PH stainless steel
Case: 300 Series stainless steel
Pressure Connection:
- **15 to 10,000 psi** = ¼ NPT
- **15,000 and 20,000 psi** = female Aminco fitting

Electrical Connection: 0.9 m (36") braided-shield PVC cable or connector
Mating Connector: PT06F8-4S (sold separately)
Weight: 128 g (4.5 oz) without cable
Response Time: 5 ms
Construction: Sealed units, except PX603 ≤500 psi is vented to room



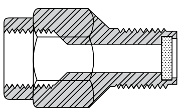
To Order

RANGE[†]	bar	MODEL NO. [*] INSERT 0 or 1	COMPATIBLE METERS**
Vacuum to 0 psig	-1 to 0	**PX6[*]3-30VAC5V**	DP3002-E, DP41-E, DP25B-E
Vacuum to 15 psig	-1 to 1.0	**PX6[*]3-30V15G5V**	DP3002-E, DP41-E, DP25B-E
Vacuum to 30 psig	-1 to 2.1	**PX6[*]3-30V30G5V**	DP3002-E, DP41-E, DP25B-E
Vacuum to 60 psig	-1 to 4.1	**PX6[*]3-30V60G5V**	DP3002-E, DP41-E, DP25B-E
0 to 15 psig	0 to 1.0	**PX6[*]3-015G5V**	DP3002-E, DP41-E, DP25B-E
0 to 30 psig	0 to 2.1	**PX6[*]3-030G5V**	DP3002-E, DP41-E, DP25B-E
0 to 60 psig	0 to 4.1	**PX6[*]3-060G5V**	DP3002-E, DP41-E, DP25B-E
0 to 100 psig	0 to 6.9	**PX6[*]3-100G5V**	DP3002-E, DP41-E, DP25B-E
0 to 150 psig	0 to 10.3	**PX6[*]3-150G5V**	DP3002-E, DP41-E, DP25B-E
0 to 200 psig	0 to 13.8	**PX6[*]3-200G5V**	DP3002-E, DP41-E, DP25B-E
0 to 300 psig	0 to 20.7	**PX6[*]3-300G5V**	DP3002-E, DP41-E, DP25B-E
0 to 500 psig	0 to 34.5	**PX6[*]3-500G5V**	DP3002-E, DP41-E, DP25B-E
0 to 1000 psig	0 to 68.9	**PX6[*]3-1KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 2000 psig	0 to 138	**PX6[*]3-2KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 3000 psig	0 to 207	**PX6[*]3-3KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 5000 psig	0 to 345	**PX6[*]3-5KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 7500 psig	0 to 517	**PX6[*]3-7.5KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 10,000 psig	0 to 689	**PX6[*]3-10KG5V**	DP3002-E, DP41-E, DP25B-E
0 to 15,000 psig	0 to 1034	**PX6[*]3-15KG5V†**	DP3002-E, DP41-E, DP25B-E
0 to 20,000 psig	0 to 1379	**PX6[*]3-20KG5V†**	DP3002-E, DP41-E, DP25B-E

Comes complete with operator's manual.
*** See **omega.com/meters** for compatible meters.*
†15,000 and 20,000 psi models supplied with female Aminco fitting.
*[*] Insert **"0"** for cable style, **"1"** for connector style.*
***Ordering Examples: PX603-500G5V,** cable-style transducer with 1 to 5 Vdc output and range of 0 to 500 psi.*
***PX613-500G5V,** connector-style transducer with 1 to 5 Vdc output and range of 0 to 500 psi.*
***PT06F8-4S,** mating connector (sold separately).*

ACCESSORIES

MODEL NO.	DESCRIPTION
PT06F8-4S	Mating connector for PX613 Series transducers
PS-4D	Pressure snubber, for oil, 225 to 500 SSU (10 to 50 SAE)
PS-4E	Pressure snubber, for water and light oil, 30 to 225 SSU
PS-4G	Pressure snubber, for air, stream and gases



¼ NPT Pressure Snubbers:
PS-4G = Gas
PS-4E = Lt Oil
PS-4D = Dense Lq

MIDWEST STEEL AND ALUMINUM

Toll Free: 1-888-744-6868

04
JUL

DAILY DEAL:
15% Off All Products

Item	Qty.	Width	Length	Tot. Wt.	Price	Discount	Total	[–]
7075 Aluminum Plate (7075ASHT2) **Size:** 2	2	17.0000	17.0000	116.756 #	$267.51	$40.13	**$454.76**	☐

» Continue Shopping

Item Total:	$535.02	Update
Discounts:	*-$80.26*	
Subtotal:	**$454.76**	

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