

Design of Subsea Energy Storage Chamber

By

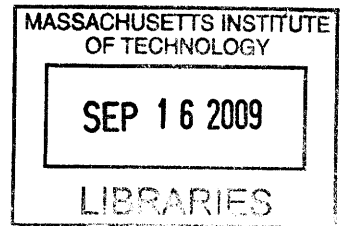
Alison S. Greenlee

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Signature of Author: _____

Alison Greenlee
Department of Mechanical Engineering
May 08, 2009

Certified by: _____

Alexander Slocum
Pappalardo Professor of Mechanical Engineering
Thesis Supervisor

Accepted by: _____

John H. Lienhard V
Collins Professor of Mechanical Engineering
Executive Officer

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Alison S. Greenlee

Submitted to the Department of Mechanical Engineering
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Requirements for the Degree of Bachelor of Science in
Mechanical Engineering

ABSTRACT

Energy generated from offshore resources is not reliable over short periods of time. Although wind and wave energy is fairly consistent in the long run, their short term capacity fluctuations prohibit these resources from replacing dependable fossil-fuel based energy systems. This limitation could be overcome if the energy harvested from these resources could be stored temporarily and then released when needed.

The large hydrostatic head at the ocean floor provides a unique opportunity for storing energy offshore. Similar to hydroelectric dams storing water at a high potential, energy could be stored offshore by displacing water from a subsea chamber. This chamber could be incorporated into the mooring system of present offshore harvesting devices to yield more favorable economics.

This report establishes the baseline assumptions for designing this energy storage device and proposes a methodology for constructing a beta level prototype. In addition to discerning the tradeoffs between different design options with respect to the marine environment, this study analyzes the cost of this structure per unit energy stored.

The contents of this report comprise of the following. First, the hazards inherent to the marine environment are explored qualitatively, and methods to address these issues are proposed. Second, the chamber shape, mooring type, and amount of material are determined based on their respective costs. Finally, this report concludes with the final dimensions of a proposed beta prototype and a list of recommendations for future work.

Thesis Supervisor: Alexander Slocum
Title: Pappalardo Professor of Mechanical Engineering

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I. LIST OF SYMBOLS

E	energy stored in storage tank
$V_{chamber}$	volume of energy storage tank
ρ_{water}	density of water (kg/m^3)
d	water depth (m)
g	gravitational constant (m/s^2)
C_1	conversion factor from Joules to kWh
F_b	buoyant force
C_2	fraction of chamber evacuated
M_a	muscle growth rate ($\text{kg/m}^2 \text{ year}$)
T	temperature of water ($^{\circ}\text{C}$)
$Cost$	monetary value (US\$)
$\rho_{material}$	material density (kg/m^3)
σ_{uts}^c	compressive strength of material (MPa)
C_3	loading safety factor
ρ_{rock}	rock fill density (kg/m^3)
w_{anchor}	weight of the anchor (N)
$\sigma_{pullout}$	anchor pullout strength (kg)
$Q_{total}(r)$	anchor pullout force as a function of the anchor's radius (N)
r_{anchor}	radius of anchor cross-section (m)
q_0	constant pullout strength for embedded suction anchor (kPa)
λ	constant scaling factor for soil conditions
ΔP	pressure difference between the hydrostatic and anchor pressure (kPa)
Q	desired anchor resistance (kN)
R_{LD}	ratio of skirt length to the diameter of the anchor cross-section
C_4	safety factor for anchor pressure vessel
$r_{chamber}$	radius of the storage tank (m)
F_{uplift}	uplifting mooring force (N)
C_5	safety factor for mooring system
SA_{vader}	surface area of the design profile (m^2)
T_{period}	average wave period (seconds)
T_{life}	lifespan of structure (40 years)
2η	height of design wave (m)
η_0	half of the average significant wave height
H_{design}	design wave height (m)
k	wave number (m^{-1})
S	endurance strength (dimensionless)
β	constant for endurance strength ratio (.0685)
R	ratio of minimum to maximum stress in loading cycle
$P_{response}$	pressure response from design wave (Pa)
P_{design}	design loading for structure (Pa)
N	number of loading cycles before failure
σ_{max1}	nominal maximum allowed stressed (Pa)
f_c	compressive strength of steel (MPa)
σ_{design}	design stress (MPa)
C_7	loading safety factor
σ_m	mean value of stress cycle
t_{design}	thickness of structure (m)

II. INTRODUCTION:

Energy generated from offshore resources is not reliable over short periods of time. Although wind and wave energy is fairly consistent in the long run, their short term capacity fluctuations prohibits these resources from replacing more dependable fossil-fuel based energy systems. This limitation could be overcome if the energy harvested from these resources could be stored temporarily and then released when needed.

The large hydrostatic head at the ocean floor provides a unique opportunity for storing energy offshore as pumped hydro. As its name suggests, pumped hydro stores potential energy by displacing water from the interior of a subsea tank that rests on the ocean floor. Similar to how hydroelectric dams store energy by keeping water at a high potential, energy could be stored offshore by displacing water from a subsea chamber. This chamber could be incorporated into the mooring system of present offshore energy harvesting devices, yielding favorable economics.

The goal of this thesis is two fold. First this thesis seeks to develop the methodology and background necessary for designing this subsea energy storage unit. This will involve the requirements for structural integrity along with an economics feasibility study. Second, this report will make baseline assumptions to dimension a beta level prototype and the cost per unit volume stored.

This thesis will approach the design of this structure by performing the following. To establish the goals of the prototype, the first section of this report will enumerate the high level functional requirements that this structure must fulfill. Second, base level assumptions will be established to enumerate the factors that constitute will determine how large the chamber must be to store a certain amount of energy at a given depth. Next, the nature of the marine environment will be discussed qualitatively, and future potential design hazards will be identified. The following section will analyze the tradeoffs between different chamber forms, and decide on the shape that the chamber will constitute. This will be followed by a review of mooring options and their respective design tradeoffs. The finalized structure will then be determined and dimensioned, based on a loading analysis. Finally, a list of future works will be presented.

Again, the scope of this report is to design the chamber and mooring structure of this energy storage device. More analysis will be needed to determine the economics and design of the pumping unit and installation process.

III. FUNCTIONAL REQUIREMENTS

The final design of the proposed energy storage unit must fulfill the following functional requirements. First, the energy storage unit has to maintain structural integrity for at least the lifespan of 40 years. While the requirements for maintaining structural integrity will be explored in later sections, let it suffice for now that the proposed design must sustain the high static and dynamic loadings that are encountered on the ocean floor. Furthermore, this structure must resist the abrasion, fouling, and chemical attacks inherent to a marine environment. Because maintenance is cost prohibitive, maintenance on the structure will not be possible after its initial installation. Therefore, the ESD must be designed to accommodate no maintenance over the duration of its forty year lifespan.

The proposed ESD's second functional requirement is to act as the mooring structure for the given OEGD. Being incorporated into the mooring's design is necessary for economic reasons

beyond the scope of this report. Suffice it to say that this system must provide the mooring support for the ESD to be considered for further development. Therefore, the design must be able to accommodate the resulting cyclic loading that will arise in this structure's charging and discharging.

IV. PRELIMINARY DESIGN PARAMETERS

The proposed prototype will be moored off the coast of Florida in approximately 30 m of water and will require 3 kWh of energy storage capacity. Before exploring possible structure configurations, preliminary calculations must be performed to establish some base level assumptions. These calculations are necessary for assuring design integrity. The following calculations will establish the design requirements for the ESD on a per unit energy storage basis. This will yield a generic set of equations that will allow the conceived structure to be modified for different environmental conditions.

The first assumption that will have to be established is the density of the water being displaced from the chamber. Although this first assumption is somewhat related to the environmental conditions that will be discussed in later sections, it is necessary for the present baseline calculations. The figure below presents the salinity content of the oceans on a per depth basis. Because the depths that are being considered for this particular project do not extend beyond a few hundred meters, it is safe to assume a constant density of 1025 kg/m^3 .

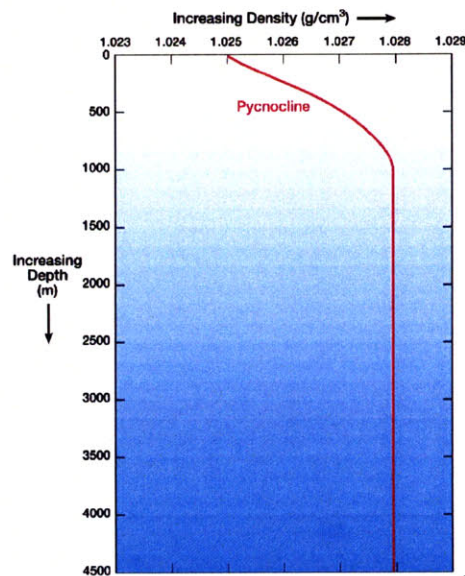


Figure 1: Ocean Pycnocline for T = 20°C [5]

To now derive the amount of energy that can be stored in the chamber per unit volume, another assumption must be made about the capabilities of the ESD's pumping unit. This preliminary report will assume that the pumping device that will be able to evacuate the ESD to a full vacuum. Although this assumption will need to be modified once the capabilities of the pumping unit have been established, for the purposes of this report, it will be assumed that the chamber is evacuated to a total vacuum. Furthermore, the ocean will be treated as a constant pressure head. Combined, these two assumptions entail that the energy stored by the ESD is equal to the product of the chamber's volume and the hydrostatic pressure of its surrounding environment. This relationship is given below to be:

$$E := V_{\text{chamber}} \cdot (\rho_{\text{water}} \cdot g \cdot d + 10^5)$$

where ρ_{water} is the density of the water, g is the gravitational constant, d is the depth of water, and V_{chamber} is the volume of the chamber. Notice that because the chamber is being evacuated to vacuum, atmospheric pressure must be taken into account as the difference in pressures. The volume then needed to store one kilowatt hour is therefore directly related to the depth of the device and is given by the expression below as:

$$\frac{V_{\text{chamber}}}{\text{kWh}} := \frac{C_1}{(\rho_{\text{water}} \cdot g \cdot d + 10^5)}$$

where C_1 is a conversion factor from Joules to kWh. As shown in the figure below, the amount of volume needed to store a constant amount of energy decreases nonlinearly, approaching an asymptote. The energy density per unit volume will play a significant role in the economic analysis that will be performed in a later section.

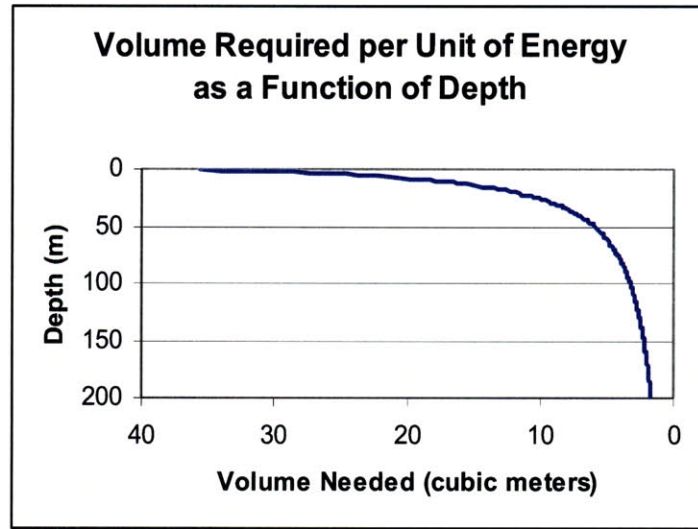


Figure 2: Specific Volume Required as a Function of Depth

Using the above calculations, the preliminary volume of this prototype structure will be just under 10 m^3 . In the next section, this volume may be increased because of additional environmental factors (like suspended sand settling to the bottom of the chamber).

Another factor that will influence the design of this structure is buoyancy. Once evacuated, a tremendous amount of counterweight will be needed just to keep the ESD on the bottom of the ocean. The amount of buoyant force per unit volume and the subsequent buoyant force per unit of stored energy are expressed below,

$$\frac{F_b}{V_{\text{chamber}}} := C_2 \cdot \rho_{\text{water}} \cdot g$$

$$\frac{F_b}{\text{kWh}} := \frac{(C_2 \cdot C_1 \cdot \rho_{\text{water}} \cdot g)}{(\rho_{\text{water}} \cdot g \cdot d + 10^5)}$$

where F_b is buoyant force, and C_2 is the fraction of the chamber evacuated. Interestingly, the amount of buoyant force exerted on the ESD is independent of the depth of the device or the state of the sea. Therefore, the amount of mooring force needed to keep this structure on the bottom is only dependent on the volume, weight, and mooring loads of the structure. This force may in fact be the most critical parameter in determining the design of this structure. Furthermore, as will be seen later, this force will be a critical factor that drives the economics and, consequentially, the design of this project. For the conceived prototype, the static derived buoyant force would then be just under 100 kN.

The nature of the floating energy device governs the uplifting mooring forces on the ESD. Again, this report's objective is to design a generic ESD that can be modified for any floating device. It will be assumed for this report that the floating energy generated unit will be completely submerged. Although this may not be true for all floating energy devices, this assumption simplifies the calculations needed because the uplift on the structure will remain constant. As discussed previously, the buoyant force on a buoy is independent of its depth. Therefore, if the buoy is completely submerged at all times, the uplift exerted on the structure is kept constant. In the future, this assumption will need to be checked against the nature of the energy generating device. However, for the purposes of this report, a constant F_{uplift} will be assumed.

V. MARINE ENVIRONMENT CHARACTERIZATION

The design of this structure will be governed by the nature of the marine environment. It is important to note that although generalizations can be made of the marine environment, every environment is different and a more in-depth analysis of a specific area's environment will be necessary before this device is installed. As it pertains to the design of this structure, the marine environment can be separated into two sections: marine fouling and marine loading. This section will characterize and begin to address the corrosion and fouling challenges that the marine environment poses. Marine fouling can then be even further divided up into two categories: organic and ionic. Both contribute to the wear of the structure and must be accounted for in both the material selection and the overall design.

a. Organic Marine Environment

i. External Fouling

Fouling is defined as the amassing of organisms onto the sides of a structure. Although the type of growth is location dependent for shallow water depths, the organisms found in water depths greater than 100 m are the same around the world^[3]. In these depths, muscles are the dominant growth type, and their growth is dependent on the seasonal temperature of the water. Marshall gives the empirical equation, shown below, to quantify this growth as:

$$M_a := 2 \cdot T - 1$$

where M_a is the rate of muscle fouling growth (kg/m^2), T is the temperature of the water in Celsius. Although this expression is useful for determining initial growth rates, the rate decreases

over time, and the accumulation of marine growth levels out. As shown below, with the case of the North Sea, the accumulated growth levels out after the first few years.

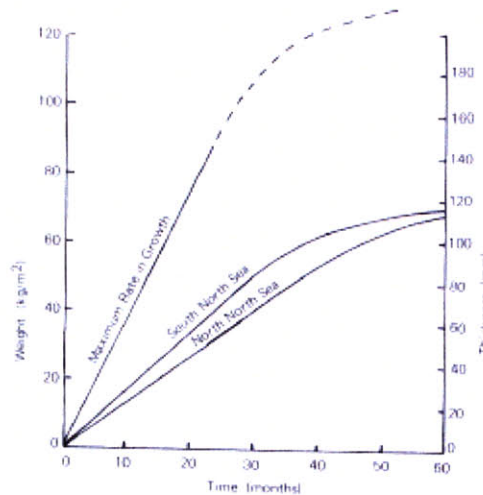


Figure 3: Marine Growth Rates for Specific Regions^[3]

Because the prototype will be installed off the coast of Florida, where temperatures would entail a near optimal environment for marine growth, it will be assumed that 180 mm of growth will amass on the exterior of this structure over time.

This growth could be problematic and restrict water from being pumped in and out of the chamber. A reduction in the diameter of this area could also strain the pump, reducing its life in the long run. Further investigation must be performed to address this risk.

ii. Internal Fouling

Growth amassing inside the ESD chamber would also give rise to problems. Although green algae and other photosynthesizing organisms would not be able to grow inside the dark chamber, muscles and other organisms could. In fact, the sheltered nature of the tank may provide an ideal habitat for these organisms. Their residency inside the tank could result in the following two issues. First, the space they occupy would take away from the amount of vacant space available for energy storage. Even after their death, their carcasses would be trapped in the tank, leading to even less of the tank being utilized for energy storage. Second, the dead organisms would remain in the tank and decay. Decaying organic material may attract many corrosive microorganisms, like sulfur-reducing bacteria, which may compromise the integrity of the structure.

One option that could prevent marine growth inside the structure may be to have either a protective coating or inhibitive membrane inside the structure. This study is unsure if there is such a coating in existence that would be effective for the structure's lifetime. Therefore, further study must be performed to analyze this risk.

iii. Suspended Organisms

The final aspect of the organic marine environment that will be discussed is the presence of suspended organisms. Similar to the threat of internal marine growth, the suspended organisms (like floating seaweed) may flow in with the seawater and settle to the bottom of the chamber, remaining trapped.

The marine industry has already developed a solution for this problem. To prevent suspended organisms from entering the engine cooling systems, the marine industry has developed the sea strainer. These devices run the incoming ocean water, one way, through a screen before the water enters more sensitive machinery. As shown in the figure below, the separated material is stored in the suspended center filter.



Figure 4: Potential Sea Strainer

Presently, the intake lines of this system only flow one way and the system needs must therefore be emptied manually. If this system were to be adapted for the ESD, where the flow will go both ways, this system would be able to back flush itself. Although further investigation must be made to characterize the size a strainer, this report highly recommends that a sea strainer be coupled with the pumping system to prevent floating ocean life from entering the ESD chamber.

b. Ionic Environment

In addition to the organic challenges of the marine environment, the water's ionic content makes the ocean an extremely corrosive environment. The marine industry as well as the oil industry has had extensive experience with mitigating corrosion on structures. Their present methods include placing sacrificial anodes to neutralize the surrounding seawater and choosing materials that are particularly resistant to this environment. This report will address the issues associated with corrosion by selecting the appropriate construction materials in the upcoming section.

VI. DESIGN OF ENERGY STORAGE DEVICE

a. Construction Materials

Because maintenance on the ESD will be cost prohibitive, the materials chosen for the ESD must be able to maintain structural integrity throughout the structure's lifetime without maintenance. As previously stated, the oil and marine vessel industry have had extensive experience in choosing construction materials for marine structures. This study will therefore consider choose the ESD's material from these industries' three most commonly used materials. These materials include carbon steel, stainless steel, and concrete.

The construction material will be chosen based on the following criteria. First and foremost, the chosen material must be able to resist the corrosive ions that are inherent to the marine environment. Second, the material must be able to endure the installation and environmental loadings that the ESD will be subjected to. Finally, because this is an economically driven project, the material will be chosen on cost per unit compressive strength.

The first material that will be considered is marine grade cement. The marine industry has increasingly been using this material for projects because of its relatively high compression strength, high resistance to corrosion, and low cost. In particular, Lytag® concrete has been used for subsea oil storage tanks in the North Sea and for marine concrete bridges^[5]. Unfortunately, concrete has very poor tensile strength properties. To use concrete alone for this structure would inherently be very risky because of the potential for tensile loads during installation. Therefore, if the ESD is to be made out of concrete, the structure must be reinforced with either post-tensioning cables or rebar. These reinforcements increase the cost of materials and the complexity of manufacturing. The assumptions that will be made for this construction material will be that the cost per tonne of concrete is \$90. Also, because ASTM regulation requires that concrete in contact with abrasive soil have a minimum strength of 60 MPa this will be the assumed strength of the material^[6]. Further assumptions on this material can be found in Table 1 near the end of this section.

The next materials that will be examined are steel composites. Unlike concrete, steel has much more impressive tensile and compressive strength properties, making it a more robust material. Because the ESD will be a variety of unpredicted loading on the structure during installation, the ability of the material to handle both types of loading is crucial. The two types of steel that will be observed will be carbon steel and a stainless steel. This preliminary comparison will narrow down the potential materials significantly; however, more investigation is needed before the final material for the prototype is chosen.

Stainless steel is well known for its excellent resistance to corrosion, non-magnetic nature, and workability. This material trades strength for ductility, making it better able to resist impact forces. However, this material is very expensive, and is substantially weaker than its other steel contemporaries, as shown in the table below^[3].

Carbon steel, on the other hand, has very high tensile and compression strengths and has been used for many projects in the past. Unfortunately, carbon steel does not have the same high resistance to corrosion that stainless steel does, but this weakness can be easily mitigated by placing a series of sacrificial anodes around the ESD structure. Another method to inhibit corrosion on the structure would be to coat the carbon steel with a more corrosive resistant material, like marine cement or stainless steel. Both of these corrosion countermeasures are already established practices used in industry. The assumptions for the costs and properties of carbon steel are further described in the table below.

The table below shows a summary of the material options for this structure. With the proper precautions taken, each of these options is capable of withstanding the environmental conditions subjected to the ESD. However, steel would be more robust than concrete to the potential tensile loading exerted on the structure during installation. Therefore, concrete is eliminated from the potential construction materials, and, of the remaining construction materials, cost per unit strength will be the deciding criteria. This cost can be found with the following equation:

$$\frac{\text{Cost}}{\text{MPa}} := \left(\frac{\text{Cost}}{\text{tonne}} \right) \cdot \frac{\rho_{\text{material}}}{\sigma_{\text{uts}}}$$

where ρ_{material} is the material density and $\sigma_{\text{uts}}^{\circ}$ is the compressive strength of the material. Based on this metric, the cheapest material is low carbon steel. The design of the ESD will therefore use the material properties and associated costs of the low carbon steel, as shown in the table below.

Material	Density (tonne/m ³)	Compressive Strength (MPa)	Tensile Strength (MPa)	Corrosion Resistance	Cost/tonne (\$/tonne)	Cost/compressive strength (\$/MPa)
Low Carbon Steel (A36)	7.86	1480 ¹	593	~~~~	949.91 ²	5.04
High Carbon Steel (HRC)	7.488	2160	717	~~~~	608.6 ²	2.11
Stainless Steel (CR 304)	8	707	215	Great	1366.85 ²	15.47
Lyttag Concrete Fume	1.97	60	----	Great	90	2.96

Table 1: Comparison of Potential Construction Materials

Because low carbon steel was chosen as the construction material, mitigation techniques must be used to prevent the corrosion of this material over time. This report recommends that zinc sacrificial anodes be used to mitigate corrosion because of their inherent simplicity to the design. Furthermore, this report recommends investigating the methods used to shape this material for a better understanding of the material's capabilities and costs.

b. Chamber Shapes

The basic form of the ESD must be designed to withstand the large subsea hydrostatic pressures when the chamber is evacuated. This section will examine the inherent benefits of designing the chamber to assume the form of a hexagonal prism, a cylindrical extrusion with a flat base, a cylinder extrusion with a spherical end-cap, and a sphere. These forms will be evaluated based on the following criteria. First, the proposed form must yield minimal stress concentrations. Substantial stress concentrations could give way to a variety of problems, including shape distortion and loss of structural integrity. Also, heterogeneous loading would require more material to maintain structure integrity, adding to the cost of the structure. The inspection of stress concentration will be visually conducted using the COSMOSXpress Wizard Analysis feature in Solidworks. Next, the form will be evaluated by its ratio of surface area to volume. The smaller this ratio, the smaller the amount of material needed for this structure's construction.

A polygon form, or in this case a hexagonal prism, provides a number of problems when subjected to hydrostatic loading. Although the manufacturing of this shape is relatively simple, the stress concentrations inherent to this structure will require a large safety factor in its design as well as reinforcements along the surface planes. As shown in the COSMOSXpress figure

¹ The steel material properties were taken from matweb.com. Further material description can be found in Appendix A

² The steel prices were taken from Steelbb.com on April 21, 2009. This website reveals the current world market price of steel.

below, the stresses on the structure's faces are over an order of magnitude greater than the stresses found at the edges. Furthermore, the ratio of the surface area to volume is significantly lower than that of the cylinder or sphere form.

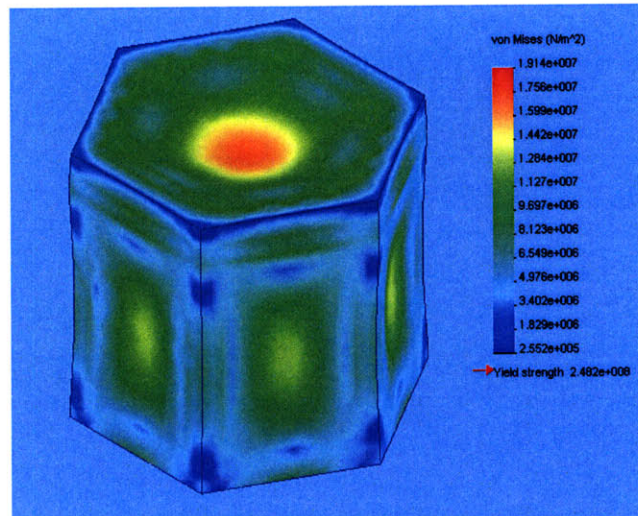


Figure 5: Stress Distribution of Hexagonal Prism Under to Hydrostatic Loading

A cylindrical shape overcomes the limitations of the polygon prism and distributes loading fairly evenly around the body of the structure. The stress gradient on the top of the structure can be overcome with a spherical end cap, as shown below. This even distribution of loading yields a lower necessary safety factor. Furthermore, the surface area to volume ratio improves significantly over the polygon prism as well.

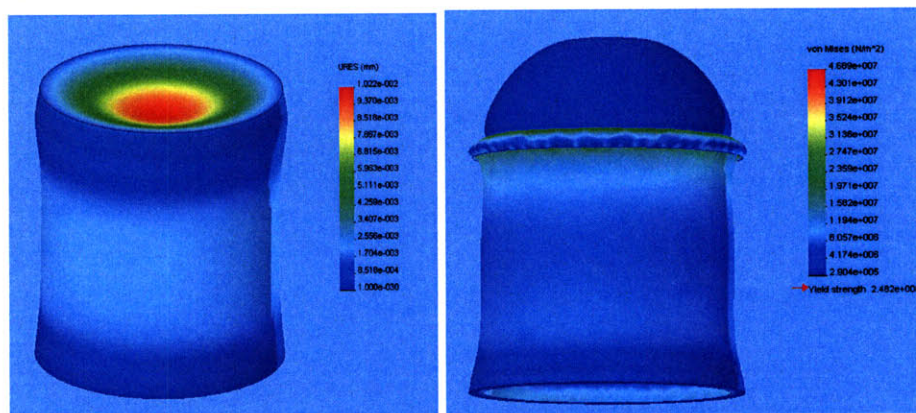


Figure 6: Stress Distribution of Two Cylindrical Geometries Under Hydrostatic Loading

The above spherical end cap shape is in fact already used as the form for caisson for Condeep offshore structures.

The final form that will be investigated is a sphere. A sphere has the largest volume to surface area ratio possible. Furthermore, the distribution of the loading throughout the structure is uniform. Therefore, this shape yields the lowest material needed for a given volume.

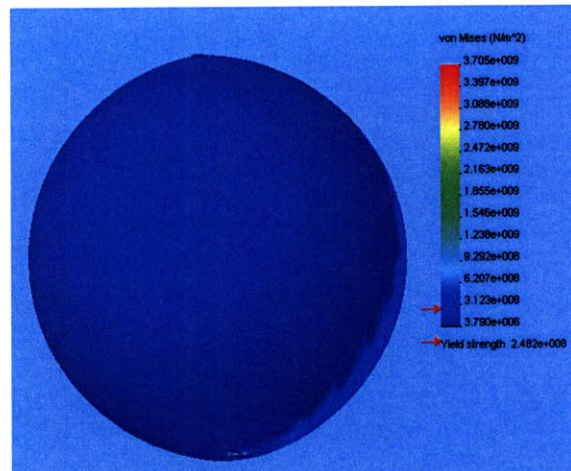


Figure 7: Sphere Stress Distribution Under to Hydrostatic Loading

The ESD will therefore adopt a spherical shape for the afore mentioned reasons. Once the mooring component of the ESD is determined in the next section, the dimensions of this tank will be defined.

c. Mooring Types

The proposed mooring options will be evaluated based on the following criteria. First and foremost, the chosen mooring system must be capable of remaining in an absolute location over the course of its lifetime. If this criterion can be met, the mooring system will be chosen based on the cost of the mooring system per unit of counteracted force.

i. Gravity Anchor

The first mooring option that will be considered is gravity anchor. The ESD structure could be designed to have a large containment area for where cheap, rock fill could be placed on top, as shown below.

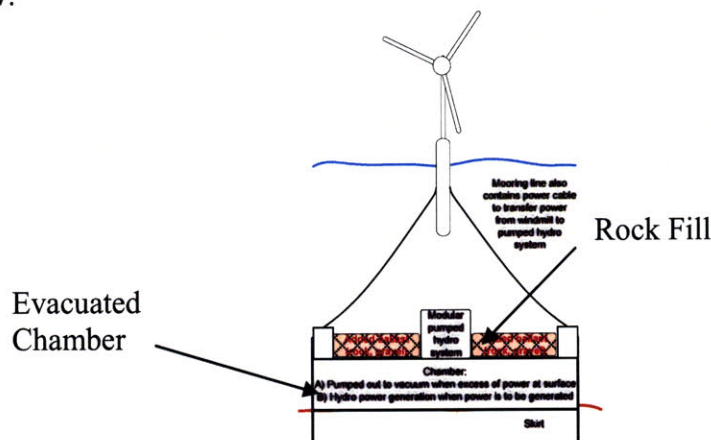


Figure 8: Example of Gravity Based Mooring System

A gypsum composite, like the present material used for many gravity dams, will be the assumed rock fill. Its associated cost is assumed to be \$45/tonne, and its density is assumed to be 2.3 kg/m³.^[7] For the rock fill, the cost required to counteract a Newton of uplift on the structure is given by the equation below.

$$\frac{\text{Cost}}{\text{Newton}_{\text{uplift}}} := C_3 \cdot \left(\frac{\text{Cost}}{\text{tonne}} \right) \cdot \frac{\rho_{\text{rock}}^2}{g \cdot (\rho_{\text{rock}} - \rho_{\text{water}})}$$

Where ρ_{rock} is the density of the rock and C_3 is the loading safety factor. For a safety factor of 1.5, the cost of uplift resistance is \$19.05/kN. Additional costs for the modification of the structure to accommodate for this loading would be minimal and, for this study, will be disregarded.

ii. Mushroom Anchors

The second mooring option that will be explored is a mushroom anchor. This anchor type actually sinks into the soil over time, and uses the suction of the ground to resist a significantly higher amount of force than the gravity anchor on a per weight basis. However the pullout strength of this anchor is very dependent on the soil conditions of the area, as can be seen in the table below.

COMPARISON OF BREAKOUT FORCES FOR VARIOUS ANCHORS		
ANCHOR TYPE	BOTTOM CONDITION	BREAKOUT FORCE (KG)
Mushroom anchor 160kg	1.5m sunk in mud	909
Mushroom anchor 230kg	sand	775

Figure 9: Anchor Breakout Force Comparison provided by G. Bourgos Co.

But even more significantly, mushroom anchors take a significant amount of time to settle into the soil (on the order of weeks). This may then require a two stage installation process, where the tugboat drops the ESD and then comes back later to add on the buoy, adding substantially to the cost of a structure. Further investigation of the time required to assure the integrity of this mooring system must be conducted if this system is chosen. For now, the cost of only the mushroom anchor itself will be used as the base level analysis for the cost per Newton resistance. From the Bourgos table listed above, the anchor properties for a sandy ocean floor will be assumed. The equation for finding the cost per Newton uplift is given by the expression below.

$$\frac{\text{Cost}}{\text{Newton}_{\text{uplift}}} := C_3 \cdot \left(\frac{\text{Cost}}{\text{tonne}} \right) \cdot \left(\frac{w_{\text{anchor}}}{g \cdot \sigma_{\text{pullout}}} \right)$$

where Cost/tonne is the that of low carbon steel, w_{anchor} is the weight of the anchor (kg), σ_{pullout} is the anchor pullout strength (N). Using the previous assumption for the cost of low carbon steel, this anchor type will cost \$17.57/kN uplift. The cost per unit resistance for this anchor type is less then that of the basic gravity anchor; however, the additional cost of installation may make this option more expensive. So because there is less than a 15% difference in material cost, the gravity anchor would be chosen over this option.

iii. Embedded Suction Anchors

The last mooring system that will be investigated is an embedded suction anchor. Originally developed for North Sea applications, this anchor typically involves a hollowed out cylinder or 'can' with an attached pumping unit, similar to the one that will be used for the evacuation of the

ESD chamber. Once the can is placed on the ocean floor, the pump engages and sucks the anchor into the ocean. In addition to being a formidable anchor, this structure can handle significantly more horizontal loading than conventional anchors.^[9] This would allow for a smaller mooring radius, resulting in less high tension cable being needed for the floating energy generating device. Although this cost saver will not be used in this initial cost analysis, it will be important for future discussion.

As the holding capacity of this anchor type is largely dependent on the nature of the ocean floor, a bottom survey is needed before any structure is installed. For installation off the coast of Florida, the ocean floor will be assumed to comprise of loose sand.

Many studies have been conducted to define the nature of these anchors, and the following relationships have been found. Qualitatively, the main geometric parameters that govern the anchor capacity are the diameter of the pile and the length of the skirt. Because the capacity of the anchor is largely set by the friction between the anchor and the surrounding soil, the amount of surface area in contact with the soil governs the initial holding capacity.^[10] It is important to note that the anchor resistance increases significantly with a denser soil media. In the graph below, each line corresponds to a suction anchor with a different diameter. This figure shows how the holding capacity of the anchor per unit pressure increases as the surface area of the anchor rises.

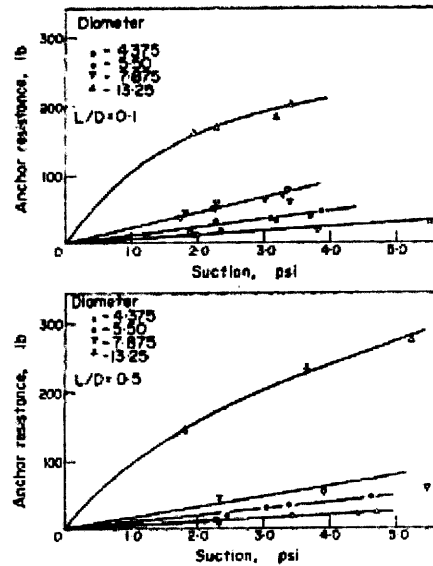


FIG. 10. Test results for loose sand.

Figure 10: Anchor Resistance as a Function of Cross-Sectional Area for Loose Sand^[8]

The curves above imply a linear relationship between the suction pressure and the anchor capacity for a constant ratio between the skirt length and the diameter of the anchor. This linear relationship between the anchor's capacity and the pressure difference is given by the equation below.

$$Q_{\text{total}}(r_{\text{anchor}}) := (q_0 + \lambda \cdot \Delta P) \cdot \pi \cdot r_{\text{anchor}}^2$$

where $Q_{\text{total}}(r_{\text{anchor}})$ is the total pullout force of the anchor, q_0 is a constant specific to the soil conditions, r_{anchor} is the radius of the cross-sectional area of the anchor in contact with the ocean floor, λ is a constant dependent on the soil conditions, and ΔP is the pressure difference between

the ocean hydrostatic and the internal pressure. For a loose sand condition, the lambda is assumed to be 2.6, the ratio of skirt length to diameter is 0.5, and the q_0 is assumed to be 2.41 kPa (10). Also, the delta P will be set to the maximum allowed pressure differential of loose sand which is approximately 40 kPa (11). These values show a quadratic relationship between the radius of the suction pile and the holding strength, as shown in the figure below.

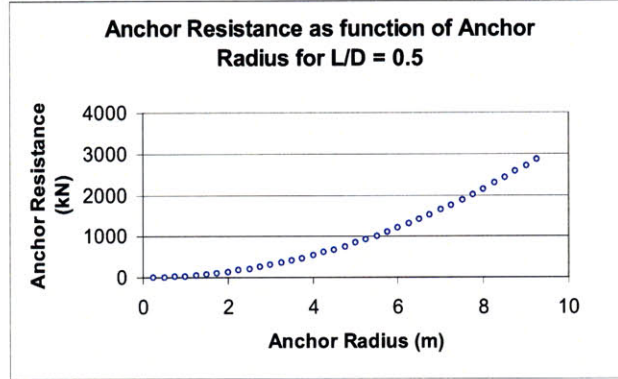


Figure 11: Anchor Resistance as a function of the Anchor Radius for L/D = 0.5

For this anchor type, the cost per unit force resisted can be found with the following base level assumptions. First, it will be assumed that this anchor is a cylinder with a length to diameter ratio of .5 and will be made of the same steel as the storage chamber. The wall thickness will be determined using thin-walled pressure vessel approximations where the loading is the pressure differential of the sand, multiplied by a safety factor of 20 to accommodate for uncertainty of soil conditions^[9]. Using thin pressure vessel approximations, the structure's thickness and corresponding volume of steel can be found. The cost per kN of resistance is expressed by the equation below:

$$\frac{\text{Cost}(Q)}{\text{kN}} := \frac{\text{Cost}}{\text{tonne}} \cdot (\sqrt{\pi \cdot Q}) \cdot \frac{(4 \cdot R_{LD} + 1)}{\rho_{\text{material}}} \cdot \left(\frac{1}{q_0 + \lambda \cdot \Delta P} \right)^{1.5} \cdot \left[\frac{(C_4 \cdot \Delta P)}{2 \cdot (\sigma_{\text{uts}})} \right]$$

where Q is the desired resistance, R_{LD} is the ratio the skirt length to the diameter (.5), C_4 is the safety factor for the pressure vessel, and the σ_{uts} is the ultimate strength of the steel. Because the per unit cost changes as a function of the resistance, the advantage of this anchoring system over the gravity anchor is different for each value. However, the cost of resistance is significantly less for all reasonable loading types, less than a few cents per kN resistance. This cost assumption does not include that associated with the pump required for this anchor type. It will be assumed that the pump that evacuates the storage chamber can be used for this suction as well.

Because of its significant inherent cost reduction over the other options, an embedded suction anchor will be used for this device's mooring type. However, it is important to note that the cost per unit resistance formula will only be used to distinguish the embedded suction anchors from other mooring types in this section. This cost per unit resistance will not be used for further economic analysis. The next section of this report will use the governing equations of the embedded anchor system to determine the combined form of the ESD and mooring system.

VII. FINAL SYSTEM DESIGN

At this point of the report, it has been determined that the tank of the ESD will be spherically shaped and that the mooring method will be an embedded suction anchor. The final design of

this ESD structure will be chosen based on its ability to minimize the amount of material needed for this structure. This minimization will be accomplished in two ways. First, because wall thickness will be governed by the magnitude of the stress concentrations in the structure, the final design must distribute the structure's loading as evenly as possible. Each proposed design's respective ability to distribute loading will be visually inspected with SolidWorks COSMOSXpress Analysis Wizard. The COSMOSXpress analysis assumes that the chamber is at a vacuum and that the difference between the external hydrostatic pressure and that inside the suction anchor is 40 kPa. The second way to minimize the amount of material needed for this structure is to have the tank and embedded suction anchor share boundaries. If these components can share a wall, less material will be needed, reducing the total cost of the ESD even more.

a. Required Minimum Anchor/Chamber Dimensions

Before potential designs are considered, it is important to note that the dimensions of the structure will be governed by the relationship between the radius of the embedded suction anchor, the length of the anchor's skirt, and the radius of the evacuated chamber. As discussed in the previous section, the radius of the anchor governs its resistance. Also, the resistance needed for the mooring is a function of the radius evacuation chamber and the uplift of the buoy. Using the theory derived in the previous section, the radius of the anchor is related to the radius of the evacuation chamber and the uplift force generated by the buoy, as shown in the expression below:

$$r_{\text{anchor}} := C_5 \left[\frac{\left[\rho \cdot g \cdot \left(4 \cdot \frac{\pi}{3} \cdot r_{\text{chamber}}^3 \right) + F_{\text{uplift}} \right]}{\pi \cdot (q_0 + \lambda \cdot \Delta P)} \right]^{.5}$$

where r_{anchor} is the radius of the anchor's cross-section, C_5 is a safety factor, r_{chamber} is the radius of the spherical chamber, and F_{uplift} is the mooring uplift force generated by the buoy. Although the F_{uplift} is important to the sizing of this system, this force will not be defined for this preliminary design and therefore disregarded for the sizing of the ESD. It is important to note that the radius of the anchor is independent of the structure's depth. As discussed in the previous section, it is also important to note that the length of the skirt length is assumed to be equal to the radius of the anchor. As shown in the figure below, the radius of the anchor will be larger than that of the chamber for all considered volumes.

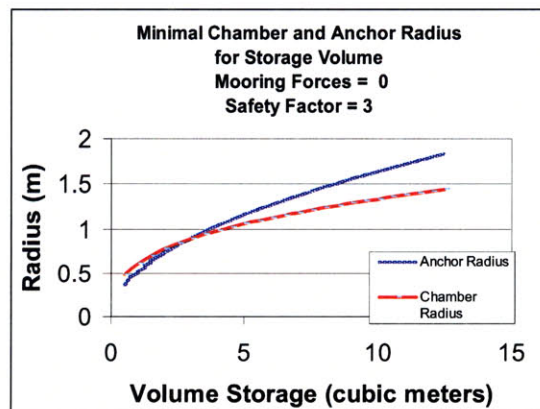


Figure 13: Chamber and Anchor Radius as a Function of the Storage Volume

b. Structure Design

While adhering to the afore stated relations, there are three potential designs that will be considered for this structure. A cross-section of the first potential design and its COSMOSXpress analysis are shown below. This design comprises involves forming the suction anchor to be a sphere, and having the internal chamber scaled to the above stated relations. Although this structure saves material by having the vacuum chamber and suction anchor share a wall, this designs in large stress concentrations in the chamber.

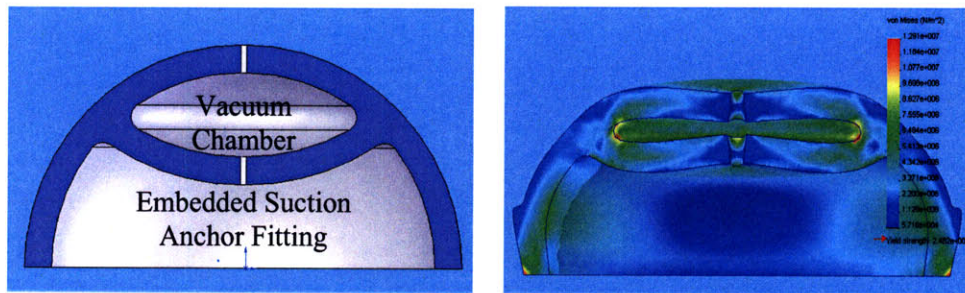


Figure 14: Unbarred Hemisphere Design

The next potential design would also have a dome casing; however, this design would use a central support to counteract the stress concentrations inherent to the first. This would require more material, and there would still be significant stress gradients in the design.

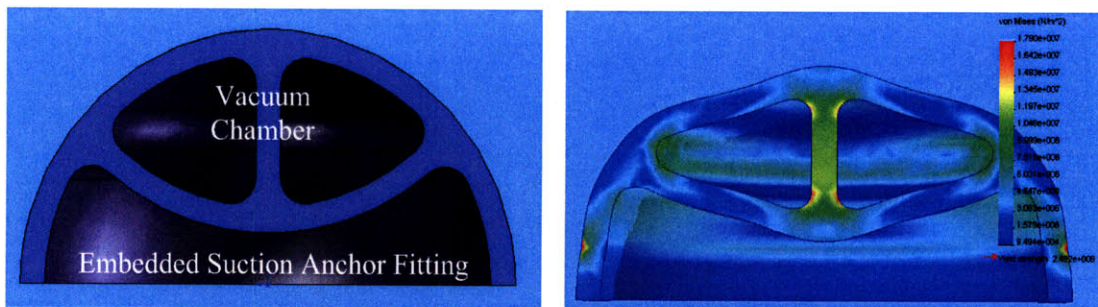


Figure 15: Barred Hemisphere Design

For the final design option, the vacuum chamber would be completely spherical, and the sides of the embedded suction anchor would constitute a flat panel that revolves around the chamber's midpoint. Resembling the profile of Darth Vader's helmet, this design yields minimal stress concentrations, as shown in the figure below. This balancing of the external 'forces' greatly reduces the amount of material required. The Darth Vader helmet profile will therefore be the chosen one for the design of the ESD.

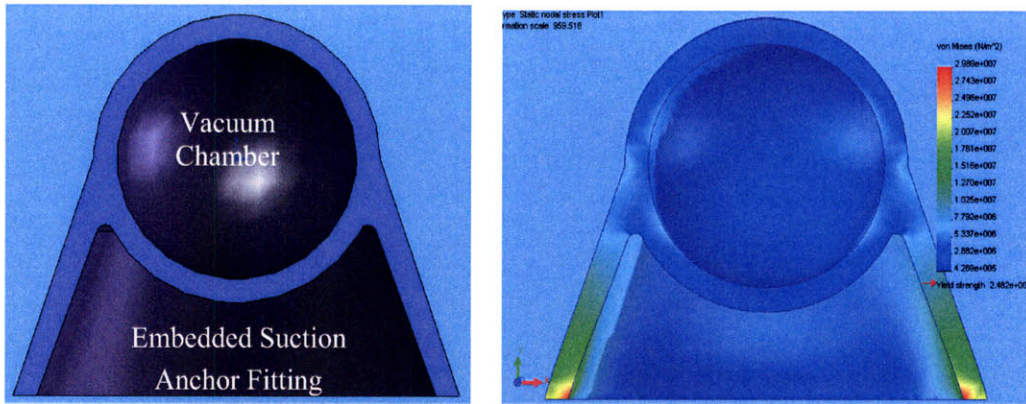


Figure 16: Darth Vader Helmet Profile

c. *Minimum Quantity of Structural Material*

i. Structure Surface Area

The amount of steel needed to build this structure must now be found as a function of chamber volume. Using thin walled assumptions, the amount of steel needed would be the product of structure's surface area and its wall thickness. Assuming that the anchor wall resembles a trapezoid when spread out, the surface area of the structure can be found by the following equation:

$$SA_{\text{vader}} := 4 \cdot \pi \cdot r_{\text{chamber}}^2 + \left[\frac{(r_{\text{anchor}} - r_{\text{chamber}})}{2} \right] \cdot \sqrt{r_{\text{anchor}}^2 + (r_{\text{anchor}} - r_{\text{chamber}})^2}$$

where SA_{vader} is the surface area of the structure. Using the definitions of r_{anchor} and r_{chamber} that were defined earlier, this expression can be rewritten in terms of the volume of the chamber.

$$SA_{\text{vader}} := (12 \cdot \pi \cdot V_{\text{chamber}})^{\frac{1}{3}} + \left[\frac{a}{2} \cdot \sqrt{V_{\text{chamber}}} - \frac{3(V_{\text{chamber}})^{\frac{1}{3}}}{8 \cdot \pi} \right] \cdot \sqrt{a^2 \cdot V_{\text{chamber}} + \left(a \cdot V_{\text{chamber}}^{\frac{1}{2}} - \frac{3 \cdot V_{\text{chamber}}^{\frac{1}{3}}}{4 \cdot \pi} \right)^2}$$

$$a := C_5 \cdot \left[\frac{(4 \cdot \rho \cdot g)}{3 \cdot (q_0 + \lambda \cdot \Delta P)} \right]^{-5}$$

For the system prototype, the volume of the chamber will be approximately 10 m^3 . Therefore, the surface area of the material will be 22.7 m^2 . Again, this surface area assumes that no uplifting mooring force is being applied to the structure. The thickness of the structure will now be found as a function of the structure's loading.

ii. Thickness Derivation via Loading Analysis

This section will determine the thickness of the structure using the following procedure. First, the maximum loading exerted on this structure will be determined for a given location. Second, assumptions on the number of loading cycles that this structure will undergo over the course of its lifetime will be defined. This number of cycles will be used to find the maximum stress that will be allowed for the structure to maintain structural integrity over the course of its lifetime. Finally, this allowed maximum stress, maximum loading, and chamber radius will be used to

find the structure's thickness. This loading analysis will not take into account the mooring forces that will be subjected to the structure.

1. Maximum Structural Loading

The largest possible load that can be subjected to the ESD is a function of the design wave for the structure's location. A design wave is the statistically highest wave that the structure could encounter in its lifetime. Designing for this wave will result in an overestimate of the loadings on the ESD that will ensure the integrity of the structure.

To determine the design wave, it is assumed that the wave spectrum resembles a Rayleigh distribution with a mean equal to the significant wave height for a specific area. This spectrum is used in conjunction with the average wave period for the area to find the statistically highest wave that the structure would encounter in lifespan. As shown with the equation below, the design wave (2η) is found by equating the cumulative distribution function of the wave spectrum to the reciprocal of the number of waves that occur over the structure's lifetime. The number of waves is equal to the average wave period (T_{period}) divided by the system's lifespan (T_{life}).

$$\frac{T_{\text{period}}}{T_{\text{life}}} := \frac{(2\eta)}{\eta_0^2} \cdot e^{\left(\frac{-\eta^2}{\eta_0^2}\right)}$$

where η_0 is half of the average significant wave height. For a structure in the Gulf of Mexico, it is assumed that the significant wave height is 3 m, the significant time period is 6.3 seconds.^[2] The design wave for our structure will therefore be 12.8 m.

This design wave would result in a pressure response on the ocean floor. This pressure response is a function of the wave height and the ocean depth at that location, as shown in the equation below:

$$P_{\text{response}} := \frac{(\rho_{\text{water}} \cdot g \cdot H_{\text{design}})}{2 \cdot \cosh(k \cdot d)}$$

where P_{response} is the pressure response at the ocean floor, H_{design} is the design wave height, and k is the wave number. For the prototype, the pressure response from the design wave at a depth of 30 m would be approximately 0.4 atmospheres. The sum of this pressure and the hydrostatic pressure would equal the maximum loading exerted on the ESD in the span of 40 years, which is given by the expression below.

$$P_{\text{design}} := \rho_{\text{water}} \cdot g \cdot d + \frac{\rho_{\text{water}} \cdot g \cdot H_{\text{design}}}{2 \cdot \cosh(k \cdot d)}$$

where P_{design} is the design loading. Under the assumption that the chamber will be completely evacuated to a vacuum, this design pressure will be equivalent to the design loading.

2. Cyclic Loading Analysis

This design will assume that this structure undergoes only one loading scheme. This loading scheme arises from the charging and discharging of the ESD, where the chamber is pumped to vacuum pressure and then refilled. It will be assumed that the ESD chamber will evacuate 20 times a day and that the ratio of the minimum to maximum loading is the ratio of the anchor suction pressure to the design loading. This ratio for the charging load, R_c , will be approximately 0.09 for the prototype.

The magnitude of stress allowed in the structure is related to the number of loading cycles and the ratio R_c by the Wöhler curve. This curve finds the maximum amount of stress allowed in the material to uphold structural integrity over the course of the structure's lifetime. The expression for this stress is given below:

$$S := [1 - \beta \cdot (1 - R) \cdot \log(N)]$$

where β is a constant equal to .0685, and S is endurance strength, which is the fraction of the maximum stress in the system over the compressive strength of the construction material. The corresponding preliminary maximum stress can be found with the relation below:

$$S := \frac{\sigma_{\max 1}}{f_c}$$

where $\sigma_{\max 1}$ is the preliminary maximum stress allowed in the structure and f_c is the construction material's compressive strength. To take into account a safety factor and the mean stress of the cycle, the Goodman relation can be used to further define the maximum allowable stress in the system. The maximum allowed stress allowed in the structure will therefore be:

$$\sigma_{\text{design}} := \sigma_{\max 1} \left(\frac{1}{C_7} - \frac{\sigma_m}{2 \cdot f_c} \right)$$

where σ_{design} will be the design stress in the structure and C_7 is the safety factor. For the prototype, the safety factor will be assumed to be 3. The maximum stress in the structure will therefore be 11.7 MPa.

3. Thin Pressure Vessel Approximations

To now determine the thickness of the structure, thin pressure vessel approximations for a sphere will be assumed. Again, mooring loads will not be taken into consideration for finding this thickness; however, future designs must take this into account. Given the material design stress (σ_{design}), the design loading (P_{design}), and the radius of the chamber, thin pressure vessel approximations designate that the thickness of the structure is given by the expression below.

$$t_{\text{design}} := \frac{(P_{\text{design}} \cdot r_{\text{chamber}})}{2 \cdot \sigma_{\text{design}}}$$

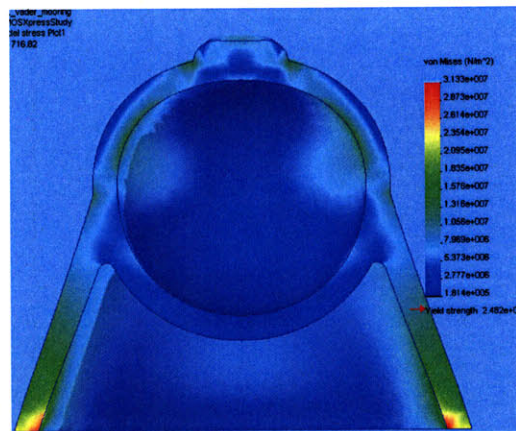
where t_{design} is the thickness of the structure. For the prototype, the design thickness will therefore be 2.5 cm. Therefore, the amount of steel needed for the beta level prototype is 0.58 m³.

VIII. DIMENSIONS AND COST OF FINAL ESD PROTOTYPE

The final dimensions for the ESD structure are given in the table below. It is important to note that these dimensions do not account for the uplift mooring forces that will be exerted on this structure. However, the theory designed in the previous section has the capability of incorporating this into the structure.

Prototype Dimension	Value
Water Depth	30 m
Energy Stored	3 kWh
Storage Volume	10 m ³
Storage Tank Radius	1.33 m
Anchor Radius	1.65 m
Skirt height	1.65 m
Structure Thickness	2.5 cm
Amount of Steel Needed	4.46 tonnes
Cost of Structural Steel	\$3033.17
Cost per kWh	\$1011.58

Table 3: Dimensions and Costs of Final Prototype



IX. FUTURE WORK

More work will be needed before the prototype can be built. Specific tasks are enumerated below.

- **Floating Buoy Dynamics**
The loads associated with these dynamics must be defined and checked against the capabilities of the designed structure. Potentially, reinforcement may need to be added to the areas connecting the cable lines to the ESD.
- **Organic Marine Growth**
The amount of marine growth around the intake channel must be further investigated. Furthermore, the potential risks involved with marine growth inside the chamber must be investigated as well. Finally, methods to address the settling of solid matter inside

the chamber will have to be further investigated to make accommodations for the final volume of the chamber.

- **Material Selection**

Although low carbon steel has been chosen to be the ESD construction material, more investigation will be to find the specific allow. This involves exploring welding capabilities and bending technologies and analyzing the tradeoffs between different alloys.

- **Mooring Line Placement**

Where the mooring lines will be attached to the structure must be determined, and the design of the structure must be modified to accommodate their loadings.

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