SANDIA REPORT
SAND2014-18082
Unlimited Release
Printed September 2014

Oscillating Water Column Structural Model

Guild Copeland, Diana Bull, Richard Jepsen, and Margaret Gordon

Prepared by
Sandia National Laboratories
Albuquerque, New Mexico 87185 and Livermore, California 94550

Sandia National Laboratories is a multi-program laboratory managed and operated by Sandia Corporation, a wholly owned subsidiary of Lockheed Martin Corporation, for the U.S. Department of Energy's National Nuclear Security Administration under contract DE-AC04-94AL85000.

Sandia National Laboratories

Issued by Sandia National Laboratories, operated for the United States Department of Energy by Sandia Corporation.

Printed in the United States of America. This report has been reproduced directly from the best available copy.

Available to DOE and DOE contractors from
U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831

Telephone: (865) 576-8401
Facsimile: (865) 576-5728
E-Mail: reports@adonis.osti.gov
Online ordering: http://www.osti.gov/bridge

Available to the public from
U.S. Department of Commerce
National Technical Information Service
5285 Port Royal Rd.
Springfield, VA 22161

Telephone: (800) 553-6847
Facsimile: (703) 605-6900
E-Mail: orders@ntis.fedworld.gov
Online order: http://www.ntis.gov/help/ordermethods.asp?loc=7-4-0#online

SAND2014-18082
Unlimited Release
Printed September 2014

Oscillating Water Column Structural Model

Guild Copeland, Diana Bull and Richard Jepsen
Water Power Technologies
and
Margaret Gordon
Materials, Devices and Energy Technologies
Sandia National Laboratories
P.O. Box 5800
Albuquerque, New Mexico 87185-MS1124

Abstract

An oscillating water column (OWC) wave energy converter is a structure with an opening to the ocean below the free surface, i.e. a structure with a moonpool. Two structural models for a non-axisymmetric terminator design OWC, the Backward Bent Duct Buoy (BBDB) are discussed in this report. The results of this structural model design study are intended to inform experiments and modeling underway in support of the U.S. Department of Energy (DOE) initiated Reference Model Project (RMP). A detailed design developed by Re Vision Consulting used stiffeners and girders to stabilize the structure against the hydrostatic loads experienced by a BBDB device. Additional support plates were added to this structure to account for loads arising from the mooring line attachment points. A simplified structure was designed in a modular fashion. This simplified design allows easy alterations to the buoyancy chambers and uncomplicated analysis of resulting changes in buoyancy.

ACKNOWLEDGMENTS

This work was funded by the Department of Energy's Wind and Water Power Technologies Office. The work was in support of the DOE sponsored Reference Model Project (see http://energy.sandia.gov/rmp).

The authors would like to acknowledge Re Vision Consulting for the detailed structural design of the BBDB and load calculations for the BBDB model device.

Rich Jepsen modeled the mooring loads, and worked on the Finite Element Analysis determination of the resulting structural design modifications necessary to support those loads.

Guild Copeland in collaboration with Diana Bull designed the simplified structural model and drafted the original report on both the detailed structural design and the simplified structural design.

Margaret Gordon was responsible for editorial work bringing these studies together.

CONTENTS

FIGURES

TABLES

NOMENCLATURE

BBDB	backward bent duct buoy
BEM	boundary element method
DOE	Department of Energy
FSE	free surface elevation
OWC	oscillating water column
PCC	power conversion chain
RAO	response amplitude operator
SNL	Sandia National Laboratories

INTRODUCTION

An oscillating water column (OWC) wave energy converter is a structure with an opening to the ocean below the free surface, i.e. a structure with a moonpool. The area above the moonpool is enclosed to create an air chamber which is open to atmosphere through a turbine. The turbine, with its associated control strategy, 'links' the pressure fluctuations in the air chamber to the power produced by the device.

Two structural models for a non-axisymmetric terminator design, the Backward Bent Duct Buoy (BBDB) (Masuda, Yamazaki, Outa, & McCormick, 1987), are discussed in this report. The results of this structural model design study are intended to inform experiments and modeling underway in support of the U.S. Department of Energy (DOE) initiated Reference Model Project (RMP).

DETAILED STRUCTURAL MODEL

The starting point for our original reference model for the BBDB device was the design produced by Re Vision Consulting which included the load calculations for the model. Ideally the structure would be designed to withstand the dynamic loading resulting in an extreme environment. This dynamic loading would be a combination of nonlinear dynamic pressure, green water (water on top of the structure), and/or slam loads from waves crashing on top of the structure or the structure hitting the surface of the water. However, all of these loads are highly nonlinear, and the tools to assess these loads (see RANS solvers, LAMP, AEGIR) are beyond the scope of the Reference Model project. These tools are best applied at higher WEC TRLs, like WEC TRL 5, as described in (Ruehl & Bull, 2012).

Design Pressure

Without these tools or experimental data, a design load had to be estimated in order to more fully understand a more realistic structural design. A design load was required that corresponded to the hydrostatic pressure with a green water depth of 6.0 m. This green water depth is applied to the entire structure and the lowest point is used to uniformly design the structural requirements. Eq.'s **1** and **2** below more fully describe the calculation of this design load.

$$D = dd + dg = 17.5 \text{ m} + 6.0 \text{ m} = 23.5\text{m} \qquad 1$$

Where *D* is the design depth, *dd* is the maximum device draft and *dg* is the green water depth. The design depth is then used in the calculation of the design pressure as shown below:

$$DP = D * \rho * g = 23.5 \text{ m} * 10.25 \frac{\text{kg}}{\text{m}^3} * 9.81 \frac{\text{m}}{\text{s}^2} = 236300 \frac{\text{N}}{\text{m}^2}. \qquad 2$$

Where *DP* is the Design Pressure, ρ is the density of sea water and *g* is the gravitational constant. Although simplistic, this severe hydrostatic loading should be conservative enough to account for the dynamic loading expected in extreme events.

(Note: RM3(Neary et al., 2014) underwent small scale experimental testing to derive the survival loads. To date four distinct sets of loads have been used as the survival loads. In the official RM3 report 8500 kN was reported to be applied to the entire structure (this would be 29,732 N/m^2 on the float and 12,526 N/m^2 on the heave plate). Other reports have used distinct extreme loading estimates that were also supplied by the experimental testing campaign as the design loads: 1,417 kN on the float (4,957 N/m^2), 2,275 kN on the heave plate (3,353 N/m^2), and 241,317 N/m^2 on the spar. Regardless of which of these estimates you focus on, the dynamic pressure is a factor of 10 to 100 less than the hydrostatic submergence predicted in Eq. **2**. A change of this scale for the design condition will dramatically alter the recommended design. Currently it is not clear what the best design load would be. However since the design needed little alteration to handle the loads at the mooring connection points, it is likely that the hydrostatic design pressure is a much more conservative estimate than should be realistically applied.)

Stiffener Spacing

With an estimate of the design pressure, Re Vison was then able to use the Det Norske Veritas (DNV) Rules for Classification of Ships: Hull Structural Design, Ships with Length Less Than 100 Meters, Part 3(Veritas, 2009) to specify the correct relationship between plate thickness and stiffener spacing. The equation for plate thickness, t, for a structure less than 100 m as specified by DNV is shown in Eq. **3**.

$$t = \frac{15.8 * K_a * s * \sqrt{p}}{\sqrt{\sigma f_1}} + t_k \quad (mm) \qquad 3$$

where K_a is a correction factor for aspect ratio of plate field (minimum value of 0.72; maximum value of 1.0), s is the stiffener spacing, p is the design lateral pressure, σ is the allowable local stress (assumed as 2/3 of yield stress, DF = 1.5), f_1 is the material factor (1.0) and t_k is the corrosion addition in mm. Re Vision used ASTM A36 steel for its model so material characteristics should be consistent with this. **Error! Reference source not found.** shows the values Re Vision used in calculating stiffener spacing.

Table 1: Stiffener Spacing Parameters.

K_a=	1	(max value)
p=	236.5	kN/m^2
σ=	166.7	N/mm^2
f_l=	1.0	
t_k=	1.5	mm

This equation results in the linear relationship as shown in Figure 1.

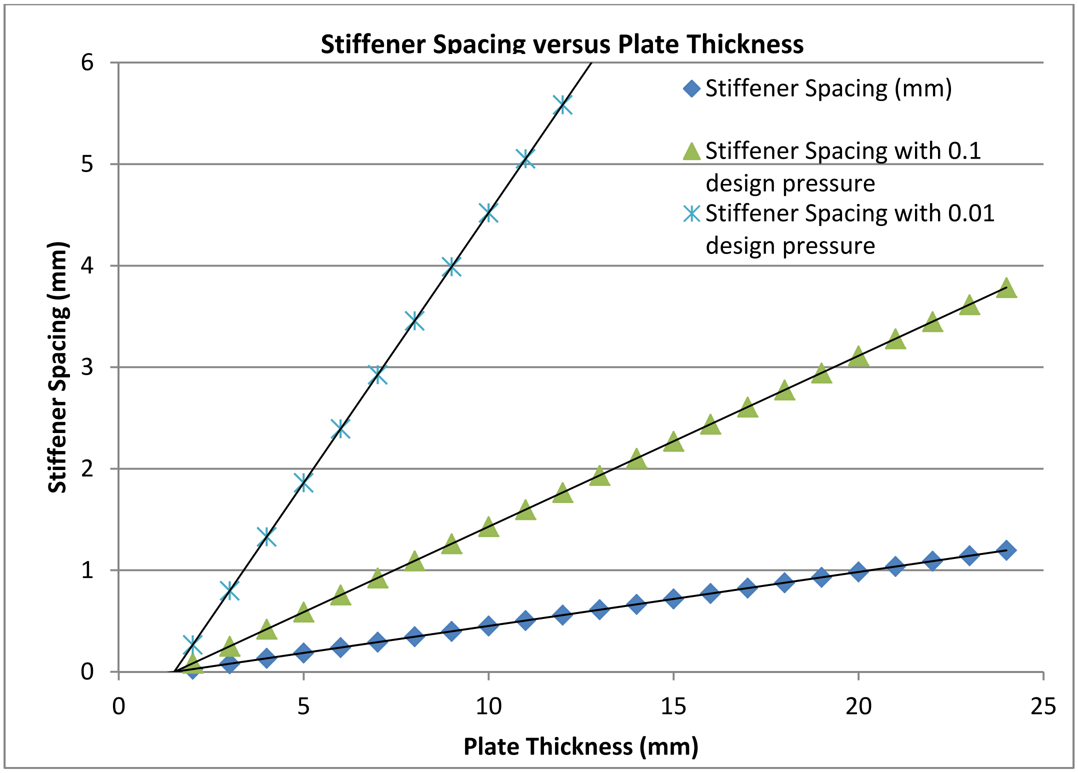

Figure 1: DNV relationship between stiffener spacing and plate thickness. The blue triangles illustrate the relationship used by Re Vision Consulting. The green triangles and blue stars illustrate how this relationship changes with the design pressure (values chosen specifically in relationship to RM3 reference case discussed above).

The relationship laid out in Eq. **3** and shown in Figure 1, blue triangles, is then able to specify the structural design required to withstand the design pressure. Choosing a plate thickness of 5/8" (15.875 mm) results in a calculated stiffener spacing of 0.7638 mm which was rounded to 0.75 mm in the model. Re Vision Consulting offered no explanation for the choice of this plate thickness.

(Note: For the same plate thickness, the stiffener spacing would grow from 0.75 mm to 2.4 mm for a factor of 10 reduction in design pressure and 7.7 mm for a factor of 100 reduction in design pressure).

Stiffener and Girder Sizing

A structure must stiffened both vertically and horizontally. There are distinct stiffening elements referred to as stiffeners and girders. Girders typically have a T-shaped cross-section and are the main horizontal and vertical support for a structure. The stiffeners are smaller in dimension and act locally as support between the girders.

Once the stiffener spacing is understood, the size of the stiffener must also be specified. Again, DNV (Veritas, 2009) was used to offer guidance on these relationships. Re Vision used the formula presented in Eq.**4** in calculating support element sizing.

$$\mathrm{Z} = \frac{1000 * l^2 * s * p * w_k}{m * \sigma * f_1} \quad (cm^3) \qquad 4$$

where Z is the section modulus, *l* is the length of member in m, *s* is the stiffener spacing in m, *m* is the bending moment factor, *p* is the design lateral pressure in kN/m^2, σ is the allowable local stress (assumed as 2/3 of yield stress), f_1 is the material factor (taken as 1.0), and w_k is the section modulus corrosion factor. Eq. **4** is also used to calculate the appropriate girder sizing. Re Vision used the values found in **Error! Reference source not found.** for Stiffener Sizing and the values in **Error! Reference source not found.** for Girder Sizing.

Table 2: Stiffener Sizing Parameters.

l	1.97	M
s	0.75	M
p	236.5	kN/m2
w_k	1.15	
m	8	
σ allow	166.7	N/mm2
f_1	1	

$$Z_{Required} = 597\ cm^3 \qquad 5$$

Here $Z_{Required}$ is the stiffener section modulus calculated to be necessary to provide the desired stiffness.

$$Z_{Actual} = 610\ cm^3 \qquad 6$$

Z_{Actual} is supplied by Re Vison as the true stiffener section modulus for their specific stiffener design executed in ASTM A36 steel. The Re Vision design uses stiffeners that are L-shaped and approximately 0.1 x 0.2 m with a cross sectional area of 0.0048 m^2. Figure 2 illustrates the stiffeners used in the Re Vision design.

Table 3: Girder Sizing Parameters.

l	4.5	M
s	1.97	M
p	236.5	kN/m2
w_k	1	
m	8	
σ allow	166.7	N/mm2
f_l	1	

$$Z_{Required} = 7093\ cm^3 \quad (7)$$

Here $Z_{Required}$ is the girder section modulus calculated to be necessary to provide the desired stiffness.

$$Z_{Actual} = 7162\ cm^3 \quad (8)$$

Z_{Actual} is supplied by Re Vison as the true girder section modulus for their specific girder design executed in ASTM A36 steel. The Re Vision design uses T-shaped girders which are approximately 0.4 x 0.4 m and have a cross sectional area of 0.032 m^2. Figure 2 illustrates the girders used in the Re Vision design.

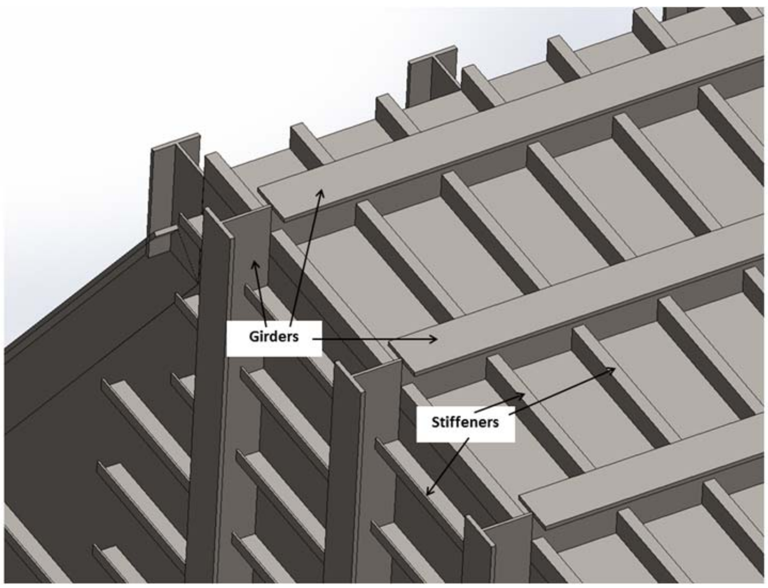

Figure 2: Illustration of girders and stiffeners used in Re Vision design.

Structural design remarks

As stated above, these calculations are able to yield an estimate of how the structure could be built. Re Vision used the calculations detailed above to produce a SolidWorks model from which physical characteristics could be computed including total weight, center of gravity, centers of inertia and reserve buoyancy. Figure 13 illustrates the structural model that Re Vision designed. This original model was created as a single SolidWorks part hence making alterations to the design difficult. **Error! Reference source not found.**, **Error! Reference source not found.** and **Error! Reference source not found.** in the next section identify the physical characteristics of this structural design.

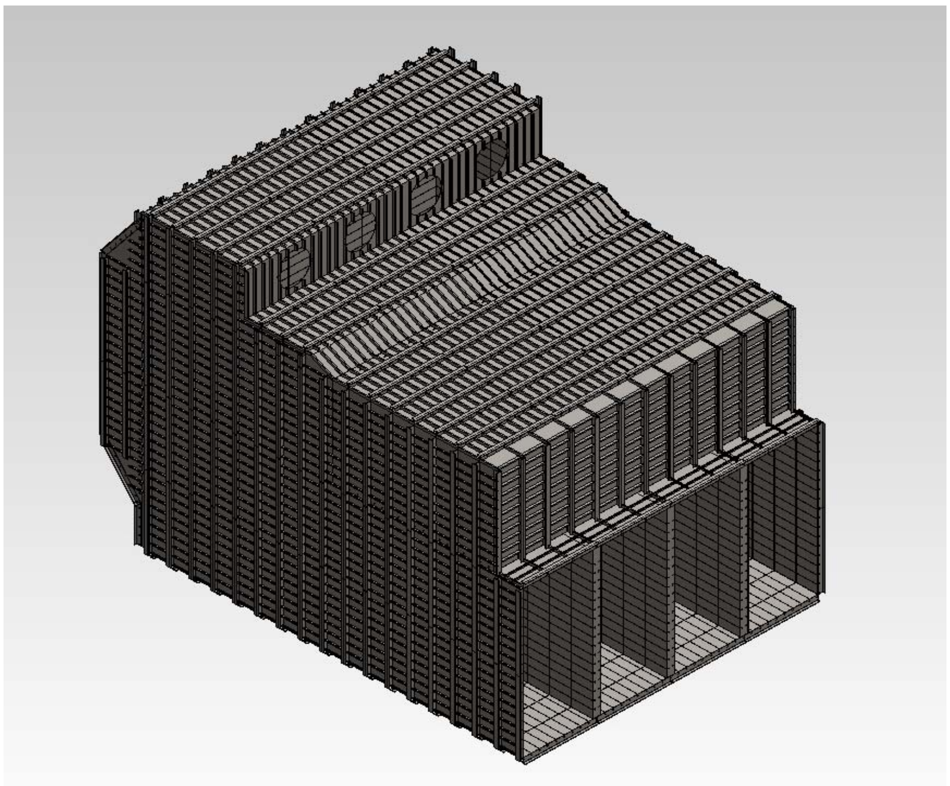

Figure 3: Detailed Structural Design Model.

However, the applicability of the chosen design pressure and the DNV standards to this particular design are not well understood. The design pressure was applied to the entirety of the structure as opposed to applying distinct pressure regimes to distinct areas. Again, the applicability of this choice is beyond the scope of the Reference Model project. Additionally the plate thickness selected by Re Vision does not appear to be a carefully motivated design choice.

Thus, although a design is presented here, this design is not promoted as the most economical or efficient. It is intended to be conservative and to highlight the beginning steps one would take to design a WEC structure. It is known that this reference device is much heavier than commercial analogs (i.e. Ocean Energy Ltd. (Ocean Energy Ltd., n.d.)).

STRUCTURAL MODIFICATIONS FROM MOORING LOADS

In addition to withstanding the loads presented by hydrostatic pressure, the BBDB structure must also be designed to withstand the load forces presented by the mooring lines at the points of attachment.

The mooring loads were determined by Sandia with a three line configuration and using Orcaflex in extreme wave conditions for survival loads. Figures 4 and 5 show the mooring line configuration in plan and side views as modeled in Orcaflex, respectively. Attachment points were placed at 8.75 m above the bottom of the structure for the Orcaflex analysis. This provided the maximum load and direction in which the mooring lines would act on the OWC structure.

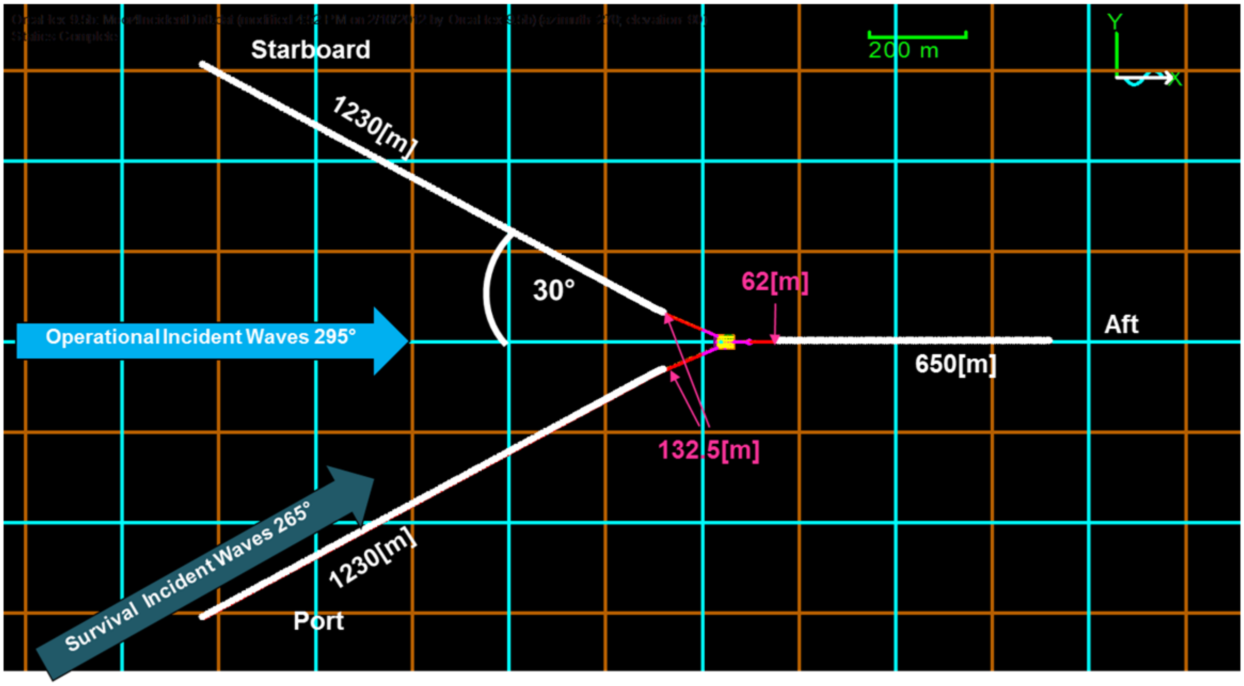

Figure 4: Plan view of mooring line configuration in Orcaflex.

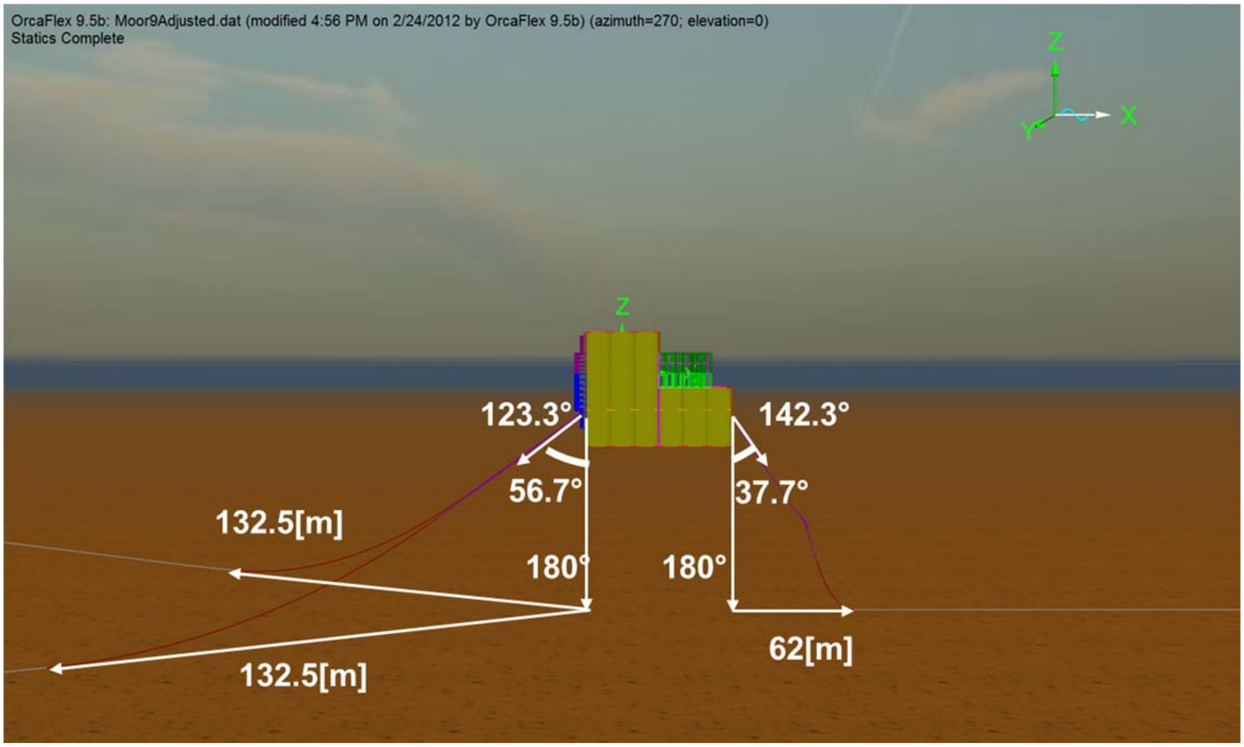

Figure 5: Side view of mooring line configuration in Orcaflex.

From the Orcaflex analysis, the maximum load 2318 kN is applied on the structure from the mooring line at the port side attachment at about 90 degrees. This is, by far, the highest load the structure experiences from mooring constraints and occurs with the survival incident wave condition shown in Figure 5. Table 4 shows the loads for each mooring line during the survival condition.

Table 4: Mooring loads during survival wave conditions.

Loading--Waves, Wind, and Current at 30° from Operational Heading (265°, along port leg)--Last Wave											
			Port			Starboard			Aft		
OWC Connection Point	Maxium Tension @ Angle	[kN]	2318	[kN] @ [°]	94.4	1177	[kN] @ [°]	113.5	1292	[kN] @ [°]	114.5
	Minimum Tension @ Angle	[kN]	1127	[kN] @ [°]	102.83	169	[kN] @ [°]	101.82	-152	[kN] @ [°]	115.76
	Maxium Declination @ Tension	[deg]	110.5	[°] @ [kN]	1796	114.5	[°] @ [kN]	1121	131.4	[°] @ [kN]	83
	Minimum Declination @ Angle	[deg]	93.5	[°] @ [kN]	2224	97	[°] @ [kN]	518	103.5	[°] @ [kN]	4

OWC Structural Simulation-Port and Starboard attachments

For model simulation of structural loads to the OWC, the port side tension load was applied at an attachment point 8.75 m above the bottom of the structure since it was the largest of the mooring loads. Initial model simulations demonstrated that there was little contribution from DNV specified girders and struts necessary for hydrostatic structural needs towards resisting the port side load of 2318 kN. Because of large computational time in meshing and solving with the more complex structure associated with the girders and struts, it was decided to remove these features and focus on designing the support assembly that would survive the large load applied by the mooring line within a factor of safety of 1.5. Although the DNV specified supports do provide some load spreading into the OWC vessel, it is a conservative approach to remove these and

Figure 6 shows the portion of the vessel that was modeled relative to the overall structure. This is the plate that houses the rear buoyancy chamber for the OWC vessel.

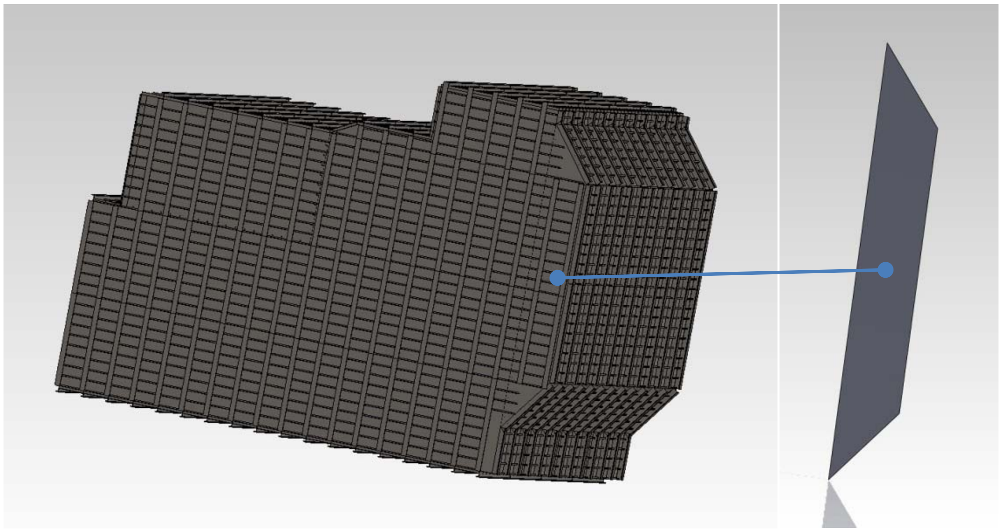

Figure 6: OWC buoyancy chamber plate where mooring attachment point is located.

The plate shown in Figure 6 was shown to not be sufficient in size or material strength to withstand the maximum 2320 kN load applied by the mooring line. Therefore, supporting plates were added to the buoyancy chamber plate with a geometry similar to the load spreading that resulted from the preliminary simulations in which the plate failed. Figure 7 shows the supporting plates that were optimized and simulated for the minimum additional size and weight needed to accommodate the required factor of safety.

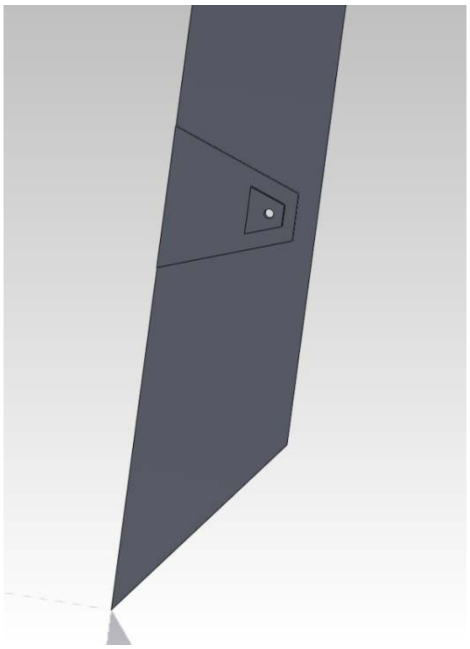

Figure 7: Additional plates to support the maximum mooring load. Attachment point also shown.

The simulation was done using Solidworks Simulation static analysis with a peg inserted into the attachment hole with no contact defined. The mooring load was applied as a bearing load at the attachment point. The back and top edges of the buoyancy chamber plate were fixed. Figure 8 shows the load and fixed locations on the plate for the simulation.

The mesh used for the simulation was a solid curvature based mesh with four Jacobian points. The maximum element size was 0.15 m and the minimum size was 0.05 m. Mesh control was used to refine the elements near the mooring attachment (outer cylindrical edge) with a defined size of 0.025 at the bearing location with a ratio of 1.5 extending away. The total nodes in the domain were 48,156 and the total elements were 23,691. Figure 9 shows the mesh for the simulation.

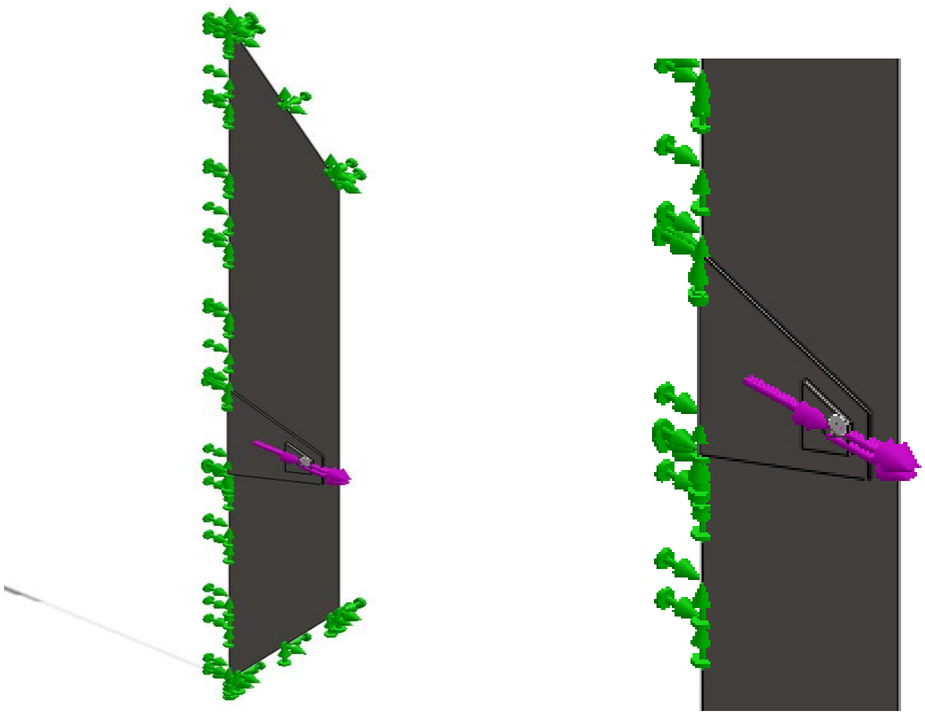

Figure 8: Full and enlarged view for the buoyancy plate with fixed and load bearing locations shown.

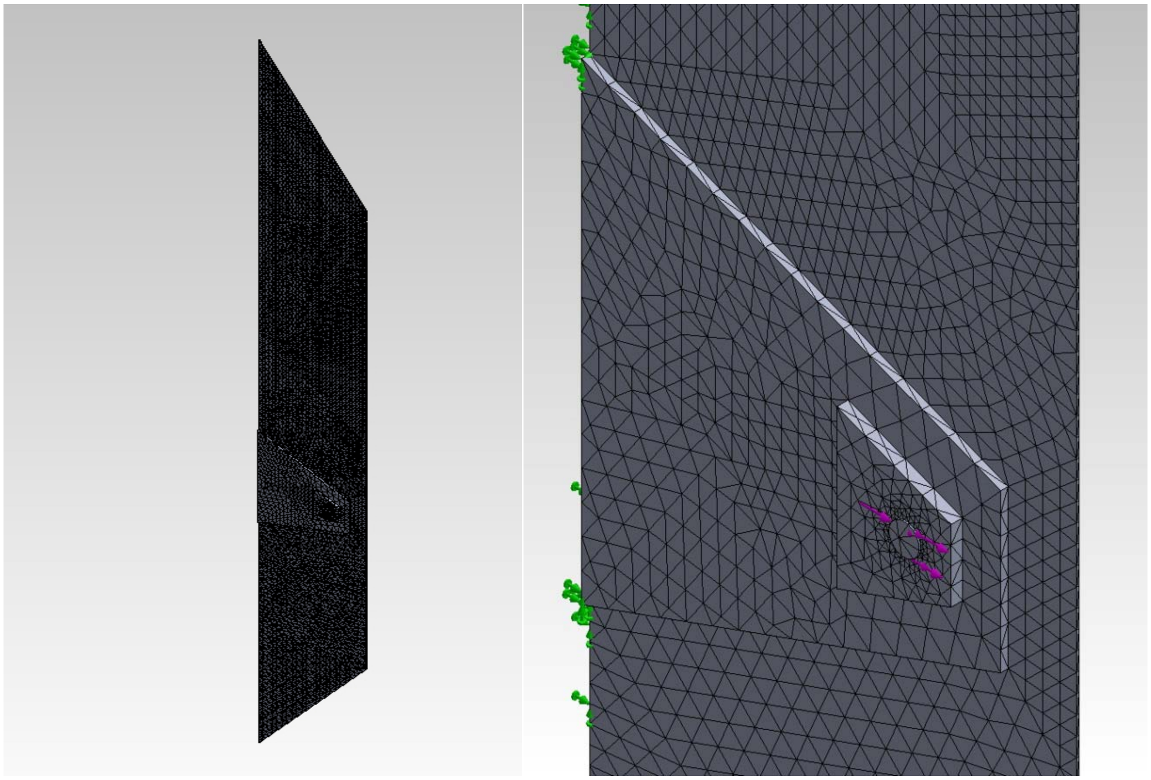

Figure 9: Full and enlarged view for the buoyancy plate simulation with mesh shown.

Model Simulation Results

Port and Starboard Moorings

The results of the simulation (Figure 10) demonstrated that the intermediate support plate had stresses less than half of the yield strength and provided the necessary factor of safety of at least 1.5. However, there are stress concentrations on the outer support plate along the concentric edge in which the load is applied that are near 60 ksi. Several plate thicknesses and geometries were attempted to reduce this stress and none were successful. It appears the only method to accommodate this stress is to use steel with a material strength significantly greater than the A36 steel used elsewhere in the design of the OWC. The results show that 120 ksi steel would be sufficient in this application. Since the outer most plate is quite small compared to the rest of the OWC structure, it is reasonable to modify the material at this location only to survive the larger mooring load applied at this point. It should also be noted that steel typically used in rigging and mooring applications (shackles, bolts, etc.) is 180 ksi in yield strength.

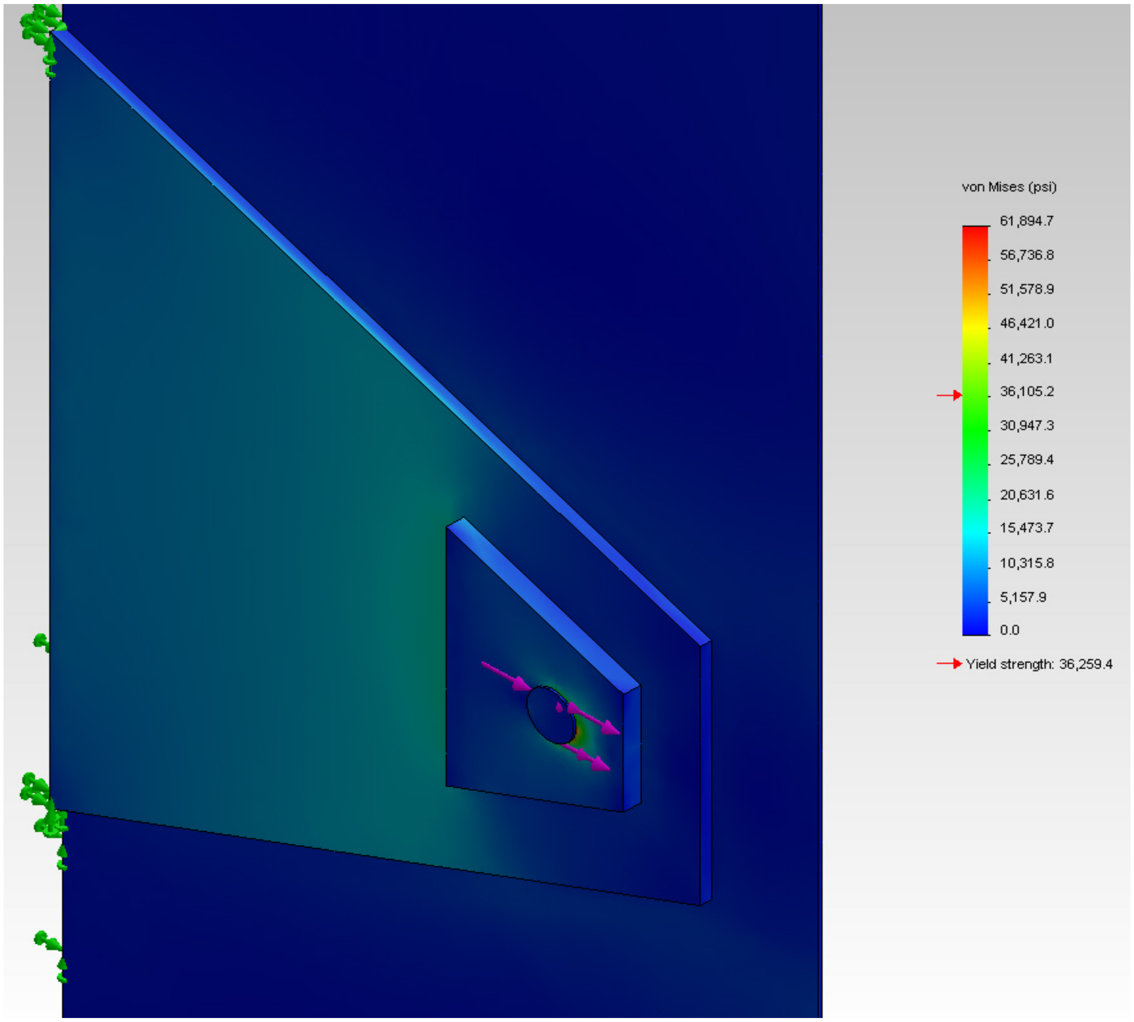

Figure 10: Simulation results for a 2320 kN mooring load.

Aft Mooring

The maximum mooring loads at the aft location are almost half of that for the port side. Model simulations show that only the smaller, 120 ksi support plate is necessary to withstand the maximum load.

Material and Design Modifications

Additional materials and design needed to support OWC mooring loads are as follows. Several plate thicknesses and dimensions were attempted and the optimal sizes were determined. An intermediate plate of 2 5/8 inch thick A36 steel with dimensions shown in Figure 11 is needed to adequately spread the load to the rest of the buoyancy plate. The outer plate must be 120 ksi steel and 5 5/8 inches thick with the dimensions shown in Figure 12. The added weight compared to the original design from these plates is approximately 2,200 kg each for the port and starboard attachments and is virtually insignificant compared to the rest of the OWC structure (~2,000 tons). For the aft mooring location, only the smaller support plate is needed which weighs 250 lbs.

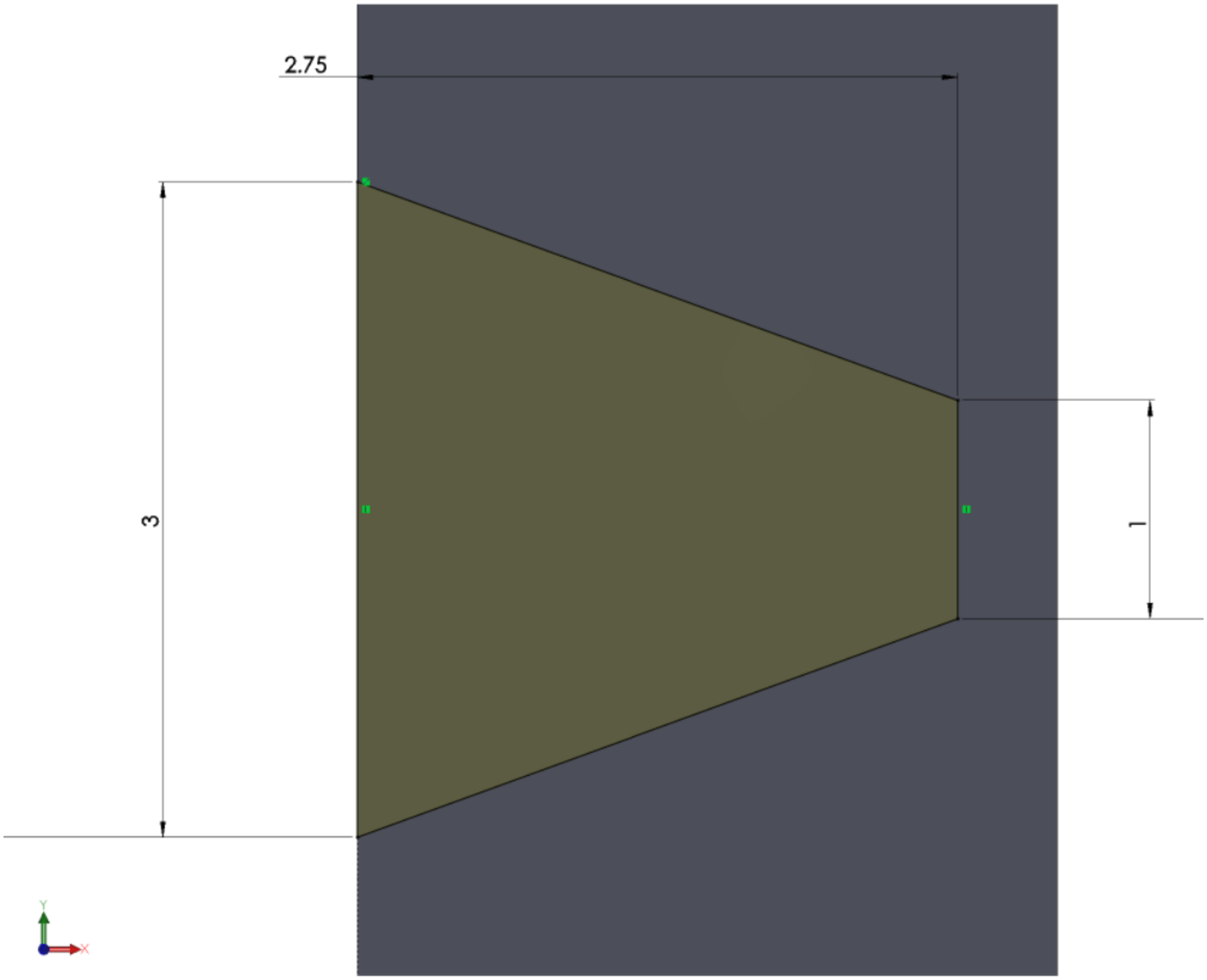

Figure 11: Dimensions of intermediate support plate in meters.

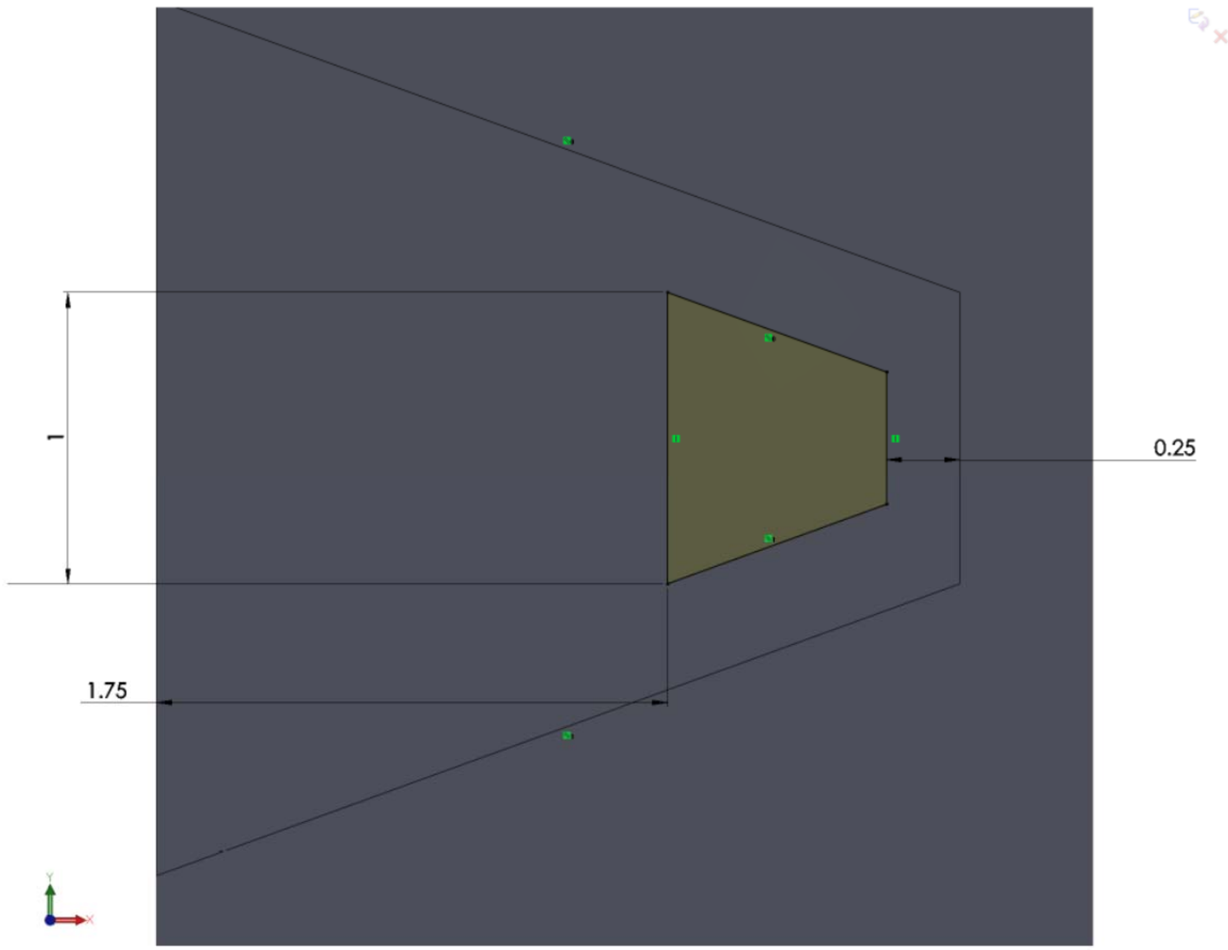

Figure 12: Dimensions of outer support plate in meters.

The installation of these plates would be inserted into cut-out sections of the hull plates such that through welds are achievable. In this way, the support plates spread the load throughout their volume and not just along the edges. The plates cannot be simply placed on top of the existing hull plates and welded along the edges. This is because the loads at the welds would be above their yield strength.

SIMPLIFIED STRUCTURAL MODEL

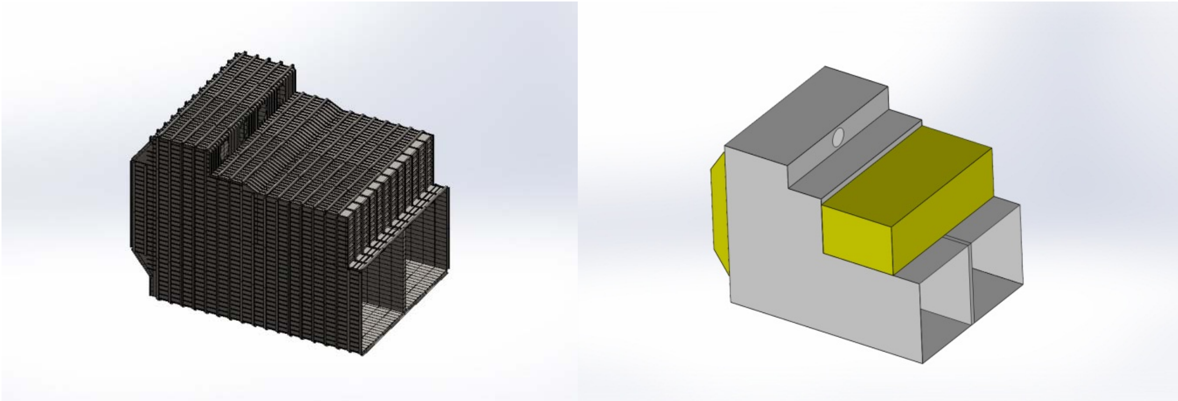

Figure 13: **Original Model** **New Model**

The motivation for creating a new simplified model was the need to represent the structure in a simple modular fashion so that alterations of the buoyancy chambers would be easy to implement and the analysis of the resulting changes in buoyancy be uncomplicated. The first step was to replace the original OWC model which was created as a single SolidWorks part with a simplified SolidWorks assembly that would have equivalent mass and centers of gravity and buoyancy. Part of this simplification was to replace the original structure's plates, girders and stiffeners with a basic plate-only construction that had an equivalent mass. Our approach was to use stainless steel plate for the model and adjust the thickness of this plate to give the same mass per unit surface area as the original plate/girder/stiffener construction.

Plate Only construction

To calculate a representative mass per unit area for the original design, two different approaches were compared: First, models of portions of the original structure were created in SolidWorks using the same plate/girder/stiffener construction. The masses for these models were calculated and a mass per unit area was computed. Second, a representative unit cell was generated in SolidWorks based on the smallest repeating element of the plate/girder/stiffener pattern. Again, a mass and a mass per unit area were calculated. The results from these two approaches are shown in Table 5.

Table 5: Comparison of Mass for Unit Cell versus Plate/Stiffener/Girder Construction.

	Unit Cell	Bottom Assembly	Front Buoy. Chamber, Upper Plate	Front Buoy. Chamber, Middle Plate	Duct Structure Back Plate
Width(m)	2	34.0254	27	27	27
Height(m)	0.75	27	5.1225	15	8.5
Area(m^2)	1.5	918.6858	138.3075	405	229.5
Mass(kg)	413.76	260136.93	36842.67	111171.71	62118.45
Mass/Area(kg/ m^2)	275.84	283.1620234	266.3823003	274.4980494	270.66863
Mass/Area, % change from Unit Cell	na	2.65%	-3.43%	-0.49%	-1.87%

Given that the mass per area of the plate/girder/stiffener models varied slightly from one section to another and that they all fell reasonably close to that of the unit cell, calculating the thickness of the stainless steel plate in our model using the mass/area of the unit cell was a reasonable approach. The unit cell had a mass/area of 275.84 kg/m^2. A typical density for type 316L stainless steel is 8027 kg/m^3. Thus a thickness of 0.0344 m would result in stainless steel plate that had the desired mass per area of 275.8 kg/m^2. By using 0.03436 m thick 316 stainless steel in our model we maintain the same mass as that of the structural components of the original construction. Once the thickness was determined this uniform stainless steel plate was used throughout the SolidWorks model except for the Duct Bulkhead section.

The Duct Bulkhead was modeled with its own original plate/girder/stiffener construction to get a calculated mass for this 14 x 35 x 0.75 m structure of 179549.475 kg. In the new model the Duct Bulkhead was a single solid stainless steel plate with lateral dimensions of 14 x 35 m. Using the same density of 8027 kg/m^3, the plate thickness required to yield the correct mass was calculated to be 0.04565 m.

Going forward, for consistency with other Reference Models, we will substitute ASTM A36 steel for 316 stainless steel. The density of A36 is 7850 kg/m^3 versus a density of 8027 kg/m^3 for 316 stainless. To maintain the same mass per area of plate used in the previous models the thickness of the plate must be increased for A36 steel. A new thickness of 0.0351 m yields the same mass/plate area of 275.84 kg/m^2 that was used in the 316 stainless steel models. Applying the correct density of A36 steel to the 14 m x 35 m Duct Bulkhead section, the plate thickness required to yield the correct mass of 179549.475 kg was calculated to be 0.0467 m.

Physical characteristics

Once the new model was generated the center of gravity (COG) and the center of buoyancy (COB) were evaluated in the SolidWorks software. The information from the original model data is not an exact analog to the newly generated data because the PTO was already included in this model whereas it is not included in the new model. The addition of the PTO in the original model accounts for an additional 40,000 kg. COG and COB for the new model and the original model are shown in Table 6 below. Table 7 compares the moments of inertia at the COG.

Table 6: Comparison of COGs for Orignal and Revised Models.

	Original Model	New Model	% Change
Mass (kg)	1980547	1807633	-8.7%
COG (x)	4.93m	4.93m	0%
COG (y)	0	0	na
COG (z)	-4.74m	-4.79m	1.0%
COB (x)	5.09m	5.12m	0.6%
COB (y)	0	0	na
COB (z)	-3.27m	-3.31	1.2%

Table 7: Comparison of Moments of Inertia for Orignal and Revised Models.

	Original Model			New Model			% Change		
Moments of Inertia at COG (kg* m^2)	3.23E+08	-13846	-3.1E+07	2.96E+08	0	-2.76E+07	-8.0%	na	-11.0%
	-138346	4.2E+08	-85222.9	0	3.84E+08	0	na	-8.6%	na
	-3.1E+07	-85222.9	4.31E+08	-2.76E+07	0	4.00E+08	-11.0%	na	-7.2%

In order for the device to draft at the desired location, the weight of the device must be equal to the displaced water, thus the device must weigh 2055234 kg. Hence, both the original model and new model designs have reserve buoyancy. Part of this reserve buoyancy will be used to counteract the weight of the PTO train. The additional reserve buoyancy will need to be neutralized through the use of ballast. Table 8 below compares the reserve buoyancy between the two models.

Table 8: Comparison of Reserve Buoyancy for Orignal and Revised Models.

	Original Model	New Model	% Change
Reserve Buoyancy (kg)	74687	247601	231.52%

Note that the large difference seen between the reserve buoyancies in Table 8 can partially be attributed to the inclusion of the Power Conversion Chain (PCC) in the original structural model. Additional changes are the result of the difference in mass between the two representations.

Power Conversion Chain

Mirko Previsic of Re Vision Consulting estimated the following weights for a 250 kW turbine PTO:(Previsic, 2012)

1. The conical fiberglass fairings and the blades are light weight and on the order of 4 metric tons.
2. Generator at 500 rpm (rated) is on the order of 30 kg/kW so a 250 kW machine is 7.5 metric tons.
3. The total air turbine weight located top-side of the structure is about 12 metric tons per turbine. Prorated for a megawatt (MG), the PTO is 48 mT.
4. Based on a picture and description of the Dresser Rand air turbine, the dimensions for a 250 kW machine is on the order of 4 m duct inlet diameter and 6 m height.
5. Frequency converter, step-up transformer and capacitor banks will add weight, but they can be placed into one of the buoyancy tanks, so you are free to place the weight where you want it. For to total 1 MW machine, this will be on the order of 20 mT.

Although a distinct turbine will be used, the estimates given above will be used for the PCC. Hence in the simplified model 70,000 kg was reserved for the PCC. This mass is centered in the y-direction on the device 7.96 m above the still water line. The x-location of this mass changes as a function of the length of the air column, it maintains a location of 36.43% of L(air) from the front face of the main duct.

The rest of the reserve buoyancy will be neutralized through the use of ballast in the form of seawater added to the buoyancy chambers. By adding ballast to select locations in the device, the COG and COB can be altered.

4. DESIGN SPECIFICATIONS

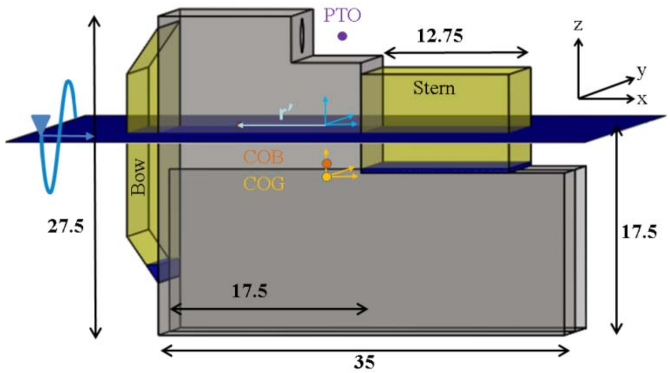

Figure 14: Model of the OWC describing dimensions, locations of principal components, locations of the COB and COG, and identifying coordinate systems.

The structural design assumes a uniform thickness of A36 steel, appropriate ballast mass and placement, and an estimate of the mass and location of the power conversion chain. An average wall thickness of 35.1 mm is applied to the entire device. This average thickness was derived from a structural design engineered to withstand the hydrostatic pressure at a submergence of 25 m. The ballast is distributed to obtain the desired draft and ensure that the center of gravity and the center of buoyancy are aligned vertically. The ballast is assumed to be seawater and is added to the buoyancy chambers as shown in Figure 14. The mass of the power conversion chain (drivetrain, generator, power conditioning electronics) is approximated and is placed at the expected center of the Wells Turbine location, also shown in Figure 14. Table 9 summarizes the structural properties of the device that are needed as input into WAMIT.

Table 9: Structural properties of the device.

Displaced Mass [kg]		2,024,657		
Structural Mass [kg]		1,808,944		
Bow Ballast Mass [kg]		22,072		
Stern Ballast Mass [kg]		123,641		
Power Conversion Mass [kg]		70,000		
COG (x,y,z) [m]		0.00	0.00	-4.29
COB (x,y,z) [m]		0.00	0.00	-3.31
Free Surface Center (x,y,z) [m]		-5.12	0.00	0.00
Radius of Gyration at COG [m]	x	12.53	0.00	0.00
	y	0.00	14.33	0.00
	z	0.00	0.00	14.54

5. REFERENCES

Masuda, Y., Yamazaki, T., Outa, Y., & McCormick, M. (1987). Study of Backward Bent Duct Buoy. In *OCEANS '87* (pp. 384–389). doi:10.1109/OCEANS.1987.1160750

Neary, V., Previsic, M., Jepsen, R., Lawson, M., Yu, Y.-H., Copping, A., … Murray, D. (2014). *Methodology for Design and Economic Analysis of Marine Energy Conversion (MEC) Technologies* (Sandia Report No. SAND2013-9040). Sandia National Laboratories.

Ocean Energy Ltd. (n.d.). Ocean Energy OEBuoy: A Backward Bent Duct Design. Retrieved April 4, 2013, from http://www.oceanenergy.ie/

Previsic, M. (2012, January 10). Weight estimate for PTO.

Ruehl, K., & Bull, D. (2012). Wave Energy Development Roadmap: Design to commercialization. In *Oceans, 2012* (pp. 1–10). doi:10.1109/OCEANS.2012.6404795

Veritas, D. N. (2009). Hull structural design ships with length 100 metres and above. Retrieved from https://dspace.ist.utl.pt/bitstream/2295/108867/1/D

DISTRIBUTION

1	MS0734	Margaret Gordon	6124 (electronic copy)
1	MS1124	Diana Bull	6122 (electronic copy)
1	MS1124	Guild Copeland	6122 (electronic copy)
1	MS1135	Rich Jepsen	1534 (electronic copy)
1	MS0899	Technical Library	9536 (electronic copy)

Sandia National Laboratories